

Substantiation of the effectiveness of the hydro-steam turbines application in mine power complexes for excess heat recovery

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Abstract. The main problems connected with an increase in the performance efficiency of the mine power complexes, should include issues of rational use of fuel in power facilities, as well as the maximum consumption of produced thermal and electrical energy. One solution to this problem has been proposed to use in schematic solutions the reaction-type hydro-steam turbine for excess heat recovery of and generation of additional electrical energy. The condition has been set in the article for the choice of the configuration of a hydro-steam turbine nozzle with a high efficiency of energy conversion, as well as the variants have been studied of principle schemes for energy complexes: the scheme for a power facility, consisting of a gas-reciprocating module, on the shaft of which a module with a hydro-steam turbine is set, as well as the scheme for a module placement of back pressure turbine and hydro-steam turbine on the same shaft with a gas-reciprocating module. A comparative analysis has been performed of energy efficiency of the proposed principle schemes for the energy complexes.

1 Introduction

One of the developing directions of the Small Energy sector is the construction of mine power complexes (MPC), which reprocess the low-grade coals, waste coals and coalmine methane into thermal and electrical energy [1]. Based on the energy-saving cogeneration technology, MPC is a very prospective power facility, enabling through the use of the modular principle, determined by the load curve, the choice of basic equipment and the parameters of the thermal diagram, to respond flexibly to fluctuations in thermal and electrical loads and to ensure high efficiency of the power complex performance as a whole. When constructing this type of power complex, it is necessary to take into account the seasonal changes in thermal load, which significantly influence on the complex efficiency and the amount of produced electrical energy. One of the ways to improve the power complex efficiency with a seasonal decrease in heat loads is to install the reaction-type hydro-steam turbine (HST).

Two main physical phenomena are realized in the work of HST. The first is the rotation

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of the HST due to the action of the reaction forces (moments) of the jets of the outflowing two-phase mixture. The second is the formation of a two-phase mixture in a stream of under heated water until its saturation at decrease in the hot water pressure in the stream to the saturated vapours pressure. It is the phenomenon of water steam formation in a stream that makes it possible in HST plant to use the enthalpy stored in heated water to generate the electrical energy.

The purpose of this paper is to increase the efficiency of the MPC operation at variable thermal loads by means of hot water utilized from the energy modules of the mine power complexes in the reaction-type hydro-steam turbines.

2 Methods

To determine the conditions for high efficiency factor of thermal energy conversion into mechanical energy, in the HST-module the methods of analysis and generalization were used of results of the well-known theoretical and experimental studies, mathematical modelling based on the fundamental laws of mechanics and thermodynamics. The method for calculating the cogeneration schemes with the HST-module as part of the MPC was in determining the costs and parameters of the working body state in some key points located along the closed circuit of the working body movement. Based on the calculation results, the energy efficiency of using the proposed schemes was assessed. When performing the work, the well-known algorithms for solving the nonlinear algebraic equations and calculating the coefficients of the regression dependences were used.

2.1 Analysis of conditions for effective energy conversion in the HST nozzle

The HST work is based on the process of converting the thermal energy of heated water into the kinetic energy of a two-phase stream being boiled under the condition of decreasing pressure. In contrast to the thermodynamic cycle of classical steam-turbine plants, the HST cycle is based on the expansion of the working medium from the left boundary curve on the « p - v »-diagram into the two-phase area [2].

When moving with boiling up in a nozzle of a hydro-steam turbine, the two-phase stream is changed in its mechanical structure. When being boiled, the initially bubble structure of a stream during its movement through the nozzle at the point of inversion turns into a droplet-laden one. The two-phase media of these structures have a high degree of compressibility. One of the most important characteristics of compressibility is the sound velocity. The sound velocity establishes the relationship between the pressure differential and the density differential in a specific continuous medium. This ratio, being used in the equation of the law of momentum conservation (the Euler equation), makes it possible to calculate the change in velocity when a continuous medium moves in a certain channel (nozzle).

