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IMPROVEMENT OF THE THERMODYNAMIC CYCLE OF STEAM TURBINE INSTALLATIONS OF THERMAL POWER PLANTS

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Abstract

The production of electricity due to the heat of combustion of fuel is carried out mainly with the help of steam turbine plants (STP) that work according to the thermodynamic Rankine cycle. In the traditional Rankine cycle, steam formation and superheating of steam is carried out in an isobaric process, which loses to an isochoric process because part of the heat that is supplied isobarically is spent on premature expansion of the working body outside the turbine. It is proposed to improve the Rankine thermodynamic cycle by leaving the isobaric process of saturated steam formation, and also to carry out boiler and intermediate steam overheating in isochoric processes. As a basic thermodynamic cycle for comparison, the cycle of the MST-14 type ship steam turbine plant with a capacity of 16,200 kW was adopted. A comparison of the efficiency indicators of the proposed thermodynamic cycle and the cycle of the basic plant indicates the significant advantages of the proposed cycle, which allows increasing the power of the plant to 29,000 kW, other things being equal. The proposed improvement of the Rankine cycle will complicate the design of the steam turbine plant and increase the cost of their manufacture, but these costs will be compensate by a reduction in fuel consumption and carbon dioxide emissions.

Keywords: basic thermodynamic cycle of steam turbine plants; heat input; heat output; expansion work, thermal efficiency, modified plant.

УДОСКОНАЛЕННЯ ТЕРМОДИНАМІЧНОГО ЦИКЛУ ПАРОТУРБІННИХ УСТАНОВОК ТЕПЛОВИХ ЕЛЕКТРОСТАНЦІЙ

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Анотація

У наш час виробництво електроенергії за рахунок теплоти згоряння палива здійснюється головним чином за допомогою паротурбінних установок, які працюють за термодинамічним циклом Ренкіна. В традиційному циклі Ренкіна пароутворення і перегрів пари здійснюється в ізобарному процесі, який програє ізохорному процесу тому, що частина теплоти, яка підводиться ізобарно, витрачається на передчасне розширення робочого тіла поза турбіною. Пропонується вдосконалити термодинамічний цикл Ренкіна, залишивши ізобарний процес утворення насиченої пари, а котловий і проміжні перегріви пари здійснювати в ізохорному процесі. В якості базового термодинамічного циклу для порівняння прийнято цикл суднової паротурбінної установки типу MST-14 потужністю 16200 кВт. Порівняння показників ефективності запропонованого термодинамічного циклу і циклу базової установки вказує на суттєві переваги запропонованого циклу, який дозволяє збільшити потужність установки до 29000 кВт за інших рівних умов. Вдосконалення циклу Ренкіна, що пропонується, ускладнить конструкцію паротурбінної установки і збільшить собівартість їх виготовлення, але це витрати будуть компенсовані зменшенням витрати палива і викидів карбон (IV) оксиду.

Ключові слова: базовий термодинамічний цикл паротурбінних установок; підведена теплота; відведена теплота; робота розширення; термічний коефіцієнт корисної ді; модифікована установка.

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Introduction

Steam turbines played and play a big role in the civilizational development of Mankind. They came a long way from the steam engine of James Watt (1736–1819) [1, p. 14.] to steam turbines of modern nuclear power plants. Currently, steam turbine installations are used in the most energy-intensive devices for converting heat into mechanical/electrical energy, which are thermal power plants, thermal power plants, nuclear power plants, ship power plants of large tonnage ships, etc. More than 15 thermal power plants are currently operating in Ukraine [2]. Their total capacity is 25,647 MW.

Since the thermodynamic process of any power plant is the main factor in its efficiency, its improvement was, is and will always be relevant.

The first direction of the improvement of all energy cycles is the replacement of the isobaric process of supplying heat to the working body with an isochoric one. The unified equation of technical thermodynamics, which is written for a static process [3; 4], looks like

$$Tds = du - pdv.$$

It follows from this equation that in a static thermodynamic process (v=idem) the added heat is completely used to increase the internal energy of the working body, therefore less heat is spent in it for a more rapid increase in temperature, pressure and internal energy, which are the main indicators of the thermodynamic efficiency of the working body bodies On the other hand, in an isobaric process, part of the supplied heat is spent on the premature expansion of the working body, since its expansion takes place outside the device (for example, a turbine) to convert its potential energy into mechanical energy [5].

The second direction of improvement of the thermodynamic cycle of steam turbine plants is a decrease in temperature/pressure at the end of the expansion of the working body, which is equivalent to their increase at the beginning of the expansion. This is possible if the condenser is pre-cooled with regenerative/recycling cooled coolant [6].

The third direction of improvement of the thermodynamic cycles of steam turbine plants is the replacement of the static method of steam generation in the steam boiler by the hydrodynamic method in the steam generating device [7].

As indicated above, the main consumers of STP are currently TPPs, NPPs and ship power plants of large tonnage vessels. There is no Internet data on the features and parameters of the thermodynamic cycles of their STP. Therefore, as a prototype carried out in the work of the study, a marine steam turbine unit of the MST-14 type with a capacity of 18,000 to 40,000 hp was adopted, which was designed by the General Electric company in 1965 for use as the main power unit of powerful tankers, since for it has enough thermodynamic information [8].

Thus, the effective power of the STP-prototype is Ne = 22,000 hp, and the relative effective coefficient is η_{re} = 0.797, which is equal to the ratio of the effective power Ne to the power of the ideal turbine unit N_0 [8, p. 115]. Then the theoretical capacity of the STP-prototype is equal to

$$N_0 = \frac{N_e}{\eta_{re}} = \frac{22000}{0.797} = 27\ 603{,}5 \text{ hp} = \\ = 20\ 316{,}2 \text{ kW}.$$

The STP-prototype uses three turbines, the steam parameters at the entrance to which are given in table 1. The condensation pressure of the spent steam in the prototype cycle is equal to 0.05 ata. We used these data as inputs in all improvements of the thermodynamic cycle of the MST-14 type steam turbine plant proposed in the work.

