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ANALYSIS OF THE DISTRICT HEATING STEAM TURBINE UNIT AT CHPP BASED ON ENERGY AND EXERGY INDICATORS

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A 20 MW district heating and condensing steam turbine unit (STU), consisting of the parts of high, medium and low pressure, which is operated at one of the CHPPs in Kharkiv, has been analyzed. According to the scheme, steam from two district heating recovery regulated extraction units is supplied to two network heating units. During the reconstruction of the CHPP, the heaters of the STU regeneration system were dismantled due to their degradation. It has been decided to focus on increasing heat recovery at the CHP plant, so no new high-pressure heaters were installed. In addition, instead of a cooling tower, it was decided to use water from the network for hot water supply in the condenser cooling circuit. An analytical review of the thermal scheme of a district heating STU in terms of energy and exergy indicators is given in the paper, which allowed to identify elements with high exergy cost, which is an indicator of their efficiency. Analytical tables with the exergy parameters of the original scheme element by element and analytical graphs were compiled during the analysis of the options of the thermal scheme of STU. According to the exergy analysis, the highest exergy cost is observed in the energy boiler, but it is known that it can be reduced by reducing exergy destruction in other elements. Therefore, a network heater that is heated by high-pressure steam from the first medium-pressure part of the first extraction was chosen as the element with the greatest potential for increasing the efficiency of the STU. Respectively, the first network heater, which is heated by low-pressure steam from the second low-pressure section, is selected as the second element. We also considered options of the thermal scheme of STU, in which the steam parameters in the turbine extractions (pressure, flow) were varied. It is shown that with a decrease in pressure and a decrease in steam flow in the first extraction, as well as a decrease in steam pressure and an increase in steam flow in the second extraction, the cost of exergy flows in network heaters decreases by almost 5%, the exergy efficiency of the STU increases by 2%, and the electrical efficiency of the unit increases by 2.16% compared to the original scheme.

***Keywords**: exergy, steam turbine unit, exergy cost, efficiency, CHPP.*

Introduction

One of the main directions of development of thermal power engineering is to increase the unit capacity of CHPP equipment. However, the possibilities of its increase, and therefore ones of district heating turbines, are limited compared to condensing turbines, since the transfer of thermal energy requires greater costs than the transfer of electrical energy. The unit capacity of a CHPP is determined by the concentration of heat consumption and the optimal size of the area connected to the CHPP for this concentration, as well as existing restrictions on environmental protection, site selection, etc.

When designing a power plant, those turbines that are mass-produced are usually chosen:

– for condensing thermal power plants – "K"-type turbines;

– for CHPPs (depending on the type of prevailing heating load), the following options are possible: when the heating load dominates, condensing "T"-type turbines are installed; when there is an equal amount of heating and technological load, condensing "PT"-type turbines are installed, and with a stable technological load of industrial enterprises, backpressure "P"- and "PR"-type turbines are installed; with significant technological loads of industrial enterprises and large, changing heating loads, it is possible to install all of the above types of turbines, combined into a CHPP section, which allows for a quick response to changing thermal loads.

The main reasons for the decrease in the efficiency of CHPPs are, firstly, a significant reduction in electricity generation for heat consumption, and secondly, significant losses in the transportation of coolants (hot water and process steam).

At the same time, most operating CHPPs have significant energy efficiency reserves associated with ensuring internal heating loads. A significant share of these loads falls on water treatment plants, which replenish both steam and condensate losses from the station cycle and network water from the heating network pipelines. The main drawback of technologies for ensuring heating loads of water treatment plants used in CHPPs is the use of high-potential steam extractions as a heating working fluid, which significantly reduces the share of electricity generation for heat consumption, and therefore, the efficiency of the power plant.

Thus, the determination of energy efficiency reserves and their involvement in the process of generating heat and electricity to increase the efficiency of CHPPs remains the current problem.

Analytical review of methods for analyzing thermal schemes of steam turbine unit

The production of electric and thermal energy worldwide is currently based mainly on steam turbine technologies [1–4].

In [2], thermal schemes of steam turbine unit (STU) were analyzed using the traditional technical and economic analysis method. Rational thermal schemes of STU operating in the condensation mode were determined. This method can be successfully used in the analysis of schemes that produce one useful product, such as electric energy in this case.

