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Theoretical Substantiation of “The Variable Phase Turbine – Downhole Recuperator” for Underground Coal Gasification

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Abstract

The article deals with the problem of increasing the efficiency and competitiveness of the underground coal gasification technology at the expense of utilizing the thermal energy of the underground gas generator, which is lost in the bowels. The authors have been analyzing the suggested by them system "variable phase turbine (VPT) – well recuperator". It could be included into the thermal circuit of the underground coal gasification and be capable of utilizing up to 30% of thermal energy losses of the gas generator, providing high efficiency of electricity generation from a low-grade heat carrier. The idea of using a powerful VPT (up to 100 kW) and corresponding recuperators for a downhole gas generator and the formation of a coal gasification zone along a single well has been a technical innovation which is to be substantiated. To realize this task the authors have developed a certain algorithm for the thermal calculation of convective heat transfer from the "well-gas generator" to the recuperator with a water heat carrier Thus the quantitative assessment of the thermal efficiency which could be obtained from a single heat removal well, has been carried out. The technical parameters of a high power VPT within the thermal circuit of underground coal gasification have been substantiated and could have been realized. Alongside, it become possible to do the calculations for the flow characteristics, the traction jet force, the consumed heat power during the turbine operation and at last for the efficiency and carnotization factor. Such calculations have been successfully carried out.

Keywords: *Underground coal gasification; Gas-generator; Heat utilization; Recuperator; Superheated water; Thermal flow; Variable phase turbine.*

1. Introduction

Safe and environmentally efficient coal technologies can provide a significant part of energy balance of the world economy, especially in countries with a developed coal industry and significant coal reserves, which is very relevant, in particular, for Ukraine and China. However, environmental and safety problems of coal mining and processing require widespread use of underground thermochemical processing of coal seams using borehole geotechnologies, especially in difficult operating conditions of coal deposits. Downhole geotechnology is an example of "clean coal technologies", which at the same time provide a high level of safety in mining operations, since they are "unmanned technologies" [1-6].

Among the main reasons that reduce the efficiency and competitiveness of the underground coal gasification technology, it is necessary to name significant unproductive losses of thermal energy within the underground gas generator (up to 30-50%), which remain unclaimed in the bowels [7-9]. One of the most effective methods for extracting and utilizing thermal energy is the use of heat-removing wells at the bottom of the gasified coal seam, and the wells equipped with a pipe recuperator with a low-grade liquid heat carrier that feeds a variable phase turbine (Fig. 1 [10]).