The monographs of V.E. Nakoryakov, B. G. Pokusaev, I. R. Schreiber, R. I. Nigmatulin, A. A. Nakorchevskiy, B.I. Bask are devoted to the theoretical calculations and experimental determination of the sound velocity in a two-phase medium [3 – 5]. The expressions for the sound velocity in a two-phase media, presented in [3, 4] are based on assumptions that replace the medium with real properties with a model medium with deamplification of non-equilibrium. In the monograph [5], the influence is analysed of various assumptions in determining the sound velocity in a two-phase medium. The interrelation has been set of this value with the condition of the occurrence of the critical flow regime. A universal approach to the definition of the concept ‘sound velocity in a two-phase mixture’ has been formulated on the basis of establishing the conditions under which the critical flow regime in a bubble stream is realized.

Based on this approach, a mathematical model of a bubble stream has been developed in [5], which takes into account the difference in the phase velocities. The differential equation for pressure [5] has been formulated on the basis of differential equations of the laws of mass conservation of the carrier and bubble phases. In these equations, there are summands $\pm(1/\rho_i) \cdot (d\rho_i/dp) \cdot (dp/dz)$, where i is the index to indicate the carrier phase parameters, which is equal to «1», and for the disperse phase parameters, equal to «2»; ρ_i – water and steam density, kg/m^3 ; p – pressure in a two-phase stream, Pa.

It should be noted that the density of the medium (water, steam) depends not only on pressure, but also on temperature. Therefore, the system of equations of the model should be supplemented by equations for determining the phases temperature. In [5], the equations for determining the temperatures are included into the mathematical model of the bubble stream. However, the coefficients of these equations with the derivatives are not used in forming the determinant for determining the conditions of the critical flow regime. In addition, to solve the problem set, it is necessary to use the equation of mechanical equilibrium of a bubble in differential form, which takes into account the pressure difference in the phases per the value of capillary pressure.

Thus, given the above additions, the mathematical model of a bubble stream will consist of nine differential equations. Two equations for the velocity of each phase

$$J_1 \cdot \left(1 + \frac{B_2}{2}\right) \cdot \left(\frac{dV_1}{dz}\right) - J_1 \cdot \left(\frac{B_2 \cdot V_2}{V_1}\right) \cdot \left(\frac{dV_2}{dz}\right) = -B_1 \cdot \left(\frac{dp_1}{dz}\right) - B_1 \cdot F_W + \left(\frac{B_1}{2}\right) \cdot \left(\frac{3 \cdot B_2}{2 \cdot R}\right) \cdot V_2 \cdot \left(\frac{dR}{dz}\right) \cdot (V_2 - V_1) + F_{21,\mu} \quad (1)$$

$$-J_1 \cdot \left(\frac{B_2}{2}\right) \cdot \left(\frac{dV_1}{dz}\right) + J_2 \cdot \left(1 + \frac{B_1}{2}\right) \cdot \left(\frac{dV_2}{dz}\right) = -B_2 \cdot \left(\frac{dp_1}{dz}\right) - B_2 \cdot F_W - \left(\frac{B_1}{2}\right) \cdot \left(\frac{3 \cdot B_2}{2 \cdot R}\right) \cdot V_2 \cdot \left(\frac{dR}{dz}\right) \cdot (V_2 - V_1) - F_{21,\mu} \quad (2)$$

where the value J_i – is determined by the expression

$$J_i = \rho_i \cdot B_i \cdot V_i \quad (3)$$

where B_i – the bulk concentrations of a phase; V_i – the velocities of a phase, m/s; R – bubbles radius, m; F_W – the frictional force of a stream against the nozzle wall, N; $F_{21,\mu}$ – the interphase friction force, N.

Two equations for the temperature of each phase

$$J_1 \cdot C_{1,p} \cdot \left(\frac{dT_1}{dz}\right) = B_1 \cdot V_1 \cdot \left(\frac{dp_1}{dz}\right) + B_1 \cdot V_1 \cdot F_W - Q_{12} \quad (4)$$

$$J_2 \cdot C_{2a,p} \cdot \left(\frac{dT_2}{dz}\right) = B_2 \cdot V_2 \cdot \left(\frac{dp_1}{dz}\right) + B_2 \cdot V_2 \cdot F_W + Q_{12} \quad (5)$$

where $C_{1,p}$ – the isobaric heat capacitance of water, $\text{J}/(\text{kg} \cdot \text{K})$; Q_{12} – the volumetric density of interphase heat transfer W/m^3 ; $C_{2a,p}$ – the isobaric heat capacitance of the steam-air mixture in the bubble, $\text{J}/(\text{kg} \cdot \text{K})$.

