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CONVERSION OF LIQUID TO STEAM. HOW AND WHY?

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Abstract

To convert liquid into vapor, static and hydrodynamic methods are used. When using the hydrodynamic method, the transformation of liquid into vapor is realized for a small amount of liquid. This makes it possible to apply the isochoric process of superheating saturated steam. A small amount of liquid is compressed and heated isobarically to saturation temperature. Further, it is supplied in sprayed form to a vertically located surface, the temperature of which, in relation to the supplied liquid, is higher. In this case, the liquid instantly turns into saturated vapor. A surface continuously heated by a hot heat source is placed in a closed volume. It is equipped with valves that regulate the moment and quantity of liquid injected, as well as the final pressure and superheat temperature of the steam. The performance and efficiency of the proposed hydrodynamic method for steam generation with an isochoric process of its overheating has been tested using the example of a thermodynamic cycle of a steam turbine plant with an intermediate overheating of steam with a power of 20000 kW. The initial data on the parameters of the steam entering the turbine blades: pressure 10 MPa, temperature 510 °C, temperature of the intermediate overheating of the steam 500 °C, condensation pressure 0.005 MPa. Comparative calculations have shown that the proposed cycle in terms of the main technical and economic indicators significantly exceeds the classical cycle of steam turbine plants widely used in practice. In addition to numerical indicators that positively characterize the proposed cycle, its design and operational indicators also testify in its favor. So, it lacks a large, massive and structurally complex steam boiler; a more thermodynamically efficient isochoric steam superheating process is used.

Keywords: Isobaric and isochoric thermodynamic processes of steam overheating; Steam turbine plant; Thermal efficiency.

ПЕРЕТВОРЕННЯ РІДИНИ У ПАРУ. ЯК І НАВІЩО?

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Анотація

Для перетворення рідини на пару використовують статичний і гідродинамічний методи. Гідродинамічний метод перетворення рідини на пару реалізується для невеликої кількості рідини, що стискається та ізобарно нагрівається до температури насичення. Далі її подають у розпиленому вигляді на вертикально розташовану поверхню, температура якої вища за температуру поданої рідини. Рідина миттєво перетворюється на насичену пару. Поверхня, що безперервно нагрівається, поміщена в замкнений об'єм, обладнаний клапанами регулювання моменту і кількості рідини, а також кінцевий тиск і температуру перегріву пари. Працездатність і ефективність пропонованого гідродинамічного методу пароутворення з ізохорним процесом її перегріву перевірено на прикладі термодинамічного циклу паротурбінної установки з проміжним перегрівом пари потужністю 20000 кВт. Вихідні пара-метри пари, що надходить на лопатки турбіни: тиск 10 МПа, температура 510 °С, температура проміжного перегріву пари 500 °С, тиск конденсації 0.005 МПа. Порівняльні розрахунки показали, що пропонований цикл за основними техніко-економічними показниками істотно перевершує класичний цикл паротурбінних установок. Усі числові, конструктивні та експлуатаційні показники свідчать на його користь. Так, у ньому відсутній паровий котел, який з точки зору конструкції складний, великогабаритний і масивний, та використовується ефективніший із термодинамічної точки зору ізохорний процес перегріву пари.

Ключові слова: Ізобарний та ізохорний термодинамічні процеси перегріву пари; Паротурбінна установка; Термічний коефіцієнт корисної дії.

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Introduction

One of the disadvantages of STP is the presence of a massive large-sized steam boiler with large losses of fuel combustion heat to the environment. The static method of vaporization used in it predetermines considerable time for its commissioning and decommissioning. In addition, a large amount of heat expended when putting it into operation is released when the steam turbine plant is stopped. In modern boilers, saturated steam is superheated at constant pressure. In the isobaric process of heat supply, there is a concomitant expansion of the working fluid. Since this expansion is carried out outside the mechanism of converting the body's potential energy into mechanical energy, its efficiency decreases. In other words, during the isobaric transformation of liquid into vapor, part of the available heat of combustion of the fuel is spent on its unproductive (premature) expansion. This issue has been studied and taken into account in the works [1; 2].

Modern technology does not yet possess the technology of a direct continuous isochoric process of converting liquid into superheated vapor. However, isochoric superheating of saturated steam into superheated steam is quite realizable with the modern development of automation equipment. This method of converting liquid into superheated vapor is proposed, calculated and analyzed in this paper. Since the current method of converting a liquid into superheated vapor simultaneously involves a large mass of a low-mobility liquid, it can be conventionally called *static.* In contrast, a *hydrodynamic method* is proposed, which is carried out with a small amount of liquid that is in the process of converting to vapor under the action of gravitational forces.