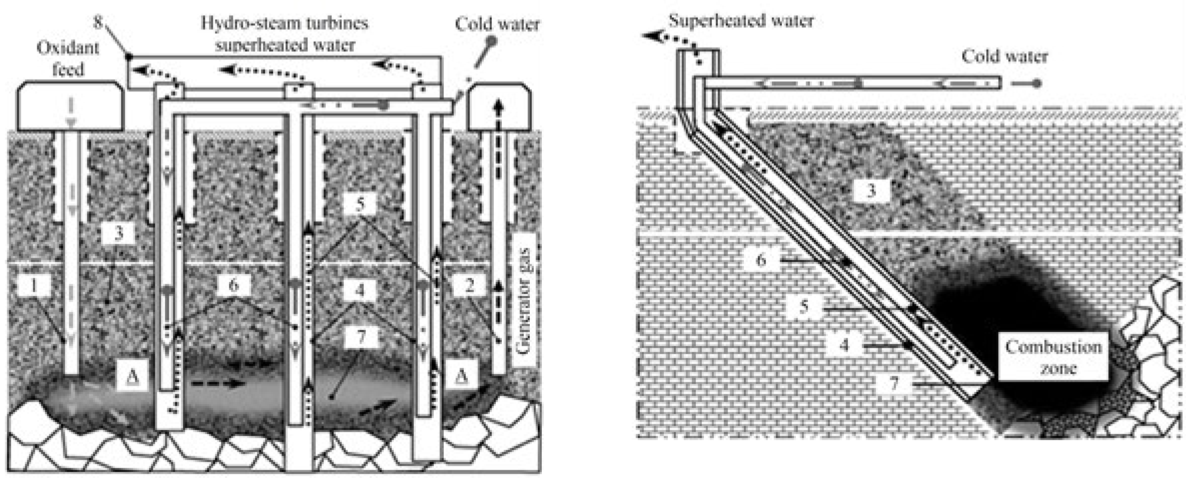

Figure 1. - General scheme of heat energy utilization using recuperators with a liquid heat carrier (a - plan, b - section along the well): 1, 2 - air supply and gas outlet wells, 3 - coal seam, 4 - heat removal wells, 5 - recuperator (closed on one side pipe), 6 - cold water feed sleeve, 7 - formation combustion zone (high-temperature heat exchange), 8 – VPT (variable phase turbine)

The scientific and technical achievements of the last two decades in the field of geothermal energy technologies have opened up fundamentally new opportunities for using a coolant with a low thermal potential (T=110-250°C) for efficient generation of electricity on VPT. The VPT is a device that converts the enthalpy of a hot water flow in a Laval nozzle into the kinetic energy of a two-phase steam-water mixture flow, which is generated at the nozzle outlet. The reactive force of the steam-water jet serves to rotate the turbine rotor, mechanically connected to an electric generator, which produces electricity [11-12].

In the studies of ZAO Turbokon (Kaluga, Russia), the researchers consider the Heron's wheel in which there is no fixed nozzle, and Laval nozzles are installed along the perimeter of the impeller, tangentially to the wheel, in which boils a two-phase flow with high humidity The jet force of the flow provides the driving moment of the impeller. However, to date, only a turbine with a maximum power of 10 kW has been tested at the Kaluga "Turbokon". The efficiency of such a device is determined by the internal efficiency and ranges from 10 to 20% [13-14].

Saint Petersburg State Technical University has developed axial and radial single-stage turbines operating on boiling water. To accelerate the two-phase flow, Laval nozzles with steam-generating grids installed at the inlet were used. The efficiency of experimental models of such VPT per unit power from 1 to 20 kW was at the level of 30...46% [11,15].

The VPT technologies developed by Energent Corporation (USA) for single-stage axial turbines with nozzles and steam generator grids (meant for air conditioning systems), have achieved much higher efficiency. These turbines had an efficiency of about 70...80%, and their use in industry has become widespread. The Energent Corporation has developed and successfully tested a 100 kW VPT [16-17].

At the M. Polyakov Institute of Geotechnical Mechanics of the National Academy of Sciences of Ukraine (Dnipro), a ratio was obtained for calculating the torque of the moving jet and the efficiency of the VPT unit depending on the four main parameters of its operation (mass flow rate of the working medium, thermal velocity due to the temperatures at the nozzle, the linear speed of rotation of the turbine wheel, the radius of the turbine wheel) and an analysis of their interrelationships was carried out. For VPT typical operating parameters, the value of the thermal efficiency coefficient is 46% of the maximum possible for the Carnot cycle. The works [18-19] substantiated the expediency of using the VPT technology as part of the energy complex of mines, including the utilization of mine methane and excess heat from mine energy facilities.

For the first time, papers [10, 20-22] proposed incorporate hydro-steam turbines into a thermal system of underground coal gasification. In addition, they substantiated certain technological parameters of the new technique to utilize thermal power of an underground gas generator. Nevertheless, the developed utilization techniques (Fig. 1) were adapted for the use of low-capacity VPTs (i.e. 10 kW) while predicting high-temperature heat-exchange zone (T = 300-1000°C) limited by several meters. The creation of a 100 kW VPT in the USA, have offered opportunities to apply the recuperators, developed by the authors, in terms of gas generator wells [10], where the gasification process travels along a well and high temperature heat-exchange zone may achieve dozens of meters (Fig.2).