The equations for densities (water, steam, air)

$$d(\rho_i \cdot B_i \cdot V_i \cdot S)/dz = (-1)^i \cdot 4 \cdot \pi \cdot R^2 \cdot N_b \cdot j_{12} \cdot S \quad (6)$$

$$d(\rho_a \cdot B_2 \cdot V_2 \cdot S)/dz = 0 \quad (7)$$

where $i = 1, 2$; N_b – the volumetric density of bubbles, $1/m^3$; j_{12} – mass transfer rate per a bubble, $kg/(m^2 \cdot s)$; ρ_a – air density in the bubble, kg/m^3 ; S – the cross-sectional area of the HST nozzle, m^2 .

The equation of mechanical equilibrium of a bubble, written in differential form as follows

$$\left(\frac{dP_{VN}(\rho_2, T_2)}{dz} \right) + \left(\frac{dP_a(\rho_a, T_2)}{dz} \right) = \left(\frac{dp_1}{dz} \right) - \left(\frac{2 \cdot \sigma}{R^2} \right) \cdot \left(\frac{dR}{dz} \right) \quad (8)$$

where P_{VN} – the Vukalovich-Novikov equation of state for water steam; P_a – the air state equation for high pressures; σ – water surface tension, N/m .

The last equation for determining the bulk concentration of the disperse phase

$$\frac{dB_2}{dz} = \left(\frac{3 \cdot B_2}{R} \right) \cdot \left(\frac{dR}{dz} \right) - \left(\frac{B_2}{V_2} \right) \cdot \left(\frac{dV_2}{dz} \right) - \left(\frac{B_2}{S} \right) \cdot \left(\frac{dS}{dz} \right) \quad (9)$$

The calculations have been performed of the 9th order determinant values, formed on the basis of refined mathematical model of a bubble stream. It should be noted that out of 81 elements of our determinant, 53 elements were equal to zero based on the analytical form of the entire equation system. Analysis of the obtained numerical values of the determinant has shown that the velocities of the phases and the bulk concentration of the dispersed phase have the greatest influence on its value. It has also been established that at technically reasonable phase velocities in the HST nozzle, the critical flow regime does not occur in a bubble stream, i.e. the above determinant in the process of changing the stream parameters does not change its sign compared with the value in the nozzle inlet section. Thus, the bubble stream in the HST nozzle is in a subsonic flow regime. Based on this result, it was concluded that a two-phase stream is formed in the HST nozzle with a continuous increase in velocity.

2.2 Variants of principle schemes for energy complexes with the HST application

Consider the possible construction schemes, when the HST equipment is included into the composition of MPC. Figure 1 presents the principle scheme of the power facility [6], which consists of a gas-reciprocating module, on the shaft of which a module with a reaction-type hydro-steam turbine is set. This schematic solution allows the mine power complex to generate additional electrical energy through the useful heat recovery from the cooling system of a gas-reciprocating unit (GRU). The installation of a hydro-steam turbine, which realises the thermal energy of the hot water from the GRU cooling system into the kinematic chain of the power facility, ensures the direct transfer of the turbine mechanical energy to the twisting moment on the engine shaft.

In the preceding works [7, 8], a construction scheme for an improved reaction-type hydro-steam turbine has been studied, as well as a method for calculating the power and energy HST parameters.