In [3], an energy analysis of the main and auxiliary steam turbines of a traditional coal-fired power plant was performed, as well as an analysis of energy flows from the main turbine cylinders to the condensate/feedwater heaters, which can later be the basis for detailed study and optimization. The energy consumption in thermal schemes of steam turbine plants was analyzed in this paper, but no economic analysis was performed. As in the previous paper, the generation of electric energy was considered. The same schemes of condensing STU were the subject of attention in the paper [5], the optimization of circuit solutions is carried out purely according to energy characteristics [6].

It is worth noting that in cases where a cogeneration system is considered, that is, the plant produces electric and thermal energy, the technical and energy analysis does not take into account the quality of energy. It is known that the measure of energy quality is exergy. Thus, electric energy is recognized as the energy of the highest quality, and thermal energy – as of the lowest one [7]. Exergy analysis can be applied to any system, regardless of its complexity, and is the most effective method for assessing and improving thermodynamic efficiency, especially in complex systems. As shown in the paper [7], exergy analysis can stimulate and direct creativity, helping in the development of completely new concepts of energy conversion.

In addition, using exergy analysis, it is possible to assess the efficiency of energy conversion processes not only on the basis of thermodynamics, but also taking into account economic and environmental aspects and the impact of the processes under study [8]. This comprehensive approach to the use of energy resources has the definition of sustainable ways of using energy resources as one of the most important features [9].

There are also modern pinch analysis methods in combination with the exergy method for optimizing the STU regeneration system, which allows finding the parameters of the system with minimal costs [10].

The purpose of this paper is to analyze the thermal scheme of the district heating STU as part of the CHPP using traditional energy and modern exergy methods in developing proposals for increasing efficiency and its further modernization. The main objective of the study is to determine the rational parameters of steam in the turbine extractions used for district heating.

Characteristics of the thermal scheme of condensing STU installed at one of the CHPPs in Kharkiv

The schematic thermal diagram of the district heating and condensing PT-20-2.9/1.0 with a capacity of 20 MW, which consists of the part of a high (HPP), medium (MPP) and low pressure (LPP), is shown in Fig. 1. Heat release is carried out as follows: steam from two regulated district heating extractions is supplied to two network heating installations (NH1, NH2). During peak load times, hot water for heating is heated in a peak water-heating boiler (not marked on the diagram).

The regeneration system consists of two deaerators – low (LPD) and high pressure (HPD). The connection type is cascade (without the use of drainage pumps). The first steam extraction from the CHPP is mainly sent to NH2, but some of it is fed to HPD. During the reconstruction of the CHPP, the heaters of the STU regeneration system were dismantled due to their malfunction. It has been decided at the CHPP to focus on increasing the heat transfer, so the high and low pressure heaters were not replaced. In addition, it was decided to cool the condenser instead of the cooling tower with water from the network that goes to hot water supply.

Factory data for the PT-20-2.9/1.0 turbine is presented below.

Electric power N_e=20 MW.

The initial steam parameters are as follows: the steam pressure at the inlet to the HPP P_1 is 2844 kPa; the temperature is T_1=400 °C.

In HPD, the pressure is 456 kPa, in LPD – 122 kPa.

Estimated values of internal relative efficiency by compartments: η_{oi}^{HPP}=93%, η_{oi}^{MPP}=93%, η_{oi}^{LPP}=92%.

Isentropic efficiency of pumps η_i^P=85%.

Steam consumption per turbine in nominal mode D_1=44.4 kg/s.

Steam consumption in the first extraction D_3=8.33 kg/s, in the second one – D_6=33.4 kg/s.

To determine the steam and water consumption, the material and heat balance equations [3] are solved according to the scheme (Fig. 1).

The main technical parameters of the PT-20-2.9/1.0 turbine are also shown in Table 1.

Taking into account the throttling of steam in the regulating bodies of the HPP, the steam pressure at the inlet to the flow part is determined by the formula

$$P_2=P_1\cdot\eta_{th}^{HPP}=2844\cdot0.95=2702 \text{ kPa},$$

where η_{th}^{HPP} is the coefficient determining pressure loss during steam throttling.