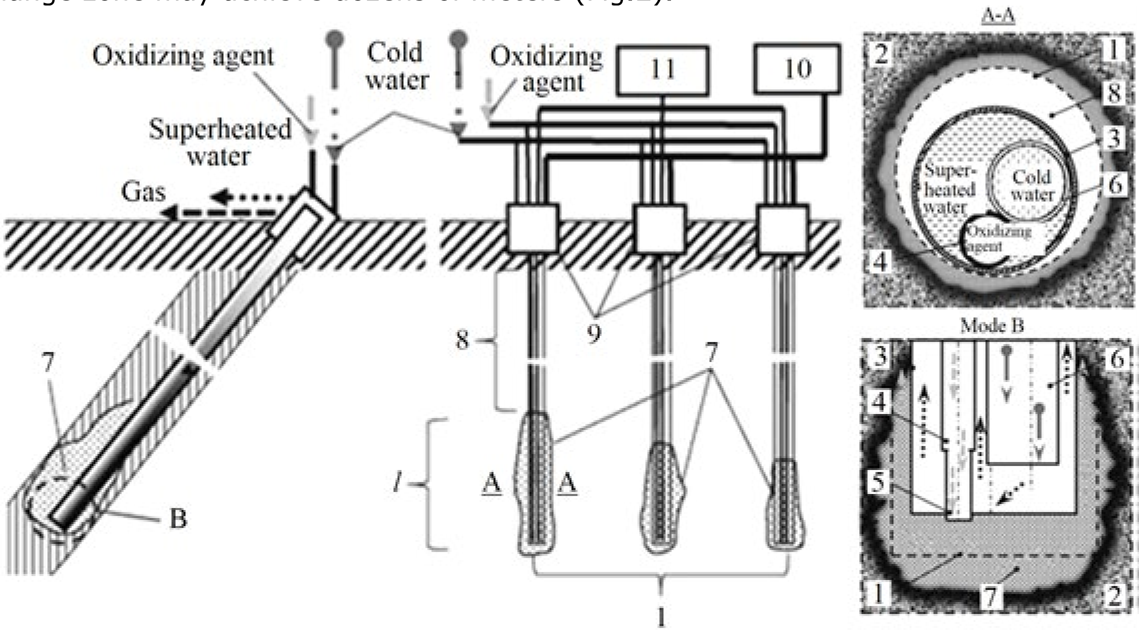

Figure 2. General scheme of heat energy utilization using recuperators in the "well-gas generator" system: 1 - well system for simultaneous heat and gas removal, 2 - coal seam, 3 - recuperator, 4 - oxidizer (air) feed sleeve, 5 - a branch pipe, 6 - heat carrier (water) feed sleeve, 7 - a formation combustion zone (high-temperature heat exchange), 8 - a zone for moving generator gas along a well, 9 - a conductor, 10 - gas turbines, 11 – VPT (variable phase turbine)

The need to study such heat and power complexes actualizes the purpose of our work - the theoretical substantiation of the system "VPT - recuperator" to assess the potential of the underground borehole gas generator for thermochemical processing of coal seams. The objectives of the work are: substantiation of technical parameters for a powerful VPT (in particular, the calculation of the required flow characteristics, the thrust reactive force, the useful power, the consumed thermal power during turbine operation, the degree of carnotization), the development of an algorithm for the thermal calculation of convective heat transfer from the "well-gas generator" to a recuperator with a water heat carrier, obtaining and analyzing the results.

2. Methods

To identify VPT efficiency, the researchers used the method of calculating the hydraulic turbine wheel for the thrust reactive moment [23], which made it possible to obtain the basic technological parameters for the “VPT - borehole recuperator” system. They have also developed the algorithm for thermal calculation of convective heat transfer from a "gas generator well" to a recuperator with a water heat carrier. The latter uses undeveloped and advanced boiling regimes (also the transition regime), taking into account the hydraulic resistance in the pipes. Certain calculation coefficients were taken from similar conditions of high temperature heat exchange in nuclear power plants.