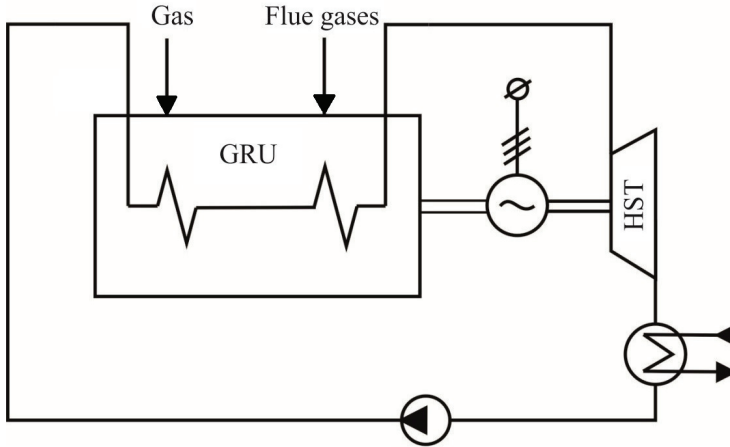


Fig. 1. The gas-reciprocating unit (GRU) with a hydro-steam turbine (HST) in the kinematic chain.

The shaft power of an improved hydro-steam turbine is determined as follows

$$P_{HST} = G \cdot 10^{-3} \cdot \left(v_v \cdot R_r \cdot \omega + F_{c2} \cdot R_{c2} \cdot \frac{\omega}{G} - 6.28 \cdot r_3 \cdot (R_r - r_3) \cdot \omega^2 - F_{c1} \cdot R_{c1} \cdot \frac{\omega}{G} - 2.22 \cdot r_2 \cdot (R_r - 2 \cdot r_3) \cdot \omega^2 - (R_r - 2 \cdot r_3 - r_2)^2 \cdot \omega^2 \right) - P_p - P_f \quad (10)$$

where G – the hot water consumption per turbine with a three-nozzle scheme, kg/s; v_v – the velocity of steam-water mixture efflux, m/s; R_r – the radius of the reactive force application, m; ω – the circumferential rotor velocity, 1/s; F_{c1} – the centrifugal force acting on the channel conjugation, N; R_{c1} – the radius of the F_{c1} force application, m; r_3 – the rounding radius of a curvilinear section, m; F_{c2} – the centrifugal force acting in the curvilinear section, N; R_{c2} – the radius of the F_{c2} force application, m; r_2 – the rounding radius of the conjugation section, m; P_p – the power consumed by pump, kW; P_f – the power expended on a disk friction in a steam-water mixture, kW.

The centrifugal forces F_{c1} and F_{c2} in accordance with [9] were determined by the expression

$$F_{ci} = 2 \cdot G \cdot v_i + (2 \cdot p_i - \Delta p_i) \cdot S_i, \text{ N}, \quad (11)$$

where v_i – the fluid velocity in the corresponding section of a channel, m/s; p_i – the pressure at the inlet of a section, Pa; Δp_i – the pressure losses due to friction and local resistance in the section, Pa; S_i – the cross sectional area of a section, m^2 . The radii of these forces application were determined according to the additive rule of parallel forces.

Having differentiated the expression for the resultant power (10) by ω , we obtain the expression for the optimal circumferential velocity at which the turbine power reaches its maximum value

$$\omega_{opt} = \frac{v_v \cdot R_r + F_{c2} \cdot R_{c2} / G - F_{c1} \cdot R_{c1} / G}{12.56 \cdot r_3 \cdot (R_r - r_3) + 4.44 \cdot r_2 \cdot (R_r - 2 \cdot r_3) + 2 \cdot (R_r - 2 \cdot r_3 - r_2)^2}, \text{ 1/s}. \quad (12)$$

The power losses due to friction during the rotor rotation in a chamber, filled with a steam-water mixture, can be determined by the known dependences given in [10, 11]. The internal power P_{HST} received on the turbine shaft is less than the resultant rotor power by the value of power expended on disk friction in the steam-water mixture P_f and for the

needs of the pump P_p [12, 13].

The electrical efficiency factor of the considered power facility is equal to

$$\eta_p = \frac{P_{GRU_e} + P_{HST} \cdot \eta_g}{P_{ch}}, \quad (13)$$

where P_{GRU_e} – the GRU electrical capacity, kW; η_g – the efficiency factor of a power generator; P_{ch} – chemical energy of fuel, kW.