We also take into account pressure losses (5%) when throttling steam at the inlet to the LPP and condenser. The steam pressure at the inlet to NH1 and NH2 is taken equal to the steam pressure in the extractions, pressure losses are not taken into account, just like the steam mass losses in the STU.

The pressure and enthalpy of steam in the extractions are shown in Fig. 2 and in Table 2.

The mechanical power (ideal and real) of the HPP is defined as

$$N^{HPP}=D_1\cdot(h_1-h_2),$$

where h is the steam enthalpy.

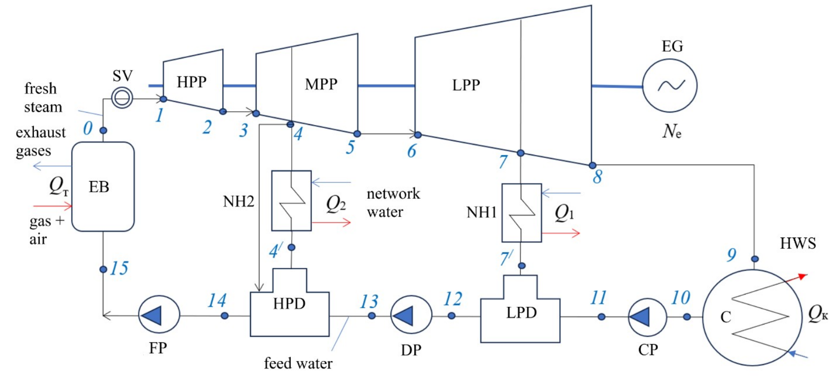

Fig. 1. Thermal scheme of a 20 MW STU:
EB – energy boiler; FP, DP, CP – feed, drainage, condensate pumps; HWS – hot water supply; SV – stop valve; EG – electric generator; C – condenser

Table 1. Main technical parameters of the PT-20-2,9/1,0 turbine

Parameter	Specification
Steam distribution type	nozzle
Turbine design	HPP+MPP+LPP
Number of stages (total), pcs.: – HPP, pcs. – MPP, pcs. – LPP, pcs.	11 4 4 3
Nominal speed, Hz	50
Number of steam extractions, pcs. – of which are adjustable, pcs.	2 2
Adjustable production steam extraction: – nominal pressure, kPa – pressure change range, kPa	 1231.7 980.7–1232.0
Heating steam extraction: – nominal pressure, kPa – pressure change range, kPa	 294.0 245.2–392.3
LPP exhaust area, m^2	0.764

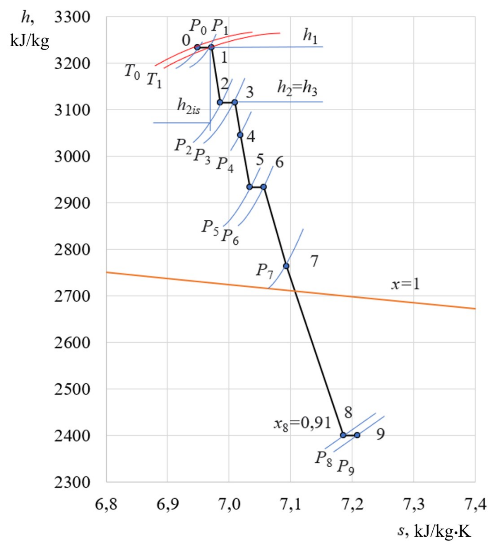

Fig. 2. Steam expansion process in a turbine

The steam enthalpy h_3 is derived from the equation

$$\eta_{oi}^{\mathrm{HPP}} = \frac{h_1 - h_2}{h_1 - h_{2is}}.$$

From Fig. 2 it is seen that the enthalpy h_{2is} for an ideal isentropic process is determined for the parameters of the pair P_3, T_3 at the entropy of s_1=s_2.

By analogy, the parameters of the pair for the LPP are determined.

The thermodynamic parameters at the points of the thermal scheme and the mass flow rates of the working fluid are shown in Table 2. The internal efficiency of the cycle η_i and the efficiency coefficient of the initial thermal scheme of the STU η_{unit}, taking into account the products in the form of electrical energy, thermal energy for heating and hot water supply (HWS) are shown in Table 3.