3. Experimental thermal analysis algorithm and results

3.1. VPT characteristics calculations

These calculations [23] aim to estimate the efficiency of the VPT using the data on the reactive torque of the HST wheel.

Initial Data. Let us set the initial parameters for a typical reactive VPT [23-24]:

T_{IN} = 383 K, T_O = 303 K, ω_T =314 s^{-1}, R_T = 1 m, d_m = 10 mm, $P_{IN}(T_{IN})$ = 3 bar, where: $P_{IN}(T_{IN})$ – supply pressure of heated water to the nozzle by the feed pump; $P_m = P_S(T_{IN})$ – pressure of heated water in min nozzle cross section, beginning of boiling flow

While:

$$P_S(T_{IN}) < P_{IN}(T_{IN}) \tag{1}$$

Accordingly, $P_O = P_S(T_O) = P_V$ – pressure of the two-phase steam-water mixture in the nozzle outlet section:

$$P_S(T_O) < P_S(T_{IN}) \tag{2}$$

The flow of the two-phase mixture, leaving the nozzles installed on the VPT wheel, creates a traction reactive force, which drives the shaft of the generator into rotation.

Euler's Equation for One-dimensional Flow (for a given device).

$$F = G(w_o - w_m), \tag{3}$$

where: F - the resultant force projection on the nozzle axis of symmetry; G - the working medium mass flow rate; w_o - the two-phase medium velocity at the nozzle outlet; w_m - the heated water velocity in min nozzle cross section ahead of boiling front.

The Resultant Force. Taking into account (2) and (3), the resultant force is the sum of all forces projections on the nozzle axis of symmetry:

$$p_0(S_0 - S_m) + p_S((T_{in})S_m - p_0 S_0 - F_{mj} = G(w_0 - w_m), \tag{4}$$

where: S_0 - the nozzle outlet cross sectional area; S_m - min nozzle cross-sectional area; F_{mj} - the traction jet force.

The Traction Jet Force.

$$F_{MJ} = G(w_0 - w_m) - (p_s(T_{in}) - p_0)S_m. \tag{5}$$

The second term of (5) actually expresses the drag force on the nozzle design elements protruding beyond the VPT wheel. The value of the actual traction jet force is given by the first term in (5), for which it is necessary to calculate w_o and w_m.

Expressions for w_o and w_m. In [23], an expression was obtained for w_o:

$$w_o^2 = 2\left[h^{'}(T_{in}) - h^{'}(T_o) - c_p^{'}T_o\, ln\left(\frac{T_{in}}{T_o}\right)\right] + w_m^2, \tag{6}$$

where: $h^{'}(T_{in})$; $h'(T_o)$ - the liquid phase specific enthalpy in the nozzle min and outlet cross sections; $c_p^{'} \approx c_s^{'}$ - isobaric heat capacity of water at saturation.

Bernoulli's Equation. The value of the water velocity in the nozzle min cross section w_{min} is determined by the feed pump pressure p_{in}, as well as the pressure created by the field of centrifugal forces during the rotation of the turbine wheel. To determine this effect on the flow rate velocityw_m, we compose the Bernoulli's equation between the nozzle confuser inlet cross section and its min cross section just before the boiling front:

$$p_{in} + \frac{\rho^{'}(T_{in})w_{in}^2}{2} + \frac{\rho^{'}(T_{in})\omega_T^2 R_T^2}{2} = p_s(T_{in}) + \frac{\rho^{'}(T_{in})w_m^2}{2}, \tag{7}$$

where: $\rho^{'}(T_{in})$ - the water density at temperature T_{in}; ω_T - the VPT wheel angular speed of rotation; R_T - distance from the axis of rotation of the turbine wheel to the axis of the nozzle; w_{in} - water velocity in the nozzle confuser inlet cross section; determined by the pump capacity Q_{pump} (m^3/s) required for pumping a given flow rate of the working medium and the inlet cross section diameter of the confuser.

