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# IMPROVED THERMODYNAMIC CYCLE OF A STEAM TURBINE PLANT

Georg K. Lavrenchenko[1], Alexey G. Slinko[2], Artem S. Boychuk[2], Vitalii M. Halkin[2], Serhii V. Kozlovskyi[2]

*[1] LLC «Institute of Low Temperature Energy Technologies», POB 188, Odessa, 65026, Ukraine*

*[2] Odessa National Maritime University, 34 Mechnikova str., Odesa, 65029, Ukraine.*

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## Abstract

**In modern steam turbine installations, steam is superheated in an isobaric process. As is known from technical thermodynamics, when the working fluid is heated from a given state to the same temperature, the heat in the isobaric process is greater than the heat of the isochoric one. In addition, the expansion of the working fluid during the isobaric process of heat supply is harmful, since in this case no external work is performed, that is, the main property of the working fluid is lost – its physical expansion in order to obtain external mechanical work. In an isochoric process, steam, like a mechanical spring, accumulates more mechanical energy at a lower cost of heat. Therefore, it is proposed in steam turbine plants that both primary superheating and subsequent steam superheating processes be carried out isochorically. Comparative calculations of the characteristics of steam turbine installation, operating according to the Rankine cycle with primary superheated steam up to a pressure of 6.0 MPa and a temperature of 600 °C, at a steam pressure in the condenser of 0.004 MPa, with the characteristics of an installation with an isochoric steam superheating process with the same initial data, showed significant energy and economic advantages of the latter in comparison with basic. So, with a lower fuel consumption in the modified installation (by 10.1 %), its specific work increased by 3.9 %, and thermal efficiency – by 11.2 %. Even more significant energy and economic advantages have a modified STP with one additional reheat of steam: with a lower fuel consumption by 15.9 %, the specific work increased by 6.8 %, and the thermal efficiency increased by 18.8 %. Due to the smaller volumes of steam at the end of the expansion, the weight and size parameters of the turbines and the condenser of the modified unit are reduced by 5.2 % compared to the base unit. Considering the above conclusions and the scale of use of STP at modern power plants, where up to ten intermediate steam reheats are used, the proposed modernization of their thermodynamic cycles guarantees even greater energy and economic effects.**

Keywords: thermodynamic cycle; isobaric and isochoric steam superheating processes; steam turbine plant; plant modification; performance indicators.

# УДОСКОНАЛЕНИЙ ТЕРМОДИНАМІЧНИЙ ЦИКЛ ПАРОТУРБІННОЇ УСТАНОВКИ

Георгій К. Лавренченко[1], Олексій Г. Слинько[2], Артем С. Бойчук[2], Віталій М. Галкін[2], Сергій В. Козловський[2]

*[1]ТОВ «Інститут низькотемпературних енерготехнологій», а/с 188, Одеса, 65026, Україна,*

*[2]Одеський національний морський університет, вул. Мечникова, 34, Одеса, 65029, Україна*

## Анотація

**У сучасних паротурбінних установках пара перегрівається в ізобарному процесі. Як відомо з технічної термодинаміки, за нагрівання робочого тіла від заданого стану до однакової температури, теплота за ізобарного процесу більша, ніж за ізохорного. Крім того, розширення робочого тіла в ізобарному процесі підведення тепла є шкідливим, оскільки не виконується зовнішня робота і втрачається фізичне розширення з метою отримання зовнішньої механічної роботи. В ізохорному процесі пара, подібно до механічної пружини, накопичує більше механічної енергії за менших витрат теплоти. Тому в паротурбінних установках пропонується як первинний перегрів, так і подальші процеси перегріву пари здійснювати ізохорно. Порівняльні розрахунки характеристик паротурбінної установки, що працює за циклом Ренкіна з первинним перегрівом пари до тиску 6.0 МПа і температури 600 °С (тиск пари в конденсаторі 0.004 МПа) з характеристиками установки з ізохорним процесом перегріву пари за тих самих вихідних даних, показали значні енергетичні та економічні переваги останньої в порівнянні з базовою. Так, за менших витратах палива в модифікованій установці (на 10.1 %) її питома робота зросла на 3.9 %, а тепловий ККД – на 11.2 %. Ще суттєвіші енергетичні та економічні переваги має модифікована ПТУ з одним додатковим підігрівом пари: за меншої витраті палива на 15.9 % питома робота зросла на 6.8 %, а тепловий ККД – на 18.8 %. Завдяки меншим об'ємам пари в кінці розширення масогабаритні параметри турбін і конденсатора модифікованої установки зменшуються на 5.2 % порівняно з базовою установкою. Враховуючи наведені вище висновки та масштаби використання ПТУ на сучасних електростанціях, де використовується до десяти проміжних пароперегрівів, запропонована модернізація їх термодинамічних циклів гарантує ще більший енергетичний та економічний ефекти.**