Table 1

Steam parameters at the inlet to STP MST-14 turbines [8]

Turbine	Steam consumption, *M*		Pressure, *p*		Temperature, *t*
	kg/h	kg/s	ata	bar	°C
TVT	59 440	16.5111	103.0	101.043	510
TST	48 950	13.5972	20.5	20.11	510
TNT	46 120	12.2811	6.30	6.18	370

Results and Discussions

Thermodynamic cycle of the basic MST-14 steam turbine plant with isobaric processes of heat supply at a steam condensation pressure of 0.05 ata

In fig. 1 shows in *T,s*-coordinates the thermodynamic cycle of the basic STP of the MST - 14 type with two intermediate steam overheatings. External heat supply in all processes of this cycle is carried out isobarically.

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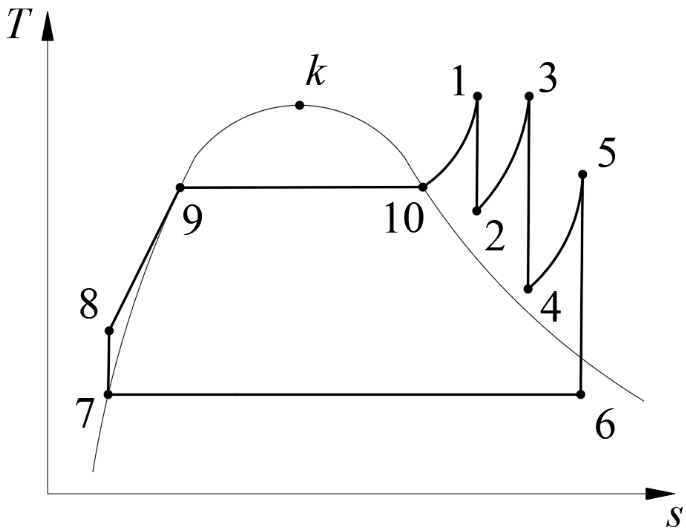

Fig. 1. Thermodynamic cycle of basic STP type MST-14 with isobaric processes of steam overheating

Determination of the thermodynamic properties of water and water vapor at the characteristic points of the thermodynamic cycle of the basic STP was carried out using the computer program REFPROP [9] with the input parameters specified above. The calculation was made according to the following algorithm in:

– point 1

t_1 = 510 °C, p_1 = 101.043 bar;

– point 2

p_2 = 20.11 bar, $s_2 = s_1$ = 6.6265 kJ/(kg·K);

– point 6

p_6 = 0.0515 bar, x_6 = 0.952,

$$s_6 = x_6(s'' - s') + s' = \\ = 0.952(8.3833 - 0.48339) + 0.48339 = \\ = 8.0041\ \text{kJ/(kg} \cdot \text{K)};$$

– point 5

t_5 = 370 °C, $s_5 = s_6$ = 8.0041 kJ/(kg·K);

– point 4

$p_4 = p_5$ = 2.6161 bar, $s_4 = s_3$ = 7.4594 kJ/(kg·K);

– point 7

$p_7 = p_6$ = 0.0515 bar, x_6 = 0;

– point 8

$p_8 = p_1$ = 101.043 bar,

$s_8 = s_7$ = 0.48339 kJ/(kg·K);

– points 9 and 10

$p_9 = p_{10} = p_1$ = 101.043 bar, x_9 = 0, x_{10} = 1.

The determined thermodynamic properties of water and water vapor at characteristic points of the thermodynamic cycle of the STP - the prototype of the proposed TPP improvements are given in the table 2.

Table 2

Thermodynamic properties of water and water vapor at characteristic points of the thermodynamic cycle of the basic MST-14 type STP with isobaric heat supply processes at p_{cond} = 0.0515 bar and t_{cond} = 33.401 °C

No	*t*, °C	*p*, bar	*v*, m³/kg	*u*, kJ/kg	*h*, kJ/kg	*s*, kJ/(kg·K)	Degree of dryness, x
1	510	101.043	0.003016	3066.0	3399.6	6.6265	-
2	267.49	20.11	0.11586	2713.5	2946.5	6.6265	-
3	510	20.11	0.17711	3134.0	3490.2	7.4594	-
4	219.01	2.6161	0.85816	2681.9	2906.4	7.4594	-
5	370	2.6161	1.1300	2918.3	3213.9	8.0041	-
6	33.401	0.0515	26.095	2311.0	2445.4	8.0041	0.952
7	33.401	0.0515	0.0010055	139.95	139.95	0.48339	0
8	33.652	101.043	0.0010011	139.97	150.08	0.48339	-
9	311.76	101.043	0.0014563	1397.8	1412.6	3.3681	0
10	311.76	101.043	0.017800	2543.7	2723.6	5.6095	1

Using tabular data, calculated:

– specific heat supplied in the cycle

$$q_1 = (h_1 - h_8) + (h_3 - h_2) + (h_5 - h_4) = \\ = (3399.6 - 150.08) + (3490.2 - 2946.5) \\ +(3213.9 - 2906.4) = 4100.72\ \text{kJ/kg};$$

– specific work of STP

$$l = (h_1 - h_2) + (h_3 - h_4) + \\ (h_5 - h_6) - |l_{pump}| = \\ = (3399.6 - 2946.5) + (3490.2 - 2906.4) + \\ +(3213.9 - 2445.4) - 10.13 = 1795.27\ \text{kJ/kg},$$

where the specific work of the pump is equal to

$$l_{pump} = h_8 - h_7 = 150.08 - 139.95 = \\ = 10.13\ \text{kJ/kg};$$

– thermal efficiency of the installation

$$\eta_t = \frac{l}{q_1} = \frac{1795.27}{4100.72} = 0.4378;$$

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– specific fuel consumption per 1 kWh of plant operation

$$b_0 = \frac{3600}{Q_p^N \cdot \eta_t} = \frac{3600}{40 \cdot 10^3 \cdot 0.4378} = = 0.2056 \text{ kg/(kW} \cdot \text{h)};$$