Water Velocity in the Nozzle Min Cross Section. From (7) we obtain an expression for w_m^2:

$$w_m^2 = \frac{2(p_{in} - p_s(T_{in}))}{\rho^{'}(T_{in})} + \omega_T^2 R2_T + w_{in}^2. \tag{8}$$

Expression for the Traction Jet Force. Taking into account (6) and (8), the relation for the traction jet force takes the form:

$$F_{MJ}(T_{in},T_o,\omega_T,R_T)=G\left\{\begin{matrix}\left[2\left(h'(T_{in})-h'(T_o)-c_p'T_o\ln\left(\frac{T_{in}}{To}\right)\right)+\frac{2(p_{in}-p_s(T_{in}))}{\rho'(T_{in})}+\omega_T^2R_T^2+w_{in}^2\right]^{1/2}-\\-\left[\frac{2(p_{in}-p_s(T_{in}))}{\rho'(T_{in})}+\omega_T^2R_T^2+w_{in}^2\right]^{1/2}\end{matrix}\right\} \quad (9)$$

Taking into account the parameters adopted for a typical VPT, the value of the complex $\frac{2(p_{in}-p_s(T_{in}))}{\rho'(T_{in})}+w_{in}^2$ can be neglected in comparison with $(\omega_T R\ \ )^2$, and expression (9) takes the form:

$$F_{MJ}=G\left(\sqrt{w_T^2+w_m^2}-w_R\right)-(p_s(T_{in})-p_0)S_m, \quad (10)$$

where $w_T^2=2\left[h'(T_{in})-h'(T_o)-c_p'T_o\ln\left(\frac{T_{in}}{T_o}\right)\right]$ is the square of the thermal velocity, which is due to the temperature difference at the nozzle; $w_R=\omega_T R_m$ - the linear speed of rotation of the turbine wheel on the radius R_m.

It should be noted that expression (10) for the traction jet force, in contrast to the similar result [23], is written taking into account the frontal resistance force on the nozzle structure elements protruding beyond the VPT wheel, as noted in expression (5). Then the expression for the traction jet power will be:

$$N_{MJ}=F_{Mj}w_R=\left[G\left(\sqrt{w_T^2+w_m^2}-w_R\right)-(p_s(T_{in})-p_o)S_m\right]w_R. \quad (11)$$

Accordingly, without taking into account the mentioned frontal resistance force [23]:

$$N_{MJ}=G\left(\sqrt{w_T^2+w_m^2}-w_R\right)w_R. \quad (12)$$

Expressions (11), (12) determine the useful power of the VPT.

The Consumed Thermal Power during the Operation.

$$N_T=Gc_p'(T_{in}-T_o). \quad (13)$$

The Thermal Efficiency (without taking into account the mentioned frontal resistance force [23]).

$$\eta_t=\frac{N_{MJ}}{N_T}=\frac{\left(\sqrt{w_T^2+w_R^2}-w_R\right)w_R}{c_p'\left(T_{in}-T_o\right)}. \quad (14)$$

The results of calculating the traction jet torque and the efficiency of VPT with typical parameters are shown in Table 1.

Table 1. Results of calculating the VPT characteristics

VPT Traction jet power	Nmj [kW]	97,8
VPT Thermal power	N_T [kW]	1065,6
VPT Thermal efficiency	η_t, %	9,2
Carnot efficiency	η_c, %	20,9
Carnotization factor	η_t/η_c, %	43,9

3.2. Calculation of convective heat transfer from the "Well – Gas Generator" to the water recuperator

If the ratio of the wall temperature and the average flow temperature meet the conditions:

$$t_W-t_S>0, \quad (15)$$

$$t_S-\bar{t}_f>0, \quad (16)$$

then, under such conditions, surface boiling occurs in the boundary layer near the surface – boiling of a subcooled liquid, or boiling with subcooling [25-27].