*Ключові слова*: термодинамічний цикл; ізобарний і ізохорний процеси перегріву пари; паротурбінна установка; модифікація установки; показники ефективності.

**Corresponding author: e-mail address: nemio@ukr.net*

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## Introduction.

Steam turbine plants remain the basis of the energy sector for converting thermal energy into electrical energy, therefore any improvement of them, and even more so of their base – thermodynamic cycles – is relevant. As is known, most STP operate according to the thermodynamic Rankine cycle on superheated steam. The formation and superheating of steam in this cycle is carried out at a constant pressure [1; 2]. A distinctive feature of the isobaric process in comparison with the isochoric process of heat supply is the premature expansion of steam during its formation and overheating, for which part of the supplied heat is spent [3–8]. Therefore, the isobaric process of heat supply, in comparison with the isochoric process, has a dual disadvantage in terms of converting heat into mechanical energy: additional heat consumption and undesirable (premature) expansion of the working fluid outside the mechanism for converting its potential energy (compressed spring) into kinetic energy [9–14]. Therefore, in all heat-to-work converters, it is desirable to replace the isobaric process of heat supply with an isochoric one. An example of such a replacement of the isobaric process of steam overheating by an isochoric process in the thermodynamic cycle of steam turbine plants operating according to the Rankine cycle (the basic cycle of steam power engineering) is considered in this paper [15–17].

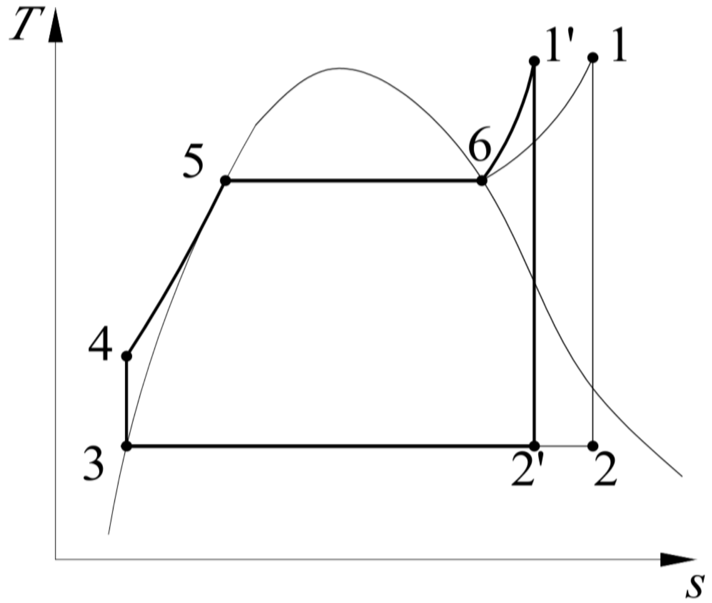


**Fig. 1. Joint image of thermodynamic Rankine cycles of STP on superheated steam with isobaric (1-2-3-4-5-6-1) and isochoric (1′-2′-3-4-5-6-1′) steam superheating processes**

*Modified thermodynamic Rankine cycle on superheated steam with isochoric process of steam superheating.* On Figure 1 shows together the thermodynamic cycles of the basic Rankine cycle (with an isobaric steam superheating process in a steam boiler) – 1-2-3-4-5-6-1, and the modified Rankine cycle (with an isochoric steam superheating process) – 1′-2′-3-4-5-6-1′. The compared cycles are constructed at the same temperatures of cold and hot heat sources and differ only in steam overheating processes: 6-1 - is an isobaric process, 6-1′ – is an isochoric process [18–23].