The essence of the proposed method is that a small amount of pre-compressed and heated liquid is converted into saturated steam. Superheating of steam is carried out in the same closed volume where this steam is formed [3–6].

Plant schematic diagram

Fig. 1 shows a schematic diagram of a STP with a hydrodynamic device for converting liquid into steam. Consider the device and principle of operation of such a steam turbine plant.

The main distinguishing and basic element of the installation is the steam-generating device 1, structurally consisting of two vertically arranged concentric cylinders, in which the inner cylinder serves as a combustion chamber 2 of liquid or gaseous fuel.

Compressed and preheated to a saturation state, the liquid is sprayed on the outer surface of the combustion chamber 2. The liquid supplied in a thin layer by the atomizer 6 instantly turns into saturated steam. This vapor completely fills chamber 5, after which the isochoric process of its overheating begins [7–9]. The moment and amount of liquid injected into the intercylinder space is regulated by valve 23.

The final pressure/temperature of superheated steam is controlled by setting the fifth from the top exhaust valve 7. The formation of saturated steam and its superheating is carried out with simultaneously closed valves 7. When the set pressure/temperature of superheated steam, the outlet valve 7 opens. By setting this valve, it is possible to regulate the indicated parameters. Valve 7 remains open until the vapor pressure in the intercylinder space drops to the pressure of the saturated liquid in front of the inlet valve 23. Through it, with the valve 7 closed, a metered amount of saturated liquid is supplied to the atomizer 6. Liquid is supplied by pump 20 through non-return valve 21 to economizer 4 of steam generating device 1.

Compressed and regeneratively heated in economizer 4 to saturation, the liquid enters receiver 22 and then into chamber 5. To reduce the pressure pulsation of superheated steam entering the turbine blades, by one superheated steam receiver 8 several steam generating devices must operate. The excess of the liquid injected into the vaporizing device and not turned into steam is returned to the condensate collector 19 through the throttle valve 25.

The base unit provides for reheating, which leads to the use of two turbines – high (HPT) 11 and medium (MPT) 12 pressure. The proposed plant uses an isochoric process of steam superheating, which makes it possible to regulate the temperature/pressure of its superheating by appropriately adjusting the inlet/outlet valves 7. This led to the appearance of a low-pressure turbine (LPT) 13 in the proposed plant. The return of the exhaust steam from the HPT and MPT to the respective intermediate superheaters is carried out using booster compressors 10, which supply the exhaust steam to the receivers 9.

The temperature distribution of the steam supplied to the turbine inlet is taken such that the entropy of the steam entering the LPT is equal to the entropy of the steam at the end of steam expansion in the basic two-turbine unit (point 4). Feed water is heated to saturation in the economizer 4 of the steam generating device 1. This eliminates the use of the appropriate process of its regenerative heating and increases the capacity of the plant.

The purpose and principle of operation of the remaining elements of the STP under consideration is no different from the purpose and principle of operation of similar elements of a conventional STP and can be clarified based on the caption given below.