The calculation of the energy indices of HST and the power facility under consideration has been performed according to dependences for the following parameters: the water temperature at the inlet of the turbine channel $t_1 = 110$ °C; the water temperature at the nozzle exit $t_2 = 46$ °C; the water velocity in a straight section and conjugation section $v_1 = 20$ m/s, the water velocity in the curvilinear section $v_2 = 100; 140$ m/s; the radius of the reactive force action $R_r = 0.8$ m; for rounding it is accepted $r_2 = r_3 = 0.1$ m; the thermal power $P_T = 3050$ kW; the GRU electrical capacity $P_{GRU_e} = 3035$ kW; chemical energy of fuel $P_{ch} = 7076$ kW; the efficiency factor of a power generator of the facility $\eta_g \approx 0.97$.

The construction turbine parameters were chosen based on the maximum power conditions, the absence of locking effects and cavitation in its channels. The calculation results are presented in Table 1.

Table 1. Thermodynamic and energy indices of HST and the power facility.

Parameters	Water velocity in the HST section	
	100 m/s	140 m/s
Mass vapor content	0.102	
Difference in water enthalpy at the inlet and outlet of the turbine channel, kJ/kg	268.75	
Hot water consumption per turbine, kg/s	11.35	
Velocity coefficient	0.63	
Jet velocity, m/s	138	
Optimal circumferential rotor velocity, 1/s	175	253
Power consumed by pump, kW	2.8	4.2
Power consumed for the rotor friction, kW	22	67
Power to the turbine shaft, kW	279	441
Electrical efficiency factor of the power facility	0.467	0.489

The Table 1 analysis shows that the installation of a HST into the kinematic chain of a power facility at a constant gas consumption raises the electrical capacity by generating the additional electrical energy when heat is recovered from the cooling system of a gas-reciprocating unit. This makes possible to increase electrical energy production by 6 % without additional fuel consumption and, consequently, increase the efficiency factor of the power facility.

In Figure 2, there is a principle scheme with the reaction-type HST and the back pressure turbine (BPT) placement on the same shaft with the GRU [14]. In the considered schematic solution, the excess heat is used, both from the GRU and from the BPT. The advantage of this schematic solution is that such basic energy modules as HST, GRU and BPT are located on a single shaft with one power generator. This increases the reliability of the cogeneration scheme operation and reduces the capital costs.

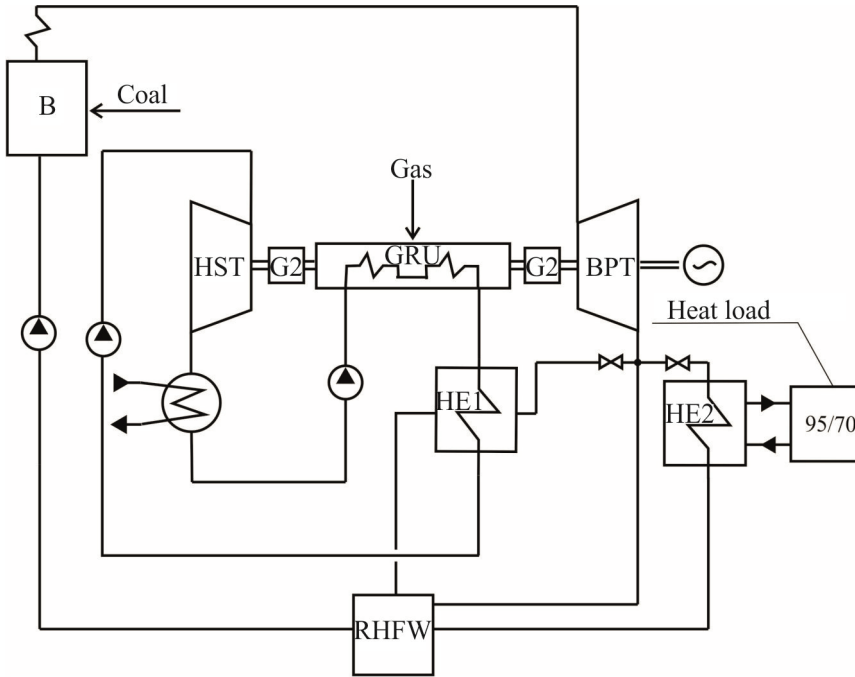


Fig. 2. The cogeneration scheme with the back pressure turbine and hydro-steam turbine placement on the same shaft with the gas-reciprocating unit: B – boiler unit; G1, G2 – gear units; HE1, HE2 – heat exchangers; RHF – regenerative heating of feedwater.