Comparative calculations were made for the STP cycle with the parameters of steam entering the turbine blades: pressure – 6.0 MPa and temperature – 600 °C. Steam condensation pressure – 0.004 MPa [24; 25].

The thermodynamic properties of water and water vapor at the characteristic points of the considered cycles were determined using the REFPROP program [2] and are given in Table 1.

*Table 1*

**Thermodynamic properties of water and steam at characteristic points of thermodynamic Rankine cycles on superheated steam (basic and modified)**

| points | *t*,°C | *p, MPa* | ρ, *kg/m³* | *u, kJ/kg* | *h, kJ/kg* | *s, kJ/(kg·K)* |
|---|---|---|---|---|---|---|
| 1 | 600 | 6.0 | 15.322 | 3267.7 | 3658.7 | 7.1693 |
| 2 | 28.96 | 0.004 | 0.034298 | 2043.1 | 2159.7 | 7.1693 |
| 3 | 28.96 | 0.004 | 995.92 | 121.38 | 121.39 | 0.42239 |
| 4 | 29.09 | 6.0 | 998.54 | 121.39 | 127.40 | 0.42239 |
| 5 | 275.59 | 6.0 | 758.00 | 1206.0 | 1213.9 | 3.0278 |

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*Continued table 1*

| | | | | | | |
|---|---|---|---|---|---|---|
| 6 | 275.59 | 6.0 | 30.818 | 2589.9 | 2784.6 | 5.8901 |
| 1′ | 600 | 11.72 | 30.818 | 3230.9 | 3611.2 | 6.8183 |
| 2′ | 28.96 | 0.004 | 0.03618 | 1943.1 | 2053.7 | 6.8183 |

Using tabular data, the following characteristics of the energy and economic efficiency of cycles are calculated [3-5].

The specific amount of heat supplied in the cycle:

– Rankine

$$q_1^R = h_1 - h_4 = 3658.7 - 127.40 = 3531.3\,kJ/kg\,;$$

– in the modified Rankine cycle

$$q_1^{mR} = (h_6 - h_4) + (u_1 - u_6) = (2784.6 - 127.4) + (3230.9 - 2589.9) = 3298.2\,kJ/kg.$$

Reducing the specific amount of heat supplied in the modified cycle:

$$\delta q_1 = \frac{q_1^{mR} - q_1^R}{q_1^R} \cdot 100 = \frac{3298.2 - 3531.3}{3531.3} \cdot 100 = -6.6\ \%.$$

The amount of heat removed to the environment in the cycle:

– Rankine

$$q_2^R = h_2 - h_3 = 2159.7 - 121.39 = 2038.31\,kJ/kg\,;$$

– in the modified Rankine cycle

$$q_2^{mR} = h_{2'} - h_3 = 2053.7 - 121.39 = 1932.31\,kJ/kg.$$

Reducing the specific amount of heat removed in the modified cycle:

$$\delta q_2 = \frac{q_2^{mR} - q_2^R}{q_2^R} \cdot 100 = \frac{1932.31 - 2038.31}{2038.31} \cdot 100 = -5.2\ \%.$$

Specific work done in a cycle:

– Rankine

$$l^R = h_1 - h_2 = 3658.7 - 2159.7 = 1499.0\ \ kJ/kg\,;$$

– in the modified Rankine cycle

$$l^{mR} = h_{1'} - h_{2'} = 3611.2 - 2053.7 = 1557.5\ \ kJ/kg.$$

Increasing specific work:

$$\delta l = \frac{l^{mR} - l^R}{l^R} \cdot 100 = \frac{1557.5 - 1499.0}{1499.0} \cdot 100 = -3.9\ \%.$$

Thermal cycle efficiency:

– Rankine

$$\eta_t^R = \frac{l^R}{q_1^R} = \frac{1499}{3531.1} = 0{,}4245;$$

– modified Rankine cycle

$$\eta_t^{mR} = \frac{l^{mR}}{q_1^{mR}} = \frac{1557.5}{3298.2} = 0.4722.$$

Increasing thermal efficiency in a modified cycle:

$$\delta\eta_t = \frac{\eta_t^{mR} - \eta_t^R}{\eta_t^R} \cdot 100 = \frac{0.4722 - 0.4245}{0.4245} \cdot 100 = 11.2\ \%.$$

Specific (per 1 kWh of energy) costs:

- steam

$$d^R = \frac{3600}{l^R} = \frac{3600}{1499} = 2.402\ \ \frac{kg}{kW \cdot h};$$

$$d^{mR} = \frac{3600}{l^{mR}} = \frac{3600}{1557.5} = 2.311\ \ \frac{kg}{kW \cdot h};$$

– heat

$$Q_1^R = \frac{3600}{\eta_t^R} = \frac{3600}{0.4245} = 8480.6\ \ \frac{kg}{kW \cdot h};$$

$$Q_1^{mR} = \frac{3600}{\eta_t^{mR}} = \frac{3600}{0.4722} = 7623.9\ \ \frac{kg}{kW \cdot h};$$

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– fuel

$$b^R = \frac{3600}{Q_H^P \eta_t^R} = \frac{3600}{40 \cdot 10^3 \cdot 0.4245} = 0.2120 \ \frac{kg}{kW \cdot h};$$

$$b^{mR} = \frac{3600}{Q_H^P \eta_t^{mR}} = \frac{3600}{40 \cdot 10^3 \cdot 0.4722} = 0.1906 \ \frac{kg}{kW \cdot h}.$$

Relative decrease in used:

- steam

$$\delta d = \frac{d^{mR} - d^R}{d^R} \cdot 100 = \frac{2.311 - 2.402}{2.402} \cdot 100 = -3.8\ \%;$$

– heat

$$\delta Q_1 = \frac{Q_1^{mR} - Q_1^R}{Q_1^R} \cdot 100 = \frac{7623.9 - 8480.6}{8480.6} \cdot 100 = -10.1\ \%;$$

– fuel

$$\delta b = \frac{b^{mR} - b^R}{b^R} \cdot 100 = \frac{0.1906 - 0.2120}{0.2120} \cdot 100 = -10.1\ \%.$$

For the convenience of analyzing the performed comparative calculations of the compared cycles, we summarize their results in Table 2.

*Table 2*

**Comparison of the main efficiency characteristics of the basic Rankine cycle on superheated steam and hot and cold heat sources modified at the same temperatures**

| No. | Characteristics | Rankine cycle | Modified Rankine cycle | Relative change in characteristic *),% |
|---|---|---|---|---|
| 1 | Specific heat input in cycle *$q_1$, kJ/kg* | 3531.3 | 3298.2 | –6.6 |
| 2 | Specific heat removed in cycle *$q_2$, kJ/kg* | 2038.31 | 1932.31 | –5.2 |
| 3 | Specific work in the cycle *l, kJ/kg* | 1499.0 | 1557.5 | +3.9 |
| 4 | Thermal efficiency, $\eta_t$ | 0.4245 | 0.4722 | +11.2 |
| 5 | Specific steam consumption *d, kg/(kW·h)* | 2.402 | 2.311 | –3.8 |
| 6 | Specific heat consumption *$Q_1$, kJ/(kW·h)* | 8480.6 | 7623.9 | –10.1 |
| 7 | Specific fuel consumption *b, kg/(kW·h)* | 0.2120 | 0.1906 | –10.1 |

*) a negative value of the characteristic deviation means that it has decreased in the modified cycle, a positive value means that it has increased.

Analyzing the tabular data, we can draw an unambiguous conclusion:

– by replacing the isobaric process of steam superheating with isochoric, one can:

– with lower fuel consumption (10.1 %), increase the specific work of the STP cycle by 3.9 % and increase the thermal efficiency by 11.2 %;

– reduce the weight and size parameters of the condenser turbine by 5.2 %.