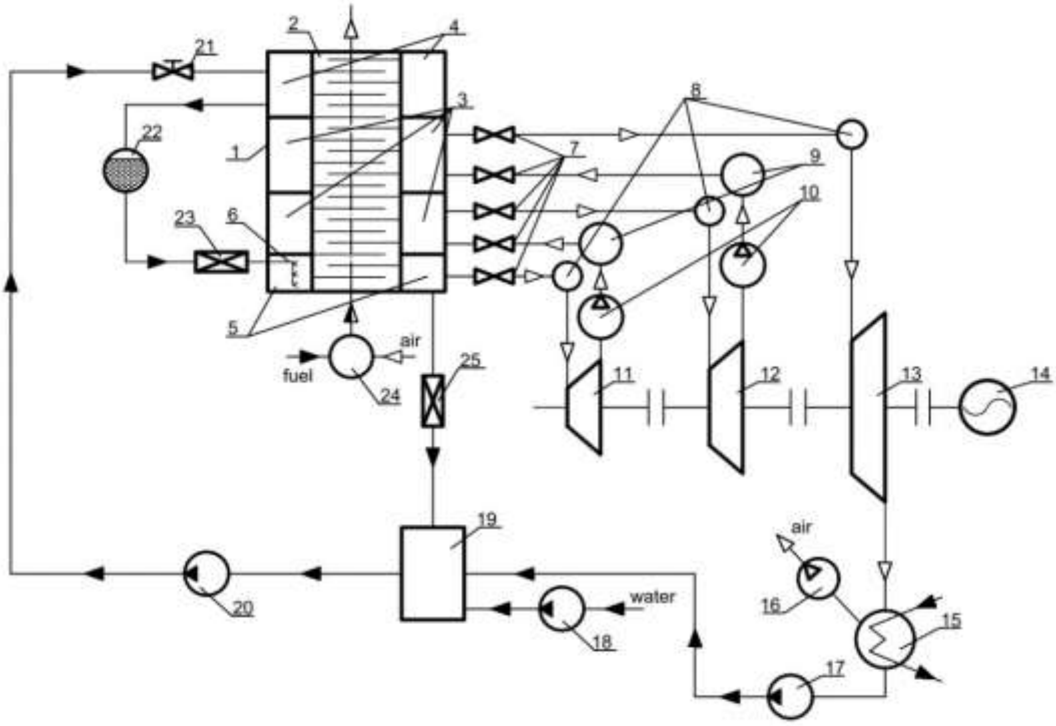

Fig. 1. Schematic diagram of a waste steam turbine plant with a hydrodynamic method of steam generation:
1 – steam generating device; 2 – combustion chamber; 3 – superheater; 4 – economizer; 5 – chamber for the formation of saturated steam and its overheating; 6 – saturated liquid atomizer; 7 – automatic outlet/inlet valves for superheated steam; 8 – inlet receivers of superheated steam; 9 – exhaust receivers of the exhaust steam of turbines; 10 – booster steam compressors of exhaust steam; 11, 12, 13 – high, medium and low pressure turbines; 14 – consumer of mechanical energy; 15 – condenser; 16 – vacuum compressor; 17 – condensate pump; 18 – additional liquid booster pump; 19 – condensate collector; 20 – feed pump; 21 – feed water valve; 22 – saturated liquid receiver; 23 – inlet valve of saturated liquid; 24 – fuel carburetor; 25 – throttle return valve of non-evaporated liquid.

A graphical comparison of thermodynamic cycles of steam turbine plants with static and hydrodynamic methods of steam production is shown in fig. 2.

Calculations and discussion

To confirm the operability and efficiency of the proposed method of steam generation, a thermal calculation of both thermodynamic cycles of a steam turbine plant with reheating of steam with a power of 20000 kW was performed. The initial data on the parameters of the steam supplied to the turbine blades: pressure 10 MPa, temperature 510 °C, steam reheat temperature 500 °C, condensation pressure 0.005 MPa.

As follows from a comparison of the cycles, their implementation took place with the same initial data, but different methods for obtaining and superheating steam [10–12].

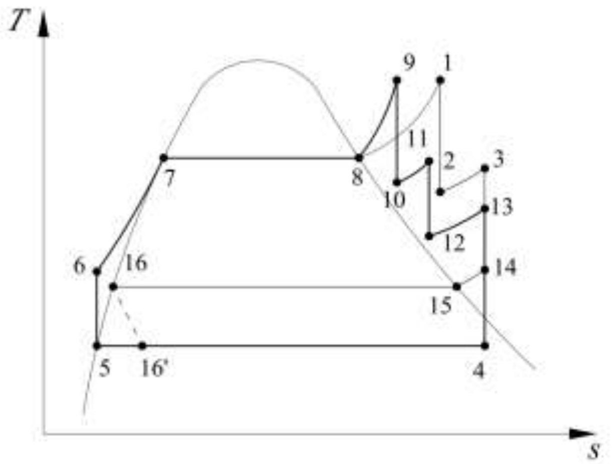

Fig. 2. Comparison of the thermodynamic cycle of STP with the isobaric process of steam overheating (1-2-3-4-5-6-7-8-1) and its modification with the isochoric process (9-10-11-12-13-4-5-6-7-8-9) with surface regenerative feed water heating (14-15-16-16′).

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The thermodynamic properties of water and water vapor in the compared cycles were calculated using the REFPROP program [13]. The results obtained are shown in Table 1.