Heat Transfer during Boiling of Subcooled Liquid [25]. As long as the wall temperature remains below the saturation temperature, heat transfer is determined by the laws of single-phase heat transfer, and the heat flux density is proportional to the excess temperature $t_W-\bar{t}_f$, where t_W is the wall temperature and $\bar{t}_f$ is the average flow temperature (Fig.3 [25]).

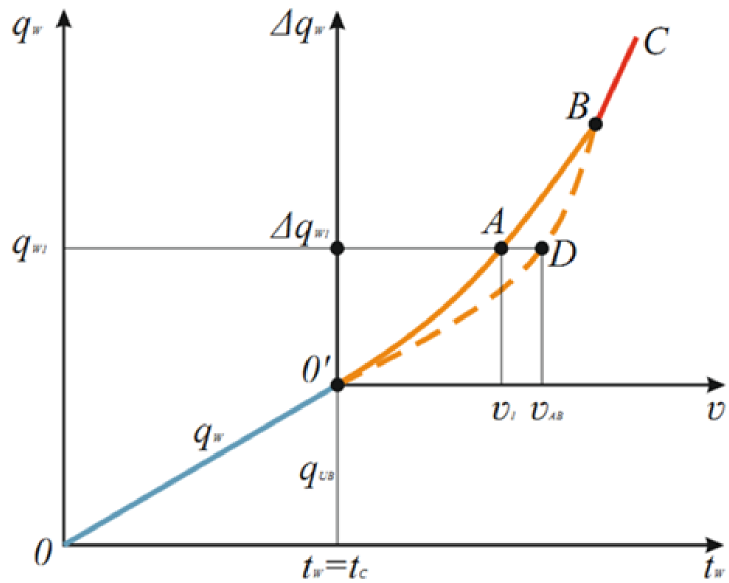

Figure 3. Dependence of the heat flux density on the wall temperature during boiling of a liquid subcooled to saturation temperature: *OO′* - single-phase flow, *O′AB* - undeveloped boiling, BC - advanced boiling, *O′DC* - calculation according to equation (22), ϑ_1 - calculation by the formula (23)

After the wall temperature exceeds the saturation temperature, a further increase in the heat flux density will be associated with boiling in the near-wall layer, i.e., it will depend on the temperature difference $t_W - t_S$, where $t_W - t_S$ is the saturation temperature. Therefore, to analyze heat transfer in the subcooled boiling region, it is convenient to switch to a new coordinate system $\Delta q_W - \vartheta$ [28] with the origin at point *O′*, where:

$$\vartheta = t_W - t_S, \tag{17}$$

$$q_W = q_{UB} + \Delta q_W, \tag{18}$$

Single-Phase Flow Regime (at small ϑ).

$$q_W = \alpha_{CONV}\left(t_W - \bar{t}_f\right) \tag{19}$$

Undeveloped Boiling Regime (at low ϑ values). Heat transfer is mainly determined by the laws of single-phase heat transfer:

$$q_{UB} = \alpha_{CONV}\left(t_S - \bar{t}_f\right) \tag{20}$$

Advanced Boiling Regime (at high ϑ values). In the case of advanced boiling of a subcooled liquid, the heat transfer coefficient can be calculated using the relations valid for the boiling of a saturated liquid [26-27], substituting in them Δq_W instead of q_W:

$$\Delta q_W = \alpha_{BOIL}(t_W - t_S), \tag{21}$$

or use empirical dependence [28]:

$$\Delta t_{AB} = 7T_{CR}^{0.82} p_{CR}^{-0.36} M^{0.18} \Delta q_W^{0.36} \exp\left(-5.6\frac{T_S}{T_{CR}}\right), \tag{22}$$

where: $\vartheta_{AB} = t_W - t_S$ - the wall overheating in the case of advanced boiling; M - molecular weight of the liquid.