Table 2. Steam consumption and its parameters at cycle points (Fig. 1) of the initial thermal scheme of the STU (P_4=1232 kPa, G_4=8.33 kg/s, P_7=294 kPa, G_7=33.4 kg/s)

Steam/water flow name	Points	Parameters				
		Enthalpy i, kJ/K	Entropy s, kJ/(kg·K)	Temperature T, °C	Pressure P, kPa	Steam flow D, kg/s
Acute steam at the outlet of the EB	0	3233.0	6.9490	400.00	2844.00	44.40
Steam admission in the HPP	1	3233.0	6.9710	398.90	2702.00	44.40
Exhaust of the HPP	2	3115.0	6.9860	337.80	1729.00	44.40
Steam admission in the MPP	3	3115.0	7.0090	336.90	1643.00	44.40
Steam in the 1st extraction	4	3045.0	7.0180	300.20	1232.00	8.33
Exhaust of the MPP	5	2934.0	7.0340	242.30	755.00	36.07
Steam admission in the LPP	6	2934.0	7.0570	241.60	717.30	36.07
Steam in the 2nd extraction	7	2763.0	7.0930	150.90	294.00	33.30
Exhaust of the LPP	8	2400.0	7.1860	64.95	25.00	2.77
Steam admission in the condenser	9	2400.0	7.2080	63.81	23.75	2.77
Condensate at the outlet of the condenser	10	267.1	0.8787	63.81	23.75	2.77
Feed water after the CP	11	267.2	0.8788	63.82	122.00	2.77
At the outlet of the LPD	12	536.2	1.6170	105.30	122.00	2.77
Feed water after the DP	13	536.6	1.6100	127.70	456.00	36.07
Feed water at the outlet of the HPD	14	625.6	1.8260	148.40	456.00	44.40
Feed water at the inlet of the EB	15	628.9	1.8270	148.80	2994.00	44.40
Saturated steam in the NH1	7"	2764.0	6.7000	165.90	717.30	33.30
Condensate after the NH1	7'	701.6	1.0070	165.90	717.30	33.30
Saturated steam in the NH2	4"	2794.0	6.4110	202.70	1643.00	8.33
Condensate after the NH2	4'	864.4	3.1320	202.70	1643.00	8.33

Table 3. Energy characteristics of the initial thermal scheme of the STU

Power N, MW				Heat Q, MW			Electric power N_{STU}, MW	Cycle efficiency η_i, %	STU efficiency η_{unit}, %
HPP	MPP	LPP	CP+DP+FP	EB	NH1+NH2	C(HWS)	EG		
5.24	7.11	7.17	0.16	115.62	86.81	5.91	19.36	16.75	96.94

However, for a more accurate calculation of the electrical efficiency (net), the following losses should be taken into account:

$$\eta_{\mathrm{net}} = \eta_{\mathrm{EB}} \cdot \eta_i \cdot \eta_{\mathrm{m}} \cdot \eta_{\mathrm{EG}},$$

where η_{EB} is the efficiency, which takes into account losses in the energy boiler (0.98); η_i is the internal efficiency; η_{m} is the mechanical efficiency, which takes into account losses in the bearings and in the drive of the oil pump of the turbine unit (0.995); η_{EG} is the efficiency of the electric generator (0.98).

In this case, for the thermal scheme of the STU η_{net} will be 16%, and when calculating the efficiency of the district heating STU, the thermal efficiency of heat exchangers (HE) (0.95) should also be taken into account. Then $\eta_{\mathrm{net}}^{\mathrm{STU}} = \eta_{\mathrm{EV}} \cdot \eta_{\mathrm{unit}} \cdot \eta_{\mathrm{m}} \cdot \eta_{\mathrm{EG}} \cdot \eta_{\mathrm{HE}}$ =88%.

The energy analysis of the STU showed that the presented thermal scheme still has a low electrical efficiency (net). In view of this, an exergy analysis must be carried out to identify the elements that need to be modernized.