*Thermodynamic Rankine cycles with one additional steam superheat.* Figure 2 shows together the thermodynamic cycles of the basic Rankine cycle withone intermediate isobaric steam superheating process – 1-7-8-9-3-4-5-6-1, and the modified cycle with isochoric steam superheating processes – 1′-7-8′-9'-3-4-5-6-1'. The compared cycles are considered at the same temperatures of cold and hot heat sources and intermediate steam superheating, but differ in steam superheating processes: 6-1 and 7-8 are isobaric processes, 6-1' and 7'-8' are isochoric processes.

The thermodynamic properties of water and water vapor at the characteristic points of the compared cycles were determined using the REFPROP program [2] and are given in Table 3.

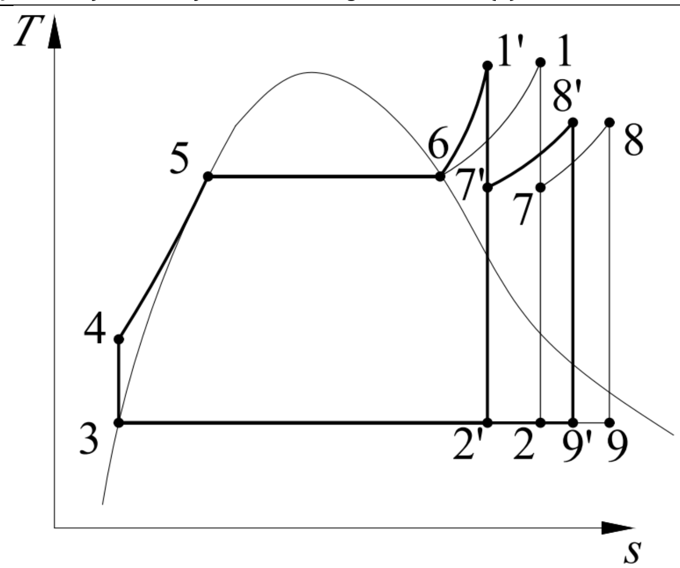


**Fig. 2. Joint image of thermodynamic cycles of STP with isobaric (1-7-8-9-3-4-5-6-1) and isochoric (1′-7′-8′-9′-3-4-5-6-1′) steam overheating processes**

*Table 3*

**Thermodynamic properties of water and steam at characteristic points of thermodynamic cycles of STP with isobaric (1-7-8-9-3-4-5-6-1) and isochoric (1′-7′-8′-9′-3-4-5-6-1′) steam overheating processes**

| points | *t*,°C | *p, MPa* | ρ, *kg/m³* | *u, kJ/kg* | *h, kJ/kg* | *s, kJ/(kg·K)* |
|---|---|---|---|---|---|---|
| 1 | 600 | 6.0 | 15.322 | 3267.7 | 3658.7 | 7.1693 |
| 2 | 28.96 | 0.004 | 0.034298 | 2043.1 | 2159.7 | 7.1693 |
| 3 | 28.96 | 0.004 | 995.92 | 121.38 | 121.39 | 0.42239 |
| 4 | 29.09 | 6.0 | 998.54 | 121.39 | 127.40 | 0.42239 |
| 5 | 275.59 | 6.0 | 758.00 | 1206.0 | 1213.9 | 3.0278 |
| 6 | 275.59 | 6.0 | 30.818 | 2589.9 | 2784.6 | 5.8901 |
| 7 | 168.37 | 0.3 | 1.5046 | 2600.8 | 2800.2 | 7.1693 |
| 8 | 560 | 0.3 | 0.78152 | 3232.4 | 3616.3 | 8.4886 |
| 9 | 28.96 | 0.004 | 0.028512 | 2418.0 | 2558.3 | 8.4886 |
| 1′ | 600 | 11.72 | 30.818 | 3230.9 | 3611.2 | 6.8183 |
| 2′ | 28.96 | 0.004 | 0.03618 | 1943.1 | 2053.7 | 6.8183 |
| 7′ | 133.52 | 0.3 | 1.7063 | 2478.6 | 2654.4 | 6.8183 |
| 8′ | 560.00 | 0.6537 | 1.7063 | 3230.1 | 3613.2 | 8.1263 |
| 9′ | 28.96 | 0.04 | 0.030038 | 2315.7 | 2448.8 | 8.1263 |

Using the tabular data, the following characteristics of the STP with one intermediate steam reheat were calculated.