– theoretical capacities of MST-14 type STP turbines:

- high pressure turbines

$$N_{T,\mathrm{HPT}} = M_{\mathrm{HPT}}(h_1 - h_2) = = 16.5111(3399.6 - 2946.5) = = 7\,481.18 \text{ kW};$$

- medium pressure turbines

$$N_{T,\mathrm{MPT}} = M_{\mathrm{MPT}}(h_3 - h_4) = = 13.5972(3490.2 - 2906.4) = = 7\,938.05 \text{ kW};$$

- low pressure turbines

$$N_{T,\mathrm{LPT}} = M_{\mathrm{LPT}}(h_5 - h_6) = = 12.2811(3213.9 - 2445.4) = = 9\,438.03 \text{ kW};$$

– power of the feed pump

$$N_{\mathrm{pump}} = M_{\mathrm{LPT}}(h_8 - h_7) = = 12.2811(150.08 - 139.95) = = 124.41 \text{ kW};$$

– the theoretical power of STP

$$\sum N_{\mathrm{STP}} = N_{T,\mathrm{HPT}} + N_{T,\mathrm{MPT}} + N_{T,\mathrm{LPT}} - N_{pump} = = 7481.18 + 7938.05 + 9438.03 - 124.41 = = 24\,732.85 \text{ kW}.$$

Thermodynamic cycle of modified stp type MST-14 with isobaric heat supply processes at steam condensation temperature 20 °C

In the basic prototype cycle, the steam condensation pressure of 0.05 ata (0.0515 bar), which corresponds to the condensation temperature of 33.401 °C. In accordance with the above-mentioned second direction of improvement of the thermodynamic cycle of STP, it is proposed to reduce the steam condensation temperature to 20 °C, which is possible if the refrigerant that cools the condenser is pre-cooled. Therefore, it is proposed to cool the coolant using a utilization heat-using combined energy-refrigerating unit (UHUCERU), proposed and studied in [10]. With its help, it is possible to obtain cold of a negative temperature at the expense of secondary heat sources, and STP are very rich in them. For example, exhaust gases of a steam boiler can be used as a hot source of heat for the operation of UHUCERU.

Since the proposed cycle differs from the basic one only by the temperature/pressure of the condensation of spent steam, the thermodynamic properties of water and steam at all characteristic points of this improved thermodynamic cycle were calculated using the above algorithm [11–14].

The determined thermodynamic properties are summarized in the table 3.

Table 3

Thermodynamic properties of water and water vapor at the characteristic points of the modified STP type MST-14 with isobaric heat supply processes at t_{cond} = 20 °C and p_{cond} = 0.023393 bar

No	t, °C	p, bar	v, m³/kg	u, kJ/kg	h, kJ/kg	s, kJ/(kg·K)	Degree of dryness, x
1	510	101.043	0.003016	3066.0	3399.6	6.6265	-
2	267.49	20.11	0.11586	2713.5	2946.5	6.6265	0.986
3	510	20.11	0.17711	3134.0	3490.2	7.4594	-
4	157.89	1.496	1.3144	2592.2	2788.9	7.4594	-
5	370	1.496	1.9795	2919.7	3215.9	8.2643	-
6	20	0.023393	54.985	2291.1	2419.7	8.2643	0.952
7	20	0.023393	0.0010018	83.912	83.914	0.29648	0
8	20.152	101.043	0.0009973	83.934	94.011	0.29648	-
9	311.76	101.043	0.0014563	1397.8	1412.6	3.3681	0
10	311.76	101.043	0.017800	2543.7	2723.6	5.6095	1

Using tabular data, calculated:

– specific heat supplied to the cycle

$$q_1 = (h_1 - h_8) + (h_3 - h_2) + (h_5 - h_4) = = (3399.6 - 94.011) + (3490.2 - 2946.5) + +(3215.9 - 2788.9) = 4276.29 \text{ kJ/kg};$$

– specific work of STP

$$l = (h_1 - h_2) + (h_3 - h_4) + (h_5 - h_6) - -|l_{pump}| = (3399.6 - 2946.5) + +(3490.2 - 2788.9) + +(3215.9 - 2419.7) - 10.10 = 1940.5 \text{ kJ/kg},$$

where the pump work is equal to

$$|l_{pump}| = h_8 - h_7 = 94.011 - 83.914 = = 10.10 \text{ kJ/kg};$$

– thermal efficiency of the installation

$$\eta_t = \frac{l}{q_1} = \frac{1940.50}{4276.29} = 0.4538;$$

– specific fuel consumption per 1 kWh of plant operation

$$b_0 = \frac{3600}{Q_p^N \cdot \eta_t} = \frac{3600}{40 \cdot 10^3 \cdot 0.4538} = = 0.1983 \text{ kg/(kW} \cdot \text{h)};$$

– power turbines of the modified STP type

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MST-14

- high pressure turbines

$$N_{T,\mathrm{HPT}} = M_{\mathrm{HPT}}(h_1 - h_2) =$$
$$= 16.5111(3399.6 - 2946.5) = 7481.18 \text{ kW};$$

- medium pressure turbines

$$N_{T,\mathrm{MPT}} = M_{\mathrm{MPT}}(h_3 - h_4) =$$
$$= 13.5972(3490.2 - 2788.9) = 9535.72 \text{ kW};$$

- low pressure turbines

$$N_{T,\mathrm{LPT}} = M_{\mathrm{LPT}}(h_5 - h_6) =$$
$$= 12.2811(3215.9 - 2419.7) = 9778.21 \text{ kW};$$

– power of the feed pump

$$N_{\mathrm{pump}} = M_{\mathrm{LPT}}(h_8 - h_7) =$$
$$= 12.2811(94.011 - 83.914) = 124.00 \text{ kW};$$

– the theoretical power of a modified STP type MST-14

$$\sum N_{\mathrm{STP}} = N_{T,\mathrm{HPT}} + N_{T,\mathrm{MPT}} + N_{T,\mathrm{LPT}} - N_{pump} =$$
$$= 7481.18 + 9535.72 + 9778.2 - 124.00 =$$
$$= 26671.1 \text{ kW}.$$

Thermodynamic cycle of modified stp type MST-14 with isobaric heat supply processes at steam condensation temperature 10 °C

Since this thermodynamic cycle differs from the ones discussed above only by the condensation temperature of the steam, which is equal to 10 °C, then using the above algorithm for calculating the thermodynamic properties of the working body and the REFPROP program [9], the thermodynamic properties of water and water vapor at its characteristic points are determined [15–18].