When the mine power complex is working according to the scheme shown in Figure 1, only additional electrical energy is generated for the power plant, but when working according to the scheme with the BPT and HST modules located on the same shaft with the GRU-module, both electrical and thermal energy is additionally generated.

The calculation of the cogeneration scheme efficiency, presented in Figure 2, was to determine the costs and state parameters of the working body, and on their basis to determine the scheme energy indices [14].

The heat exchangers, in which the heating medium is an over-heated steam and heated water, are described by the thermal-balance equation

$$G_s \cdot (i_{s1} - i_{s2}) \cdot \eta^{HE} = G_w \cdot c_w \cdot (t_{w2} - t_{w1}), \quad (14)$$

where G_s – the steam flow consumption, kg/s; i_{s1} , i_{s2} – the steam enthalpy at the inlet and outlet of the apparatus, J/kg; η^{HE} – the efficiency factor of heat exchanger; G_w – the consumption of water being heated, kg/s; c_w – the specific heat of water, J/(kg·K); t_{w1} , t_{w2} – the initial and final water temperature, K.

The fuel consumption for a steam boiler was determined by the following equation

$$B = \frac{G_B \cdot Q_B}{\eta^B \cdot Q_1^c}, \text{ kg/s}, \quad (15)$$

where G_B – the boiler unit performance, kg/s; Q_B – the amount of heat obtained in the boiler with feedwater when it is converted into steam, J/kg; η^B – the efficiency factor of the boiler unit; Q_1^c – the lower heat value of fuel, J/kg. The lower heat value of fuel equivalent is used in calculations and is equal to $Q_1^c = 29308$ kJ/kg.

For boiler units in which the over-heated steam is produced, the value Q_B is expressed as

$$Q_B = (i_s - i_{f.w}) + \frac{F}{100}(i' - i_{f.w}), \text{ J/kg}, \quad (16)$$

where $i_s, i_{f.w}, i'$ – the enthalpy of the over-heated steam, feedwater and boiler water, respectively (assumed as equal to water enthalpy at the temperature of boiling), J/kg; F – the continuous blowdown fraction, % (is 2 – 5 % of productivity boiler).

For steam and hydro-steam turbines, the thermal (internal) power was determined by the expression

$$N^t = G \cdot (i_1 - i_2) = G \cdot (i_1 - i_2) \cdot \eta^s, \text{ W}, \quad (17)$$

where G – the working body consumption through the turbine, kg/s; i_1, i_2 – the enthalpy of the working body at the turbine inlet and outlet, J/kg; i_2 – the enthalpy of the working body at the turbine outlet during isentropic expansion, J/kg; η^s – the turbine isentropic efficiency factor.

With account of the expression (17), the electrical capacity of a turbine generator is equal to

$$N^e = N^t \eta^m \eta^g, \text{ W}, \quad (18)$$

where η^m and η^g – the mechanical efficiency factor of the turbine and the efficiency factor of the power generator, respectively.

The efficiency factors of electrical energy generation for the considered schemes were determined as follows

$$\eta^e = \frac{N_{GRU}^e + N_{HST}^e}{N_{GRU}^s}, \quad (19)$$

$$\eta^e = \frac{N_{GRU}^e + N_{BPT}^e + N_{HST}^e}{N_{GRU}^s + N_B^s}, \quad (20)$$

where $N_{GRU}^e, N_{BPT}^e, N_{HST}^e$ – the electrical capacity of the gas-reciprocating unit, the back pressure turbine and the hydro-steam turbine, respectively, W; N_{GRU}^s, N_B^s – the fuel energy supplied to the GRU and steam boiler, respectively, W.

The fuel energy supplied to the steam boiler is equal to

$$N_B^s = B \cdot Q_1^c, \text{ W}. \quad (21)$$

The efficiency factor of thermal energy generation for the scheme presented in Figure 2 was determined as follows

$$\eta^t = \frac{N_{TL}}{N_{GRU}^s + N_B^s}, \quad (22)$$

where N_{TL} – the thermal load power, W.