The specific amount of heat supplied in the cycle:

– Rankine

$$q_1^R = (h_{1'} - h_4) + (h_{8'} - h_7) = (3658.7 - 127.40) + (3616.3 - 2800.2) = 4347.4\ kJ/kg;$$

– in the modified Rankine cycle

$$q_1^{mR} = (h_6 - h_4) + (u_{1'} - u_6) + (u_{8'} - u_{7'}) = (2784.6 - 127.40) + (3230.9 - 2589.9) + \\ +(3230.1 - 2478.6) = 4049.7\ kJ/kg.$$

Reducing the specific amount of heat supplied in the modified cycle:

$$\delta q_1 = \frac{q_1^{mR} - q_1^R}{q_1^R} \cdot 100 = \frac{4049.7 - 4347.4}{4347.4} \cdot 100 = -6.8\ \%.$$

The amount of heat removed to the environment in the cycle:

– Rankine

$$q_2^R = h_9 - h_3 = 2558.3 - 121.39 = 2436.9\ kJ/kg;$$

– in the modified Rankine cycle

$$q_2^{mR} = h_{9'} - h_3 = 2448.8 - 121.39 = 2327.41\ kJ/kg.$$

Reducing the specific amount of heat removed in the modified cycle:

$$\delta q_2 = \frac{q_2^{mR} - q_2^R}{q_2^R} \cdot 100 = \frac{2327.41 - 2436.9}{2436.9} \cdot 100 = -4.5\ \%.$$

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Specific work done in a cycle:
– Rankine
$$l^R = (h_1 - h_7) + (h_8 - h_9) = (3658{,}7 - 2800{,}2) + (3616{,}3 - 2558{,}3) = 1916{,}5\ kJ/kg;$$
– in the modified Rankine cycle
$$l^{mR} = (h_{1'} - h_{7'}) + (h_{8'} - h_{9'}) = (3611{,}2 - 2654{,}4) + (3613{,}2 - 2448{,}8) = 2121{,}2\ kJ/kg.$$
Increasing the specific work obtained in the modified cycle:
$$\delta l = \frac{l^{mR} - l^R}{l^R} \cdot 100 = \frac{2121.2 - 1916.5}{1916.5} \cdot 100 = 6.8\ \%.$$
Thermal cycle efficiency:
– Rankine:
$$\eta_t^R = \frac{l^R}{q_1^R} = \frac{1916.5}{4347.4} = 0.4408;$$
– modified Rankine cycle
$$\eta_t^{mR} = \frac{l^{mR}}{q_1^{mR}} = \frac{2121.2}{4049.7} = 0.5238.$$
Increasing thermal efficiency in a modified cycle:
$$\delta\eta_t = \frac{\eta_t^{mR} - \eta_t^R}{\eta_t^R} \cdot 100 = \frac{0.5238 - 0.4408}{0.4408} \cdot 100 = 18.8\ \%.$$
Specific (per 1 kWh of energy) costs:
- steam
$$d^R = \frac{3600}{l^R} = \frac{3600}{1916.5} = 1.8784\ \frac{kg}{kW \cdot h};$$
$$d^{mR} = \frac{3600}{l^{mR}} = \frac{3600}{2121{,}2} = 1{,}6972\ \frac{kg}{kW \cdot h};$$
– heat
$$Q_1^R = \frac{3600}{\eta_t^R} = \frac{3600}{0.4408} = 8166.97\ \frac{kJ}{kW \cdot h};$$
$$Q_1^{mR} = \frac{3600}{\eta_t^{mR}} = \frac{3600}{0.5238} = 6872.85\ \frac{kJ}{kW \cdot h};$$
– fuel
$$b^R = \frac{3600}{Q_{\mathrm{H}}^P \eta_t^R} = \frac{3600}{40 \cdot 10^3 \cdot 0.4408} = 0.2042\ \frac{kg}{kW \cdot h};$$
$$b^{mR} = \frac{3600}{Q_{\mathrm{H}}^P \eta_t^{mR}} = \frac{3600}{40 \cdot 10^3 \cdot 0.5238} = 0.1718\ \frac{kg}{kW \cdot h}.$$
Relative reduction:
– used steam
$$\delta d = \frac{d^{mr} - d^R}{d^R} \cdot 100 = \frac{1.6972 - 1.8784}{1.8784} \cdot 100 = -9.6\ \%;$$
– heat input $Q_1$
$$\delta Q_1 = \frac{Q_1^{mr} - Q_1^R}{Q_1^R} \cdot 100 = \frac{6872.85 - 8166.97}{8166.97} \cdot 100 = -15.8\ \%;$$
– consumed fuel
$$\delta b = \frac{b^{mr} - b^R}{b^R} \cdot 100 = \frac{0.1718 - 0.2042}{0.2042} \cdot 100 = -15.9\ \%.$$
For the convenience of analyzing the results of the comparative calculations of the compared cycles, their most important characteristics are in Table 4.