The values of the thermodynamic properties of the working body changed only in points 6, 5, 4, 7 and 8, which were determined according to the following algorithm:

– point 6

t_6 = 10 °C, x_6 = 0.952,

– point 5

t_5 = 370 °C, $s_5 = s_6$ = 8.4798 kJ/(kg·K);

– point 4

$p_4 = p_5$ =0.94017 bar, $s_4 = s_3$ = 7.4594 kJ/(kg·K);

– point 7

t_7 = 10 °C, x_7 = 0;

– point 8

p_8 = 101.043 bar, $s_8 = s_7$ = 0.15109 kJ/(kg·K).

In the table 4 gives detailed data on the thermodynamic properties of the working fluid at the characteristic points of this cycle.

Table 4

Thermodynamic properties of water and water vapor at the characteristic points of the modified STP type MST-14 with isobaric heat supply processes at t_{cond} = 10 °C, p_{cond} = 0.023393 bar.

No	t, °C	p, bar	v, m³/kg	u, kJ/kg	h, kJ/kg	s, kJ/(kg·K)	Degree of dryness, x
1	510	101.043	0.003016	3066.0	3399.6	6.6265	-
2	267.49	20.11	0.11586	2713.5	2946.5	6.6265	0.986
3	510	20.11	0.17711	3134.0	3490.2	7.4594	-
4	112.65	0.94017	1.8710	2520.0	2702.4	7.4594	-
5	370	0.94017	3.1526	2920.5	3216.8	8.4798	-
6	10	0.012282	101.20	2276.0	2400.3	8.4798	0.952
7	10	0.02282	0.0010003	42.020	42.021	0.15109	0
8	10.069	101.043	0.00099558	42.045	52.143	0.15109	-
9	311.76	101.043	0.0014563	1397.8	1412.6	3.3681	0
10	311.76	101.043	0.017800	2543.7	2723.6	5.6095	1

Using the obtained data, the following was calculated:

– specific heat supplied to the cycle

$$q_1 = (h_1 - h_8) + (h_3 - h_2) + (h_5 - h_4) =$$
$$= (3399.6 - 52.143) + (3490.2 - 2946.5) +$$
$$+(3216.8 - 2702.4) = 4405.56 \text{ kJ/kg};$$

– specific work of STP

$$l = (h_1 - h_2) + (h_3 - h_4) + (h_5 - h_6) -$$
$$-|l_{pump}| = (3399.6 - 2946.5) +$$
$$+(3490.2 - 2702.4) +$$
$$+(3216.8 - 2400.3) - 10.12 = 2047.28 \text{ kJ/kg},$$

where the pump work is equal to

$$|l_{pump}| = h_8 - h_7 = 52.143 - 42.021 =$$
$$= 10.12 \text{ kJ/kg};$$

– thermal efficiency of the installation

$$\eta_t = \frac{l}{q_1} = \frac{2047.28}{4405.56} = 0.4647;$$

–specific fuel consumption per 1 kWh of plant operation

$$b_0 = \frac{3600}{Q_p^N \cdot \eta_t} = \frac{3600}{40 \cdot 10^3 \cdot 0.4647} =$$
$$= 0.1937 \text{ kg/(kW} \cdot \text{h)};$$

– power turbines of the modified STP type MST-14

- high pressure turbines

$$N_{T,\mathrm{HPT}} = M_{\mathrm{HPT}}(h_1 - h_2) =$$
$$= 16.5111(3399.6 - 2702.4) = 11\,511.54 \text{ kW};$$

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- medium pressure turbines

$$N_{T,\mathrm{MPT}} = M_{\mathrm{MPT}}(h_3 - h_4) =$$
$$= 13.5972(3490.2 - 2702.4) = 10\ 711.87\ \mathrm{kW};$$

- low pressure turbines

$$N_{T,\mathrm{LPT}} = M_{\mathrm{LPT}}(h_5 - h_6) =$$
$$= 12.2811(3216.8 - 2400.3) = 10\ 027.52\ \mathrm{kW};$$

–power of the feed pump

$$N_{pump} = M_{\mathrm{LPT}}(h_8 - h_7) =$$
$$= 12.2811(52.143 - 42.021) = 124.31\ \mathrm{kW};$$

–the theoretical power of a modified STP type MST-14

$$\sum N_{\mathrm{STP}} = N_{T,\mathrm{HPT}} + N_{T,\mathrm{MPT}} + N_{T,\mathrm{LPT}} - N_{pump} =$$
$$= 11511.54 + 10711.87 + 10027.52 - 124.31 =$$
$$= 32126.62\ \mathrm{kW}.$$

Thermodynamic cycle of a modified MST-14 steam turbine plant with isochorous steam superheating processes at a steam condensation pressure of 0.05 ata

In fig. 2 shows the thermodynamic cycle of the MST-14 type steam turbine plant modified by replacing isobaric processes of steam overheating 10–1, 2–3 and 4–5 (Fig. 1) with isochoric processes 10–1, 2–3, 4–5 and 5′–5″.