To study the interaction of the components of exergy losses in the elements of the STU, it is necessary to use the basic equations of the theory of exergy value [11]. Each thermoeconomic model of the system is based on the equation of the exergy balance, which is written for the elements of the thermodynamic system as

$$\sum E_k^{\text{ent}} - \sum E_k^{\text{exit}} = E_{Dk} - E_{Lk} \, ,$$

or when distributing exergy flows entering the k-th element E_k^{ent} and exiting from it E_k^{exit}, according to the qualitative characteristic of "fuel" and "product", as

$$F_k - P_k = E_{Dk} + E_{Lk} \, ,$$

where F_k is the is the exergy flow, which is the "fuel" for the element by its functional characteristic; P_k is the "product" of the element; E_{Dk} is the destruction of exergy in the element; E_{Lk} is the unused exergy flow, including the one that leaves the system through the dissipative element (condenser).

The exergies of the flows entering and leaving the element are determined as follows

$$E_i = D_i[(h_i - h_0) - T_0(s_i - s_0)] \, ,$$

where h_i, s_i is the enthalpy and entropy of the flow; D_i is the mass flow rate of the working substance of the cycle (air, flue gases); T_0, h_0, s_0 is the temperature in Kelvin, enthalpy and entropy of the substance at ambient temperature and pressure.

The results of calculations of the exergy characteristics of the thermal scheme of the STU are given in Table 4 and Fig. 3. The "fuel" of the turbine was defined as the exergy flow at the inlet to the HPP/LPP minus the exergy flow at the outlet, and the "product" – as the $N^{\text{HPP}}/N^{\text{LPP}}$. Unlike the turbine, when calculating the pumps, the electric power was considered as the "fuel", and the increase in exergy as the "product".

The "products" of the NH1, NH2 and the condenser are defined as

$$E_Q = Q\left[1 - \frac{T_0 + 273.15}{T_{hot} + 273.15}\right],$$

where T_{hot} is the network water temperature, K.

For the power boiler, the "fuel" is Q_{EB}, and the "product" is the exergy of superheated steam E_1 minus the exergy of feed water E_{14}.

When calculating the pressure loss at the steam inlet to the turbine, the turbine parts were taken into account in the "fuel".

The ambient temperature was taken equal to T_0=0 °C, P_0=101 kPa.

Table 4. Exergy parameters of the initial thermal scheme of the STU

Element	Fuel F, MW	Product P, MW	Destruction of exergy E_D, MW
EB	124.300	53.530	70.770
HPP 0–2	5.695	5.242	0.453
MPP 2–4	3.523	3.132	0.391
MPP 4–5	4.143	3.984	0.159
LPP 5–7	6.744	6.169	0.575
LPP 7–8	1.078	1.007	0.071
C	1.135	1.064	0.071
CP	0.000326	0.000287	0.000040
LPD	3.531	3.408	0.123
DP	0.09866	0.08386	0.0148
HPP	5.836	5.624	0.212
FP	0.1532	0.1303	0.0229
NH1	24.040	21.080	2.960
NH2	7.049	4.864	2.185
EG	19.770	19.370	0.400

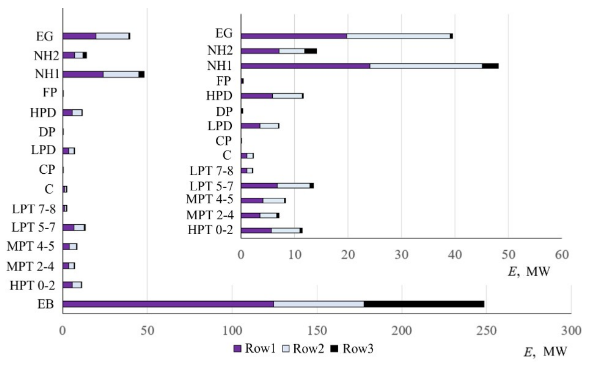

Fig. 3. Exergy parameters of the thermal scheme of the STU:
row 1 – "fuel"; row 2 – "product";
row 3 – "exergy destruction"

However, the destruction of exergy in the EB, as is known, depends on the imperfection of other elements of the STU. It can be reduced if the ways of improving the elements are correctly determined. Thus, it is clear that a large destruction of exergy is observed in NH1.

If we determine the exergy cost of the elements [12] according to the dependence $k_k = F_k / P_k$, then we can see (Fig. 4) that the first element that needs improvement is NH2.