For the undeveloped boiling regime, the temperature of the heating surface can be determined using the interpolation formula [28, 26]:

$$\frac{\vartheta}{\vartheta_{AB}} = \left[1 + \left(\frac{\vartheta_{AB}}{\Delta q_W/\alpha_{CONV}}\right)^{3/2}\right]^{-2/3}, \tag{23}$$

where α_{CONV} calculated from the circulation rate.

This dependence is valid in the entire range of ϑ variation from 0 to ϑ_{CR}, for undeveloped and advanced boiling, as well as in the transition regime [28].

The last two formulas are confirmed by experimental data obtained when boiling water and ethyl alcohol in the pressure range of 1.5÷90 bar, underheating 1÷260°C, heat flux densities $0.23\cdot10^6 \div 24\cdot10^6$ W/m^2 and circulation rates 1÷23 m/s.

The initial data for the calculation are given in Table 2.

Table 2. Data for calculation

Parameters	Values	
Feed sleeve pipe	d_i [m]	d_0 [m]
	0,09	0,1
Recuperator (pipe stand)	D_i [m]	D_0 [m]
	0,15	0,166
Core combustion temperature	t [°C]	T [K]
	1000	1273
Average temperature in the heat exchange zone	t [°C]	T [K]
	600	873
Pipe stand wall temperature	t [°C]	T [K]
	700	973
Recuperator water temperature (inlet and outlet)	t′ [°C]	t′′ [°C]
	30	110
Recuperator water flow rate	m_T [kg/s]	V_T [m^3/h]
	3,17	12
Recuperator water density	ρ [kg/m^3]	
	951	
Recuperator water velocity	w [m/s]	
	0,189	
Feed sleeve water velocity	w [m/s]	
	0,424	

3.2.1. Calculation algorithm

Flow regime: $Re = w_f d/\nu$

Convective Heat Transfer Coefficient α_{CONV} [26]. Turbulent flow in a pipe, $Pr \geq 0{,}5$, $Re \geq 10^4$ (Petukhov-Kirillov formula [25-26, 29])

$$Nu = \frac{\xi}{8}\frac{RePr}{1+\frac{900}{Re\left(Pr^{2/3}-1\right)\sqrt{\frac{\xi}{8}}}}, \tag{24}$$

where ξ - hydraulic resistance coefficient [26]:

$\xi = 0.316/Re^{1/4} \quad (Re \approx 10^4)$,

$\xi = (1.82\, lgRe - 1.64)^{-2}$ $(5\cdot 10^4 < \mathrm{Re} < 10^7)$.

The effect of the liquid thermophysical properties on temperature in practical calculations is taken into account by introducing corrections. For droplet liquids, the Mikheev correction [26]: $\varepsilon_t = \left(Pr_f \quad /Pr_w \quad\right)^{0.11}$or Petukhov correction [26]: $\varepsilon_t = \left(\mu_f/\mu_w\right)^{0.11}$.

The change in viscosity also affects the hydraulic resistance. To account for this effect, B.S. Petukhov correction [26]: $\varepsilon_\xi = \left(\mu_f/\mu_w\right)^{-0.17}$.

Thus, when calculating heat transfer and hydraulic resistance in pipes [26]: $\alpha = \alpha_0\varepsilon_t$; $\xi = \xi_0\varepsilon_\xi$.