All thermodynamic cycles of the compared STPs were built at the same temperature of the steam supplied to the first stage of the HPT (t_1 = 510 °C) and at the same dryness of the steam at the last stages of the LPT (x_6 = 0.952). This condition of comparing thermodynamic cycles led to the appearance of an additional turbine in the STP with isochoric processes of steam overheating [19;20]. We designed this additional stage as the second low-pressure turbine - LPT_2, the steam expansion process in which is depicted by the line 5–5′ (Fig. 2).

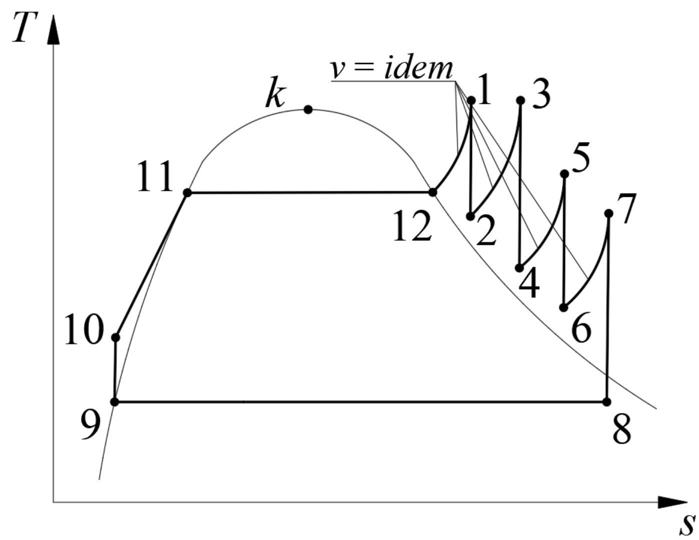

Fig. 2. The thermodynamic cycle of the modified STP type MST-14 with isochoric steam superheating processes

Determination of thermodynamic properties of water and water vapor at characteristic points of this cycle was carried out with the following input parameters:

– points 9 and 10

$p_9 = p_{10} = p_1$ = 101.043 bar, $x_9 = 0$, $x_{10} = 1$;

–point 1

t_1 = 510 °C, $v_1 = v_{10}$ =0.017800 m^3/kg;

– point 2

p_2 = 20.11 bar, $s_2 = s_1$ = 6.2820 kJ/(kg·K);

– point 3

$t_3 = t_1$ =510 °C; $v_3 = v_2$ = 0.097668 m^3/kg;

– point 4

t_4 = 219.01 °C (see Table 3);

$s_4 = s_3$ = 7.1740 kJ/(kg·K);

– point 6

p_6 = 0.0515 bar, x_6 = 0.952,

$$s_6 = x_6(s'' - s') + s' =$$
$$= 0.952(8.3833 - 0.48339) + 0.48339 =$$
$$= 8.0041\ \mathrm{kJ/(kg \cdot K)};$$

– to determine the enthalpy at the point 5″, the auxiliary point X was calculated with the following parameters

– entropy at an auxiliary point

$s_x = s_6$ = 8.0041 kJ/(kg·K); $\rho_x = \rho_4$ = 2.1197 kg/m^3.

The enthalpy h_x at point x was determined using the REFPROP program [9]:

h_x = 3589.4 kJ/kg; the heat transfer in the imaginary LPT is equal to $\Delta h = h_x - h_6$ = 3589.4 – 2445.4 = 1144.0 kJ/kg. This heat transfer is

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divided into two LPT bodies. Then the enthalpy of steam at the entrance to LPT_2 is equal to

$$h_{5''} = h_6 + \Delta/2 = 2445.4 + 1144.0/2 = 3\,017.4\ kJ/kg,$$

and the value of entropy:

$s_{5''} = s_6 = 8.0041$ kJ/(kg·K);

– point 5

$t_5 = 370$ °C, $v_5 = v_4 = 0.47177$ m³/kg;

– point 5′

$s_{5'} = s_5 = 7.5962$ kJ/(kg·K);

$\rho_{5'} = \rho_{5''} = 1 / v_{5''} = 1/1.9696 = 0.50771$ kg/m³;

– point 7

$p_7 = p_6 = 0.0515$ bar, $x_7 = 0$;

– point 8

$p_8 = p_1 = 101.043$ bar;

$s_8 = s_7 = 0.48339$ kJ/(kg·K).

The defined thermodynamic properties of the working fluid are given in the table 5.

Table 5

Thermodynamic parameters of water and water vapor at the characteristic points of the modified STP type MST-14 with isochoric processes of steam overheating at a condensation pressure of 0.05 ata

No	*t*, °C	*p*, bar	*v*, m³/kg	*u*, kJ/kg	*h*, kJ/kg	*s*, kJ/(kg·K)	Degree of dryness, x
1	510	174.88	0.017800	2996.0	3307.3	6.2820	-
2	212.65	20.11	0.097668	2575.3	2771.7	6.2820	0.986
3	510	36.029	0.097668	3121.3	3473.2	7.1740	-
4	219.01	4.7123	0.47177	2675.2	2897.5	7.1740	-
5	370	6.2308	0.47177	2913.5	3207.5	7.5962	-
5′	142.57	0.96564	1.9696	2571.9	2762.1	7.5962	-
5″	271.90	1.2724	1.9696	2766.8	3017.4	8.0041	-
6	33.401	0.0515	26.095	2311.0	2445.4	8.0041	0.952
7	33.401	0.0515	0.0010055	139.95	139.95	0.48339	0
8	33.652	101.043	0.0010011	139.97	150.08	0.48339	-
9	311.76	101.043	0.0014563	1397.8	1412.6	3.3681	0
10	311.76	101.043	0.017800	2543.7	2723.6	5.6095	1

Using tabular data, calculated:

– specific heat supplied in the cycle

$$q_1 = q_{8-1} + q_{2-3} + q_{4-5} + q_{5'-5''} = = 3025.82 + 546.0 + 238.3 + 194.9 = = 4005.02\ \text{kJ/kg},$$

where

$$q_{8-1} = (h_{10} - h_8) + (u_1 - u_{10}) = = (2723.6 - 150.08) + (2996.0 - 2543.7) = = 3025.82\ \text{kJ/kg};$$

$$q_{2-3} = u_3 - u_2 = 3121.3 - 2575.3 = = 54.0\ \text{kJ/kg};$$

$$q_{4-5} = u_5 - u_4 = 2913.5 - 2675.2 = = 238.3\ \text{kJ/kg};$$

$$q_{5'-5''} = u_{5''} - u_{5'} = 2766.8 - 2571.9 = = 194.9\ \text{kJ/kg};$$

– specific work of STP

$$l = (h_1 - h_2) + (h_3 - h_4) + (h_5 - h_{5'}) + +(h_{5''} - h_6) - |l_{pump}| = (3307.3 - 2771.7) + +(3473.2 - 2897.5) + (3207.5 - 2762.1) + +(3017.4 - 2445.4) - 10.13 = 2128.57\ \text{kJ/kg},$$

where the specific work of the pump is equal to

$$l_{pump} = h_8 - h_7 = 150.08 - 139.95 = = 10.13\ \text{kJ/kg};$$

– thermal efficiency of the installation

$$\eta_t = \frac{l}{q_1} = \frac{2118.57}{4005.02} = 0.5290;$$

– specific fuel consumption per 1 kWh of plant operation

$$b_0 = \frac{3600}{Q_p^N \cdot \eta_t} = \frac{3600}{40 \cdot 10^3 \cdot 0.5290} = = 0.1701\ \text{kg/(kW} \cdot \text{h)};$$

– theoretical power of the turbines of the modified STP of the MST-14 type:

- high pressure turbines

$$N_{T,\text{HPT}} = M_{\text{HPT}}(h_1 - h_2) = = 16.5111(3307.3 - 2771.7) = 8\,843.35\ \text{kW};$$

- medium pressure turbines

$$N_{T,\text{MPT}} = M_{\text{MPT}}(h_3 - h_4) = = 13.5972(3473.2 - 2897.5) = 7\,827.91\ \text{kW};$$

- low pressure turbine

$$N_{T,\text{LPT1}} = M_{\text{LPT1}}(h_5 - h_{5'}) = = 12.2811(3207.5 - 2762.1) = 5\,470.00\ \text{kW};$$

$$N_{T,\text{LPT2}} = M_{\text{LPT1}}(h_{5''} - h_6) = = 12.2811(3017.4 - 2445.4) = 7\,024.79\ \text{kW};$$

- STP feed pump

$$N_{pump} = M_{\text{LPT}}(h_8 - h_7) = = 12.2811(150.08 - 139.95) = 124.41\ \text{kW};$$

– the theoretical power of a modified STP type MST-14

$$\sum N_{\text{STP}} = N_{T,\text{HPT}} + N_{T,\text{MPT}} + N_{T,\text{LPT1}} + +N_{T,\text{LPT2}} - N_{pump} = 8843.35 + 7827.91 + +5470.00 + 7024.79 - 124.41 = 29\,041.64\ \text{kW}.$$

Thermodynamic cycle of modified stp type MST-14 with isohorous steam superheating processes at steam condensation temperature 20 °C

Such an improvement of the thermodynamic cycle of the STP type MST-14 differs from the previous one only by a lower temperature/pressure of steam condensation, which are equal to t_{cond} = 20 °C, p_{cond} = 0.02339 bar.

Using the program REFPROP [9] and the above algorithm, the thermodynamic properties of water and water vapor at all characteristic points of this thermodynamic cycle were calculated.

The determined thermodynamic properties are summarized in the table 6.

Table 6

Thermodynamic properties of water and water vapor at characteristic points of the modified STP type MST-14 with isochoric processes of steam overheating at t_{cond} = 20 °C, p_{cond} = 0.02339 bar.

No	t, °C	p, bar	v, m³/kg	u, kJ/kg	h, kJ/kg	s, kJ/(kg·K)	Degree of dryness, x
1	510	174.88	0.017800	2996.0	3307.3	6.2820	-
2	212.65	20.11	0.097668	2575.3	2771.7	6.2820	0.986
3	510	36.029	0.097668	3121.3	3473.2	7.1740	-
4	219.01	4.7123	0.47177	2675.2	2897.5	7.1740	-
5	370	6.2308	0.47177	2913.5	3207.5	7.5962	-
5′	119.63	0.76204	2.3576	2538.7	2718.3	7.5962	-
5″	337.80	1.1933	2.3576	2869.3	3150.6	8.2643	-
6	20	0.023393	54.985	2291.1	2419.7	8.2643	0.952
7	20	0.023393	0.0010018	83.912	83.914	0.29648	0
8	20.152	101.043	0.0009973	83.934	94.011	0.29648	-
9	311.76	101.043	0.0014563	1397.8	1412.6	3.3681	0
10	311.76	101.043	0.017800	2543.7	2723.6	5.6095	1

Using the obtained data, the following was calculated:

– specific heat supplied in the cycle

$$q_1 = q_{8-1} + q_{2-3} + q_{4-5} + q_{5'-5''} = \\ = 3081.89 + 546.0 + 238.3 + 330.6 = \\ = 4196.79 \text{ kJ/kg},$$

where

$$q_{8-1} = (h_{10} - h_8) + (u_1 - u_{10}) = \\ = (2723.6 - 94.011) + (2996.0 - 2543.7) = \\ = 3\,081.89 \text{ kJ/kg},$$

$$q_{2-3} = u_3 - u_2 = 3121.3 - 2575.3 = \\ = 546.0 \text{ kJ/kg},$$

$$q_{4-5} = u_5 - u_4 = 2913.5 - 2675.2 = \\ = 238.3 \text{ kJ/kg},$$

$$q_{5'-5''} = u_{5''} - u_{5'} = 2869.3 - 2538.7 = \\ = 330.6 \text{ kJ/kg};$$

– specific work of STP

$$l = (h_1 - h_2) + (h_3 - h_4) + (h_5 - h_{5'}) + \\ +(h_{5''} - h_6) - |l_{pump}| = \\ = (3307.3 - 2771.7) + (3473.2 - 2897.5) + \\ +(3207.5 - 2718.3) + (3150.6 - 2419.7) - \\ -10.10 = 2\,321.30 \text{ kJ/kg},$$

where the pump work is equal to

$$|l_{pump}| = h_8 - h_7 = 94.017 - 83.914 = \\ = 10.10 \text{ kJ/kg};$$

– thermal efficiency of the installation

$$\eta_t = \frac{l}{q_1} = \frac{2321.30}{4196.79} = 0.5531;$$

– specific fuel consumption per 1kWh of plant operation

$$b_0 = \frac{3600}{Q_p^N \cdot \eta_t} = \frac{3600}{40 \cdot 10^3 \cdot 0.5531} = \\ = 0.1627 \text{ kg/(kW} \cdot \text{h)};$$

– capacity of turbines of STP MST-14 installation

- high pressure turbines

$$N_{T,\text{HPT}} = M_{\text{HPT}}(h_1 - h_2) = \\ = 16.5111(3307.3 - 2771.7) = 8\,843.35 \text{ kW};$$

- medium pressure turbines

$$N_{T,\text{MPT}} = M_{\text{MPT}}(h_3 - h_4) = \\ = 13.5972(3473.2 - 2897.5) = 7\,827.91 \text{ kW};$$

- low pressure turbines

$$N_{T,\text{LPT1}} = M_{\text{LPT1}}(h_5 - h_{5'}) = \\ = 12.2811(3207.5 - 2718.3) = 6\,007.91 \text{ kW};$$

$$N_{T,\text{LPT2}} = M_{\text{LPT1}}(h_{5''} - h_6) = \\ = 12.2811(3150.6 - 2419.7) = 8\,976.26 \text{ kW};$$

$$N_{pump} = M_{\text{LPT}}(h_8 - h_7) = \\ = 12.2811(94.011 - 83.914) = 124.00 \text{ kW};$$

– the theoretical power of STP type MST-14 installation

$$\sum N_{\text{STP}} = N_{T,\text{HPT}} + N_{T,\text{MPT}} + N_{T,\text{LPT1}} + \\ +N_{T,\text{LPT2}} - N_{pump} = 8\,843.35 + 7\,827.91 + \\ +6\,007.91 + 8\,976.26 - 124.00 = \\ = 31\,531.43 \text{ kW}.$$

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Thermodynamic cycle of modified stp type MST-14 with isohorous steam superheating processes at steam condensation temperature 10 °C

This modification of the thermodynamic cycle of STP is similar to the two considered above and differs from them only by an even lower temperature/pressure of condensation of spent steam (t_{cond} = 10 °C, p_{cond} = 0.012282 bar).

Using the program REFPROP [9] and the above algorithm, the thermodynamic properties of water and water vapor at all characteristic points of this thermodynamic cycle were calculated.

The determined thermodynamic properties are summarized in the table 7.

Table 7

Thermodynamic properties of water and water vapor at the characteristic points of the modified STP type MST-14 with isochoric processes of steam overheating at t_{cond} = 10 °C, p_{cond} = 0.012282 bar.

No	t, °C	p, bar	v, m³/kg	u, kJ/kg	h, kJ/kg	s, kJ/(kg·K)	Degree of dryness, x
1	510	174.88	0.017800	2996.0	3307.3	6.2820	-
2	212.65	20.11	0.097668	2575.3	2771.7	6.2820	0.986
3	510	36.029	0.097668	3121.3	3473.2	7.1740	-
4	219.01	4.7123	0.47177	2675.2	2897.5	7.1740	-
5	370	6.2308	0.47177	2913.5	3207.5	7.5962	-
5′	103.28	0.63833	2.6967	2515.0	2687.2	7.5962	-
5″	399.81	1.15	2.6967	2967.8	3277.9	8.4798	-
6	10	0.012282	101.20	2276.0	2400.3	8.4798	0.952
7	10	0.012282	0.0010003	42.020	42.021	0.15109	0
8	10.069	101.043	0.00099558	42.045	52.143	0.15109	-
9	311.76	101.043	0.0014563	1397.8	1412.6	3.3681	0
10	311.76	101.043	0.017800	2543.7	2723.6	5.6095	1

Using tabular data, calculated:

– specific heat supplied in the cycle

$$q_1 = q_{8-1} + q_{2-3} + q_{4-5} + q_{5'-5''} = \\ = 3123.76 + 546.0 + 238.3 + 452.96 = \\ = 4361.02 \text{ kJ/kg},$$

where

$$q_{8-1} = (h_{10} - h_8) + (u_1 - u_{10}) = \\ = (2723.6 - 52.143) + (2996.0 - 2543.7) = \\ = 3\,123.76 \text{ kJ/kg},$$

$$q_{2-3} = u_3 - u_2 = 3121.3 - 2575.3 = \\ = 546.0 \text{ kJ/kg},$$

$$q_{4-5} = u_5 - u_4 = 2913.5 - 2675.2 = \\ = 238.3 \text{ kJ/kg},$$

$$q_{5'-5''} = u_{5''} - u_{5'} = 2967.9 - 2515.0 = \\ = 452.96 \text{ kJ/kg};$$

– specific work of STP

$$l = (h_1 - h_2) + (h_3 - h_4) + (h_5 - h_{5'}) + \\ +(h_{5''} - h_6) - |l_{pump}| = \\ = (3307.3 - 2771.7) + (3473.2 - 2897.5) + \\ +(3207.5 - 2687.2) + (3277.9 - 2400.3) - \\ -10.12 = 2\,499.08 \text{ kJ/kg},$$

where the pump work is equal to

$$|l_{pump}| = h_8 - h_7 = 52.143 - 42.021 = \\ = 10.12 \text{ kJ/kg};$$

– thermal efficiency of the installation

$$\eta_t = \frac{l}{q_1} = \frac{2499.08}{4361.02} = 0.5730;$$

– specific fuel consumption per 1kWh of plant operation

$$b_0 = \frac{3600}{Q_p^N \cdot \eta_t} = \frac{3600}{40 \cdot 10^3 \cdot 0.5730} = \\ = 0.1571 \text{ kg/(kW} \cdot \text{h)};$$