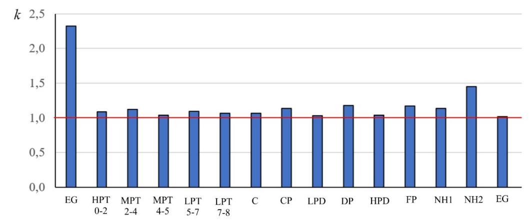

Fig. 4. Exergy cost of elements of the thermal scheme of a heating STU

According to Table 1, the pressure in the steam extractions can be changed within the specified ranges.

The thermal scheme was analyzed when the extractions pressure was reduced to a lower level, namely, the pressure in the first extraction was set to 980.7 kPa, and in the second one – to 245.2 kPa (first option). The energy characteristics of the thermal scheme of the STU with the new extractions parameters are shown in Table 5.

Table 5. Energy characteristics of the first option of the thermal scheme of the STU (P_4=980.7 kPa, G_4=8.33 kg/s, P_7=245.2 kPa, G_7=33.4 kg/s)

Power N, MW				Heat Q, MW			Electric power N_{STU}, MW	Cycle efficiency η_{net}, %	STU efficiency η_{net}^{STU}, %
HPP	MPP	LPP	CP+DP+FP	EB	NH1+NH2	C(HWS)	EG		
5.24	7.55	8.17	0.16	115.57	85.37	5.91	20.80	17.20	88.04

It can be seen that the electrical efficiency (net) for this option has increased by 1.7%, and η_{net}^{STU} does not change.

Comparative exergy analysis showed the following results.

The change in the exergy cost of each element of the STU scheme in percentage when varying the extractions parameters is shown if Fig. 5.

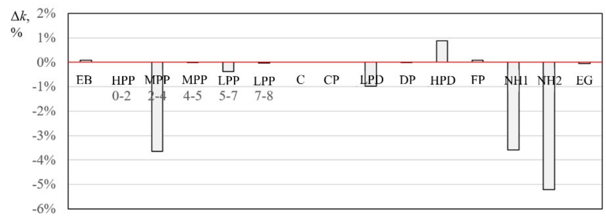

Fig. 5. Change in the exergy cost of each element of the STU scheme in % (comparison of the initial and first options)

From Fig. 5 it is seen that the cost of exergy of the flow in the STU before the first extraction decreases by almost 4%. The exergy cost of NH2 also decreases by more than 5%. This is a very good result, because the steam flow in this part of the turbine has high exergy values, i.e. the "quality" of energy is high. Thus, the thermal energy of the steam can be converted into other types of energy, and not only into thermal one. The decrease in the exergy cost shows a decrease in specific losses (energy dissipation) during energy conversion in the element. However, unlike the original scheme, in the option with reduced steam pressures in the extractions, an increase in the cost of exergy of the flow in the HPD is observed.

In addition, a decrease in the pressure in the second extraction also leads to a decrease in the cost of exergy of the flow in NH1.

However, when the steam pressure in the first extraction of the MPP is reduced, the exergy cost in NH2 still remains high (k=1.397). To reduce it, a calculation of the thermal scheme was carried out with a decrease in the steam consumption in the first extraction and an increase in the consumption in the second one while maintaining the thermal power of the STU.

The steam parameters at the cycle points of the second option of the STU thermal scheme with changed steam parameters in the first extraction and its consumption in two extractions are given in Table 6.

The energy characteristics of the last STU option are given in Table 7.

The energy analysis showed an increase in the electrical efficiency (net) by 1.31% compared to the original option, and the exergy analysis showed a decrease in the cost of the exergy flow in the EB (Fig. 6).

As a third option, a thermal scheme with the following steam parameters in the extractions was considered: P_4=980.7 kPa, G_4=5 kg/s, P_7=245.2 kPa, G_4=35.7 kg/s. This option is similar to the second one, but the steam pressure in the second extraction, just like in the second option, is lower than in the first and third ones, which allowed to further reduce the cost of the exergy flow in NH1 (Fig. 7).