Calculation $\boldsymbol{\Delta q_W}$ (for H_2O) [26]

$$\Delta t = t_W - t_S = 7T_{CR}^{0.82}p_{CR}^{-0.36}M^{0.18}\Delta q_W^{0.35}\, exp\left(-5.6\frac{T_S}{T_{CR}}\right) \tag{25}$$

Calculation $\boldsymbol{\alpha_{BOIL}}$ [30]

$$\alpha_{BOIL} = 3.1p^{0.25}\Delta q_W^{2/3} \tag{26}$$

Comparison $\boldsymbol{\alpha_{CONV}}$ and $\boldsymbol{\alpha_{BOIL}}$

Determination of the Heat Flux Density $\boldsymbol{q_W}$**:**

$$q_W = \Delta q_W + q_{UB} \tag{27}$$

$$\Delta q_W = \alpha_{BOIL}(t_W - t_S) \tag{28}$$

$$q_{UB} = \alpha_{CONV}\left(t_S - \bar{t}_f\right) \tag{29}$$

3.2.2. Results of calculation for Well D = 0.15 m, Vт = 12 m³/h and HST N = 100 kW

Table 3. Results of calculation

Heat flux density	q_W [kW/m^2]	971,8
Linear heat flux density	q_L [kW/m]	458,0
Pipe section	L [m]	2,3
VPT thermal power	N_T [kW]	1065,6
VPT traction jet power	Nmj [kW]	97,8
VPT shaft work	E [MWh]	351,9

4. Conclusions

The results of analytical studies indicate that the proposed method of thermal energy utilization is technically justified and acceptable for the interaction of a downhole recuperator with powerful VPT in the thermal circuit of underground coal gasification. The developed algorithm and the obtained results of thermal calculation make it possible to estimate the heat and power effect that can be obtained from a single heat removal well. At the same time, the jet power of VPT was 98 kW, with a consumed thermal power of 1066 kW. The shaft work was 352 MWh, and the linear heat flux was 458 kW/m.

As regards some particular problems our attention was attracted to the minimum temperature of superheated water at the outlet from the recuperator (110°C) and it has been additionally analyzed. The obtained value of the VPT carnotization factor (43.9) indicates the possibility of a significant increase in efficiency with an increase in the temperature of the superheated water. Since the section of high-temperature heat exchange in the well-gas generator can be 20 meters or more, the temperature of the liquid at the outlet can be maintained at a level of up to 250°C (the maximum parameter for the VPT under consideration), even with a significant increase in the liquid flow rate (due to the increase in the diameter of the recuperator and the increase in the superheated water speed of movement). Thus, the thermal energy of the underground borehole gas generator can ensure the efficient operation of the VPT with a capacity of 100 kW, and the system of such wells creates the basis for the operation of a powerful thermal power plant in addition to the effect of the generator gas obtained from underground coal gasification.

Symbols

T_{IN}	*initial temperature of the heated water;*
T_O	*temperature at the nozzle outlet;*
ω_T	*turbine angular velocity;*
R_T	*turbine wheel radius;*
d_m	*min nozzle cross sectional diameter;*
$P_{IN}(T_{IN})$	*supply pressure of heated water to the nozzle by the feed pump;*
$P_m = P_S(T_{IN})$	*pressure of heated water in min nozzle cross section, beginning of boiling flow;*
F	*the resultant force projection on the nozzle axis of symmetry;*
G	*the working medium mass flow rate;*
w_o	*the two-phase medium velocity at the nozzle outlet;*
w_m	*the heated water velocity in min nozzle cross section ahead of boiling front;*
S_0	*the nozzle outlet cross sectional area;*
S_m	*min nozzle cross-sectional area;*
F_{mj}	*the traction jet force;*
$h'(T_{in}),\ h'(T_o)$	*the liquid phase specific enthalpy in the nozzle min and outlet cross sections;*
$c_p' \approx c_s'$	*isobaric heat capacity of water at saturation;*
$\rho'(T_{in})$	*the water density at temperature T_{in};*
ω_T	*the VPT wheel angular speed of rotation;*
R_T	*distance from the axis of rotation of the turbine wheel to the axis of the nozzle;*
w_{in}	*water velocity in the nozzle confuser inlet cross section, determined by the pump capacity Q_{pump} (m^3/s) required for pumping a given flow rate of the working medium and the inlet cross section diameter of the confuser.*

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