*Table 4*

**Comparison of the main characteristics of the efficiency of the basic Rankine cycles of STP with one and two steam superheats and the corresponding modified cycles at the same temperatures of hot and cold heat sources and the temperature and pressure of the intermediate steam superheat**

| No | Characteristics | Rankine cycle | | Modified Cycle | |
|---|---|---|---|---|---|
| | | with one superheating | with two superheating | with one superheating | with two superheating |
| 1 | Specific heat input in cycle $q_1$, *kJ/kg* | 3531 | 4347 | 3298 | 4050 |
| 2 | Specific heat removed in cycle $q_2$, *kJ/kg* | 2038 | 2437 | 1932 | 2327 |
| 3 | Specific work in the cycle *l*, *kJ/kg* | 1499 | 1917 | 1558 | 2121 |
| 4 | Thermal efficiency, $\eta_t$ | 0.4245 | 0.4408 | 0.4722 | 0.5238 |
| 5 | Specific steam consumption *d*, *kg/(kW·h)* | 2.402 | 18784 | 2.311 | 1.6972 |
| 6 | Specific heat consumption $Q_1$, *kJ/(kW·h)* | 8481 | 8167 | 7624 | 6873 |
| 7 | Specific fuel consumption *b*, *kg/(kW·h)* | 0.2120 | 0.2042 | 031906 | 0.1718 |
| 8* | Relative change in the specific heat input in the cycle $\delta q_1$,% | – | – | -6.6 | -6.8 |
| 9* | Relative change in the specific heat removed in the cycle $\delta q_2$,% | – | – | -5.2 | -4.5 |
| 10* | Relative change in specific work in the cycle $\delta l$,% | – | – | 3.9 | 6.8 |
| 11* | Relative change in thermal efficiency $\delta \eta_t$ ,% | – | – | 11.2 | 18.8 |
| 12* | Relative change in specific steam consumption $\delta d$,% | – | – | -3.8 | -9.6 |
| 13* | Relative change in specific heat consumption $\delta Q_1$,% | – | – | -10.1 | -15.8 |
| 14* | Relative change in specific fuel consumption $\delta b$,% | – | – | -10.1 | -15.9 |

*) Characteristic deviations are calculated relative to the corresponding characteristic of the corresponding Rankine cycle(No. 8-14): a negative value of the characteristic deviation means that it has decreased in the modified cycle, a positive value means that it has increased.

Analyzing the tabular data, we can draw an unambiguous conclusion:

– by replacing the isobaric process of steam superheating with isochoric, one can:

– for a unit with one steam superheat at lower fuel consumption (10.1 %), increase the specific work of the STP cycle by 3.9 % and increase the thermal efficiency by 11.2 %;

– for an installation with two steam superheats at a lower fuel consumption (15.9 %), increase the specific work by 6.8 % and increase the thermal efficiency by 18.8 %

– reduce the weight and size parameters of the turbines and the condenser by approximately 5.2 %.

## Conclusion

Taking into account the conclusions above and the scale of use and capacity of the STP used at modern power plants, the proposed modernization of them promises a huge economic effect.

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