The following thermal technological equipment is proposed as the initial calculation data. Three steam boilers with the following steam parameters: total steam-production

capacity 26.5 t/hr; pressure 2.35 MPa; temperature – 380 °C. The feedwater temperature before the boiler is 105 °C. The back pressure turbine with the steam parameters: rated consumption 27.63 t/hr; pressure 2.35 MPa; temperature 380 °C; pressure behind the turbine 0.4 MPa; temperature behind the turbine 207 °C. The rated electrical capacity of the turbine is equal to 2500 kW. The gas-reciprocating unit: electrical capacity is equal to 3035 kW, and the supplied fuel energy is – 7076 kW. Three variants of the GRU thermal circuit performance were considered: with a cooling water temperature graphs 60/80 °C, 70/90 °C, 70/110 °C and recoverable thermal power 3187 kW, 3021 kW, 2921 kW, respectively. The HST parameters were selected based on the maximum power conditions, the absence of locking effects and cavitation in its channels.

In Table 2 a comparative analysis is represented of the energy efficiency of the considered principle schemes for the mine power complex (Figs. 1, 2).

Table 2. The energy efficiency of the cogeneration schemes for the mine power complex.

Parameters	Temperature graph of the GRU		
	60/80 °C	70/90 °C	70/110 °C
Fuel energy supplied to GRU, kW	7076		
Electrical capacity of GRU, kW	3035		
Thermal power of GRU, kW	3187	3021	2921
Cooling water consumption of GRU, kg/s	38.11	36.07	17.38
The power facility scheme (Fig. 1)			
Electrical capacity of HST, kW	98.6	91.3	141.9
Efficiency factor on electrical energy generation, %	44	44	45
Efficiency factor on thermal energy generation, %	-	-	-
Scheme with the BPT and HST placement on the same shaft with the GRU (Fig. 2)			
Fuel energy supplied to a boiler, kW	19783		
Electrical capacity of BPT, kW	2220		
Electrical capacity of HST, kW	1127.7	813.0	391.9
Thermal load power, kW	7163	9195	14617
Efficiency factor on electrical energy generation, %	24	23	21
Efficiency factor on thermal energy generation, %	27	34	54
Total efficiency factor on thermal energy and electrical energy generation, η , %	51	57	75

The Table 2 analysis shows that the installation of a HST into the kinematic chain of a power facility at a constant gas consumption raises the electrical capacity delivered to the network by the generator. As a result, the efficiency factor of the mine power complex in terms of electrical energy generation increases and reaches a maximum value of 45 %, when performing the thermal circuit of the GRU with the highest temperature of 70/110 °C. When the mine power complex is working based on the principle scheme (Fig. 2) with the BPT and HST modules placement on the same shaft with the GRU-module, both electrical and thermal energy is generated.

Conclusions

1. One of the prospective directions for using the energy of hot water produced by mine power complexes is the application of hydro-steam turbines in schematic solutions, which will make possible to respond flexible to fluctuations in thermal load and ensure high efficiency of the power complex performance as a whole.
2. In order to obtain a high efficiency factor of thermal energy conversion into

mechanical energy in the HST-module, the configuration of the turbine nozzle should be determined on the basis of the following principle. The bubble stream in the turbine nozzle should be organized in subsonic flow regime up to the inversion point of the two-phase stream, so that the parameters of the droplet-laden stream being formed are already in subsonic flow regime at the very inversion point.

3. The performed analysis has shown that with appropriate schematic and construction solutions for hot water recovery from energy modules in mine power complexes, the total efficiency factor for thermal and electrical energy generation is changed from 51 % to 75 % depending on the temperature graph of the GRU cooling water. Therefore, the choice of variant of the thermal circuit performance with the GRU depends on the needs of the mine in thermal energy. This provides year-round production of highly liquid product (electrical energy).

4. Based on the research results performed in the paper, a general conclusion can be made about the effectiveness of using the considered principle schemes for achieving the maximum technical and economic performance of the mine power complex operation.

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