Table 1

Thermodynamic properties of water and water vapor at characteristic points of compared cycles *)

points	*t*, °C	*p, MPa*	*v, m³/kg*	*u, kJ/kg*	*h, kJ/kg*	*s, kJ/(kg·K)*
1	510	10	0.033392	3066.9	3400.8	6.6325
2	285.94	0.23	0.10477	2740.7	2981.6	6.6325
3	500	0.23	0.15241	3114.4	3465.0	7.3660
4	32.87	0.005	25.367	2191.6	2318.4	7.60204
5	32.87	0.005	0.0010053	137.74	137.75	0.4762
6	33.12	10.0	0.001001	137.77	147.78	0.4762
7	311.0	10.0	0.0014526	1393.5	1408.1	3.3606
8	311.0	10.0	0.01803	2545.2	2725.5	5.6160
9	510.0	17.298	0.01803	2997.9	3309.8	6.2895
14	350.0	0.53557	0.53194	2882.5	3167.4	7.60204
15	154.44	0.53557	0.35130	2563.0	2751.2	6.7975
16	154.44	0.53557	0.0010955	650.77	651.36	1.8867
Modified cycle points						
10	219.90	0.23	0.086924	2602.0	2802.0	6.2895
11	500.0	3.9787	0.086924	3100.4	3446.3	7.0949
12	207.57	0.5	0.43263	2655.7	2872.0	7.0949
13	388.85	0.69933	0.43263	2943.2	3245.7	7.60204

*Points 10 to 13 refer to the modified STP cycle with hydrodynamic steam generation and isochoric process of steam superheating.

Specific work done in cycles:

- in base

$$l_p = (h_1 - h_2) + (1 - \alpha)(h_3 - h_4) - l_{pump} =$$
$$= (3400.8 - 2981.6) + (1 - 0.5009)(3465.0 - 2318.4) - 10.03 = 991.47\ kJ/kg,$$

where

$$l_{pump} = h_6 - h_5 = 147.78 - 137.75 = 10.03\ kJ/kg - \text{ pump work;}$$
$$\alpha = \frac{\mathrm{h}_7 - \mathrm{h}_6}{\mathrm{h}_{14} - \mathrm{h}_{16}} = \frac{1408.1 - 147.78}{3167.4 - 651.36} = 0.5009 - \text{steam extraction coefficient;}$$

- in modified

$$l_v = (h_9 - h_{10}) + (h_{11} - h_{12}) + (h_{13} - h_4) - l_{pump} =$$
$$= (3309.8 - 2802.0) + (3446.3 - 2872.4) +$$
$$+(3245.7 - 2318.4) - 10.03 = 1998.7\ kJ/kg.$$

Specific work increasing

$$\delta l = \frac{l_v - l_p}{l_p} \cdot 100 = \frac{1998.70 - 991.47}{991.47} \cdot 100 = 101.6\%.$$

Steam consumption:

– in base

$$m_p = \frac{N_\mathrm{T}}{l_p} = \frac{20000}{991.47} = 20.17\ kg/sec;$$

– in modified

$$m_v = \frac{N_\mathrm{T}}{l_v} = \frac{20000}{1998.7} = 10.07\ kg/sec.$$

Steam consumption reduction

$$\delta m = \frac{m_v - m_p}{m_p} \cdot 100 = \frac{10.07 - 20.17}{20.17} \cdot 100 = -50.07\%.$$

Heat of vaporization in both cycles

$$q_{p,v} = h_8 - h_7 = 2725.5 - 1408.1 = 1317.4\ kJ/kg.$$

Specific heat of isobaric processes of steam overheating in the basic cycle

$$\Delta q_p = (h_1 - h_8) + (h_3 - h_2) = (3400.8 - 2725.5) + (3465.0 - 2981.6) = 1158.7\ kJ/kg.$$

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Specific heat of isochoric steam superheating processes in the modified cycle

$$\Delta q_v = (u_9 - u_8) + (u_{11} - u_{10}) + (u_{13} - u_{12}) = (2997.9 - 2545.2) + (3100.4 - 2602.0) + \\ +(2943.2 - 2655.70) = 1238.6\ kJ/kg.$$

Total specific heat supplied in the base cycle

$$q_p = q_{p,v} + \Delta q_p = 1317.4 + 1158.7 = 2476.1\ \ kJ/kg.$$

Total specific heat supplied in the modified cycle

$$q_v = q_{p,v} + \Delta q_v = 1317.4 + 1238.6 = 2556.0\ \ kJ/kg.$$

Total heat supplied in the base cycle per unit time

$$Q_p = m_p q_p = 20.17 \cdot 2476.1 = 49942.94\ kW.$$

Total heat supplied in the modified cycle per unit time

$$Q_v = m_v q_v = 10.07 \cdot 2556.0 = 25738.92\ kW.$$

Thermal efficiency:

– base cycle

$$\eta_{t,p} = \frac{N_{\mathrm{T}}}{Q_p} = \frac{20000}{49942.94} = 0.4005;$$

– modified cycle

$$\eta_{t,v} = \frac{N_{\mathrm{T}}}{Q_v} = \frac{20000}{25738.92} = 0.7770.$$

The excess of the thermal efficiency of the modified cycle over the efficiency of the base

$$\delta\eta_t = \frac{\eta_{t,v} - \eta_{t,p}}{\eta_{t,p}} \cdot 100 = \frac{0.7770 - 0.4005}{0.4005} \cdot 100 = 93.9\%.$$

Specific consumptions per 1 kW·h of work:

– steam

$$d_p = \frac{3600}{l_p} = \frac{3600}{991.4} = 3.63\ kg/sec;$$

$$d_v = \frac{3600}{l_v} = \frac{3600}{1998.7} = 1.80\ kg/sec;$$

– heat

$$Q_1^p = \frac{3600}{\eta_{t,p}} = \frac{3600}{0.4005} = 8988.8\ kJ/(kW \cdot h);$$

$$\mathrm{Q}_1^{\mathrm{v}} = \frac{3600}{\eta_{t,v}} = \frac{3600}{0.7770} = 4633.2\ kJ/(kW \cdot h);$$

– fuel

$$b^P = \frac{3600}{Q^p \cdot \eta_{t,p}} = \frac{3600}{50 \cdot 10^3 \cdot 0.4005} = 0.1798\ kg/(kW \cdot h);$$

$$b^V = \frac{3600}{Q^p \cdot \eta_{t,v}} = \frac{3600}{50 \cdot 10^3 \cdot 0.777} = 0.09266\ kg/(kW \cdot h).$$

Relative reduction in cycle:

– steam

$$\delta d = \frac{d_v - d_p}{d_p} \cdot 100 = \frac{1.80 - 3.63}{3.63} \cdot 100 = -50.41\ \%;$$

– heat Q_1

$$\delta Q_1 = \frac{Q_1^v - Q_1^p}{Q_1^p} = \frac{4633.2 - 8988.8}{8988.8} = 48.46\ \%;$$

– fuel

$$\delta b = \frac{b_v - b_p}{b_p} \cdot 100 = \frac{0.09266 - 0.1798}{0.1798} \cdot 100 = 48.47\ \%.$$

For the convenience of analyzing the most important indicators of the performed comparative calculations of the compared cycles, summarized in Table 2.

Table 2

Comparison of the main characteristics of the efficiency of the basic cycle with isobaric processes of steam generation and overheating of steam and modified method of steam generation, as well as isochoric processes of steam overheating with the same parameters of the working fluid at the initial and final points of the cycles and plant power

No.	Characteristics	In base cycle	In modified cycle	Relative change *), %
1	Specific heat input in cycle q_1, *kJ/kg*	2476.1	2556.0	+3.2
2	Specific work in the cycle *l*, *kJ/kg*	991.47	1998.7	+101.6
3	Steam consumption per second *m*, *kg/sec*	20.17	10.07	-50.07
4	Thermal efficiency, η_t	0.4005	0.7770	+93.9
5	Specific steam consumption *d*, *kg/(kW·h)*	3.63	1.80	-50.41
6	Specific heat consumption Q_1, *kJ/(kW·h)*	8988.8	4633.2	-48.46
7	Specific fuel consumption *b*, *kg/(kW·h)*	0.1798	0.09266	-48.47
8	Specific heat supplied per unit time Q, kW	8988.8	4633.2	-46.46

*) 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, it can be stated that the STP cycle with the hydrodynamic method of steam generation and isochoric steam superheating processes is significantly superior to the cycle with the static steam generation method and isobaric steam superheating processes [14–20].

From a constructive point of view, the proposed cycle is simpler, although it requires the implementation of a thermodynamically more complex isochoric process of steam overheating. A high degree of valve synchronization is required, estimated in fractions of a second.

Conclusion

The performed calculations testify to the energy and weight and size advantages of the hydrodynamic method of vaporization over the currently used static method. Increasing the thermodynamic efficiency by almost 2 times with a high power of the STP will result in savings of tens or even hundreds of tons of fuel. The weight and size indicators of the hydrodynamic steam generating device are incomparably less than the corresponding characteristics of the corresponding steam boiler.

The structural simplicity of the steam generating device in comparison with an engineering-complex steam boiler is also a tangible advantage in favor of using the hydrodynamic method of steam generation in steam power engineering.

Finally, the hydrodynamic method of steam formation in STP equalizes them in terms of mobility with GTP and eliminates significant heat losses during their shutdown.

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