Table 6. Steam consumption and its parameters at cycle points (Fig. 1) of the second option of the thermal scheme of the STU (P_4=980.7 kPa, G_4=5 kg/s, P_7=294 kPa, G_4=35.7 kg/s)

Steam/water flow name	Points	Parameters				
		Enthalpy i, kJ/K	Entropy s, kJ/(kg·K)	Temperature T, °C	Pressure P, kPa	Steam flow D, kg/s
Acute steam at the outlet of the EB	0	3233.0	6.9490	400.00	2844.00	44.4
Steam admission in the HPP	1	3233.0	6.9710	398.90	2702.00	44.4
Exhaust of the HPP	2	3115.0	6.9860	337.80	1729.00	44.4
Steam admission in the MPP	3	3115.0	7.0090	336.90	1643.00	44.4
Steam in the 1st extraction	4	2992.0	7.0260	272.60	980.70	5.0
Exhaust of the MPP	5	2934.0	7.0340	242.30	755.00	39.4
Steam admission in the MPP	6	2934.0	7.0570	241.60	717.30	39.4
Steam in the 2nd extraction	7	2763.0	7.0930	150.90	294.00	35.7
Exhaust of the LPP	8	2400.0	7.1860	64.95	25.00	3.7
Steam admission in the condenser	9	2400.0	7.2080	63.81	23.75	3.7
Condensate at the outlet of the condenser	10	267.1	0.8787	63.81	23.75	3.7
Feed water after the CP	11	267.2	0.8788	63.82	122.00	3.7
At the outlet of the LPD	12	531.2	1.6040	105.30	122.00	3.7
Feed water after the DP	13	531.6	1.5970	126.50	456.00	39.4
Feed water at the outlet of the HPD	14	630.2	1.8370	148.40	456.00	44.4
Feed water at the inlet of the EB	15	633.4	1.8380	149.90	2994.00	44.4
Saturated steam in the NH1	7"	2764.0	6.7000	165.90	717.30	35.7
Condensate after the NH1	7'	701.6	1.3450	165.90	717.30	35.7
Saturated steam in the NH2	4"	2794.0	6.4110	202.70	1643.00	5.0
Condensate after the NH2	4'	864.4	5.4750	202.70	1643.00	5.0

Table 7. Energy characteristics of the second option of the thermal scheme of the STU (P_4=980.7 kPa, G_4=5 kg/s, P_7=294 kPa, G_4=35.7 kg/s)

Power N, MW				Heat Q, MW			Electric power N_{STU}, MW	Cycle efficiency η_{net}, %	STU efficiency η_{net}^{STU}, %
HPP	MPP	LPP	CP+DP+FP	EB	NH1+NH2	C(HWS)	EG		
5.24	7.75	8.08	0.16	115.42	84.23	7.89	20.91	17.31	88.89

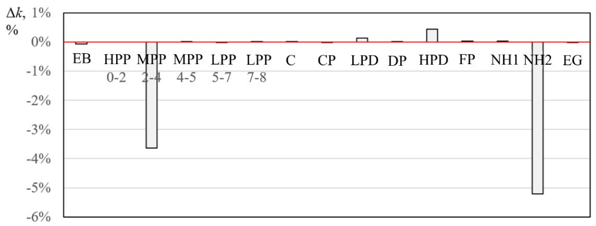

Fig. 6. Change in the exergy cost of each element of the STU scheme in % (comparison of the original and second options)

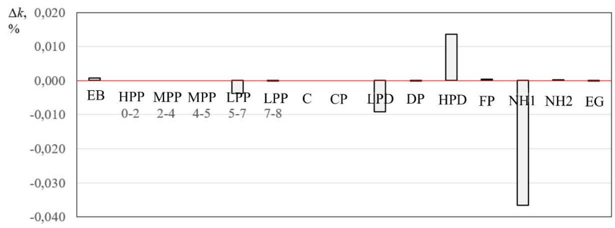

Fig. 7. Change in the exergy cost of each element of the STU scheme in % (comparison of the second and third options)

At the end of the analysis of the STU, the exergy efficiency of the unit was established by the formula

$$\eta_{ex} = \left(1 - \frac{E_{D\mathrm{sum}}}{Q_{\mathrm{EB}}}\right),$$

where $E_{D\mathrm{sum}}$– the exergy destruction of the unit, which is defined as the sum of the exergy destruction of all elements (Table 4).