– power turbines of the modified STP type MST-14

- high pressure turbines

$$N_{T,\text{HPT}} = M_{\text{HPT}}(h_1 - h_2) = \\ = 16.5111(3307.3 - 2771.7) = 8\,843.35 \text{ kW};$$

- medium pressure turbines

$$N_{T,\text{MPT}} = M_{\text{MPT}}(h_3 - h_4) = \\ = 13.5972(3473.2 - 2897.5) = 7\,827.91 \text{ kW};$$

- low pressure turbines

$$N_{T,\text{LPT1}} = M_{\text{LPT1}}(h_5 - h_{5'}) = \\ = 12.2811(3207.5 - 2687.2) = 6\,389.86 \text{ kW};$$

$$N_{T,\text{LPT2}} = M_{\text{LPT1}}(h_{5''} - h_6) = \\ = 12.2811(3277.9 - 2400.3) = 10\,777.89 \text{ kW};$$

$$N_{pump} = M_{\text{LPT}}(h_8 - h_7) = \\ = 12.2811(52.143 - 42.021) = 124.31 \text{ kW};$$

– the theoretical power of STP type MST-14 installation

$$\sum N_{\text{STP}} = N_{T,\text{HPT}} + N_{T,\text{MPT}} + N_{T,\text{LPT1}} + \\ +N_{T,\text{LPT2}} - N_{pump} = 8\,843.35 + 7\,827.91 + \\ +6\,389.86 + 10\,777.89 - 124531 = \\ = 33\,714.70 \text{ kW}.$$

For the possibility of comparing the efficiency of the basic and proposed modified STPs of the MST- 14 type, the most important indicators of their efficiency are summarized in the table 8.

Table 8

Comparison of the most important characteristics of the basic thermodynamic cycle of the MST-14 type STP with modified thermodynamic cycles

No	Performance indicators	Thermodynamic cycles with isobaric heat transfer processes			Thermodynamic cycles with isochoric processes of steam overheating		
		Base, t_{cond}= 33 °C	Modified		Modified		
			t_{cond}= 20 °C	t_{cond}= 10 °C	t_{cond}= 33 °C	t_{cond}= 20 °C	t_{cond}= 10 °C
1	Specific heat supplied in the cycle, q, kJ/kg	4 100.72	4 276.29	4 405.56	4 005.02	4 196.79	4 361.02
2	Specific work of vocational training, l, kJ/kg	1 795.27	1 940.50	2 047.28	2 128.57	2 321.30	2 499.08
3	Thermal efficiency, η_t	0.4378	0.4538	0.4647	0.5290	0.5531	0.6573
4	Theoretical power of vocational training, N_{STP}, kW	24 732.9	26 671.1	32 126.62	29 041.64	31 531.43	33 714.70
5	Specific fuel consumption per 1 kWh of plant operation, b_0	0.2056	0.1983	0.1937	0.1701	0.1627	0.1571
6	Relative increase in efficiency, %	-	3.70	6.1	20.8	26.3	50.1
7	Relative reduction in specific fuel consumption, %	-	3.70	6.1	17.3	26.4	30.9
8	Relative change in specific heat supplied in the STP cycle, %	-	+4.3	+7.4	-2.3	+2.3	+6.3
9	Relative increase in theoretical power, %	-	7.8	29.9	17.4	27.5	36.3

Conclusions

Analyzing the data of the comparative table 8, it can be unequivocally stated that all five improvements of the thermodynamic cycle of STP proposed in the paper are effective. Thus, the deviations of the thermal efficiency of the improved cycles, calculated relative to the basic cycle of the STP type MST-14, range from 3.7 to 50 %. From a constructive point of view, the simplest improvement is an improvement with a preliminary disposal cooling of the refrigerant that cools the condenser. The existing Ukrainian TPPs, the total capacity of which is 25 647 MW, should be equipped with utilization heat-using combined energy refrigerating units, with the help of which to cool the coolant that cools the condenser, in order to save 13 719 159 tons of fuel, the heat of combustion of which is $40{\cdot}10^3$ kJ/kg, every year. If transferred to conventional fuel ($Q_н$ = $29.33{\cdot}10^3$ kJ/kg), the savings of conventional fuel will amount to 18 710 070 t/year.

A more structurally complex, but also more effective improvement of the thermodynamic cycle of STP is an improvement in which isobaric processes of steam overheating are replaced by isochoric ones with preliminary cooling of the refrigerant that cools the condenser. If the existing Ukrainian thermal power plants are modified in this way, it is possible to save 14 288 256 tons of fuel, the specific heat of combustion of which is $40{\cdot}10^3$ kJ/kg, or 19 486 200 tons/year of conventional fuel.

Finally, in all proposed improvements of the thermodynamic cycle of the STP, the theoretical power of the STP increases with proportionally less heat supplied in the cycle.

The lower the condensation temperature of STP steam, the greater is not only the annual fuel economy, but most importantly, the thermal pollution of the Earth's atmosphere is significantly reduced, which is very relevant in connection with the current situation surrounding the problem of global climate warming.

It is clear that the proposed improvements will complicate the design of STPs and increase the cost of their production, but these are one-time costs, while fuel is used and thermal pollution of the atmosphere is carried out constantly during the operation of the TPP.

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