The exergy efficiency of the unit for all options of the thermal scheme of the STU is shown in Fig. 8.

From Fig. 8 it is seen that the third option of the thermal scheme has the highest efficiency. This confirms the fact that the steam from the first extraction with high exergy is not effectively used for heating water. Its exergy potential should be converted into mechanical energy.

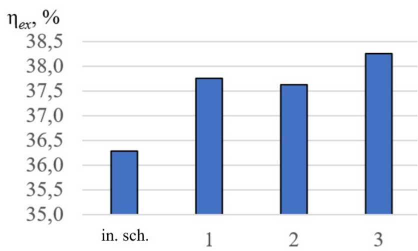

Fig. 8. Exergy efficiency of the unit:
in. sch. – initial scheme;
1,2,3 – options of the thermal scheme of the STU

Conclusions

1. The thermal scheme of a 20 MW STU was analyzed in terms of energy and exergy indicators, which allowed to identify elements with high exergy value. The highest exergy value is observed in the power boiler, but it can be reduced by reducing the destruction of exergy in other elements. Therefore, the network heater heated by high-pressure steam from the first extraction of the MPP was selected as the element with the greatest potential for increasing the efficiency of the STU. The second element was the first network heater heated by low-pressure steam from the second extraction of the LPP.

2. Options of the thermal scheme of the STU were considered, in which the steam parameters in the turbine extractions (pressure, flow rate) were varied. It was shown that with a decrease in pressure and a decrease in steam flow rate in the first extraction, as well as a decrease in steam pressure and an increase in steam flow rate in the second extraction, the cost of exergy flows in network heaters decreases by almost 5%, the exergy efficiency of the STU increases by 2%, and the electrical efficiency of the unit increases by 2.16% relative to the original scheme.

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Аналіз теплофікаційної паротурбінної установки ТЕЦ за енергетичними та ексергетичними показниками

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Проаналізовано теплофікаційно-конденсаційну паротурбінну установку (ПТУ) потужністю 20 МВт, що складається з частин високого, середнього й низького тисків й експлуатується на одній з ТЕЦ м. Харків. Виходячи зі схеми, пар із двох теплофікаційних регульованих відборів подається на дві мережеві підігрівальні установки. У ході реконструкції ТЕЦ підігрівачі системи регенерації ПТУ було демонтовано у зв'язку з їх спрацюванням. На ТЕЦ було ухвалено рішення про акцентування на збільшенні теплофікації, тому нові підігрівачі високого тиску не встановлені. Крім того, замість градирні було вирішено в охолоджувальному контурі конденсатору використовувати воду з мережі, яка йде на гаряче водопостачання. У статті проведено аналітичний огляд теплової схеми теплофікаційної ПТУ за енергетичними й ексергетичними показниками, що дозволило визначити елементи з високою ексергетичною вартістю, що є показником їх ефективності. Під час аналізу варіантів теплової схеми ПТУ складено аналітичні таблиці з ексергетичними параметрами вихідної схеми поелементно й аналітичні графіки. Відповідно до ексергетичного аналізу найбільша ексергетична вартість спостерігається в енергетичному котлі, але, як відомо, її можна знизити шляхом зменшення деструкції ексергії в інших елементах. З огляду на вказане як елемент із найбільшим потенціалом підвищення ефективності ПТУ обрано мережевий підігрівач, що обігрівається парою високого тиску з першого відбору частини середнього тиску. Відповідно, другим елементом обрано перший мережевий підігрівач, який обігрівається парою низького тиску з другого відбору частини низького тиску. Розглянуто варіанти теплової схеми ПТУ, в яких варіювалися параметри пари у відборах турбіни (тиск, витрата). Показано, що при зниженні тиску й зменшенні витрати пари у першому відборі, а також зниженні тиску пари й підвищенні витрати пари у другому відборі вартість ексергетичних потоків у мережевих підігрівачах знижується майже на 5 %, ексергетичний коефіцієнт корисної дії (ККД) ПТУ підвищується на 2%, а електричний ККД установки зростає на 2,16% відносно вихідної схеми.

Ключові слова*: ексергія, паротурбінна установка, ексергетична вартість, ефективність, ТЕЦ.*

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