THE USE OF NATURAL HEAT OF DRY HOT ROCKS OF THE EARTH WITH A GAS COOLANT

by

Sergey V. ALEKSEENKO[a] and Lev A. OGURECHNIKOV

Kutateladze Institute of Thermophysics SB RAS, Novosibirsk, Russia

Abstract

The energy options for utilizing the deep heat of the Earth using n-pentane as a working fluid for generating electricity in the organic Rankine cycle (ORC) are analyzed. The relationships between the initial and final parameters of the cycle are determined. The ratio of the average diameters of turbine stages to the working blades height along the flow part is estimated. A satisfactory profile of the flow part of an axial single-flow multistage jet turbine with subsonic flow rates of the working gas has been obtained.

Keywords: Rankine cycle, axial-type turbine, thermodynamics, diverging angle of the turbine stage

1. Introduction

The transition to renewable energy sources is a promising direction of energy development. The extraction of heat from dry hot rocks of the Earth and its transportation to the surface for subsequent conversion into electricity and heat of increased potential is a serious scientific and technical problem.

When utilizing underground energy, the efficient heat removal from the heat transfer surface of a heat source at a great depth should be organized. It is necessary to circulate the working fluid with the removal of thermal energy with the specified parameters from hot underground layers. This is achieved by an artificially created petrothermal circulation system (PCS), which provides movement of a heat carrier extracting the heat of hot rocks of the Earth, its input and output to the surface with subsequent use in heat and power supply systems [1-4]. The most important element of the PCS is a permeable reservoir between wells, formed by hydraulic fracturing or by stimulating natural defects. The latter approach prevails in practice. The PCS efficiency is characterized by an increase in heat removal from the heat transfer underground surface, whose role is performed by a system of macro- and microcracks. Recently, the “Eavor” company (Canada) has proposed new PCS schemes with the formation of an underground reservoir by drilling vertical wells connected by multi-barrel horizontal holes [5]. A patented working fluid is used as a coolant instead of water, and a protective coating is applied to the well walls to avoid coolant losses.

* Corresponding author, e-mail: aleks@itp.nsc.ru

In Russia, at the moment there is only one circulation system of the so-called doublet type at the Khankala geothermal water deposit with an aquifer temperature of 96°C at a depth of 900 m and a distance between water intake and return of 1 km [6]. The technologies for the creation of underground circulation systems are at the research stage. Works in this direction are very relevant because they open up the possibility of constructing an underground energy facility with a developed heat transfer surface.

In the issue of deep heat utilization, it is necessary to distinguish two aspects: first, the creation of an underground (at depths of more than 3 km) petrocirculatory system that allows achieving the required thermodynamic parameters; second, an increase in the effectiveness of underground thermal energy conversion. Research on the use of the heat of dry hot rocks of the Earth continues.

2. Working fluids

Traditional steam-water heating technology dictates an increase in the efficiency of power plants in the direction of increasing the initial temperature of the thermodynamic Rankine cycle. In terms of energy, a substance with a high critical temperature is preferable for generating electricity, which allows the maximum of the enthalpy difference in the turbine with the minimal flow rate of the working fluid. However, for an underground power boiler, which is the most capital-intensive element of this system, a lower boiling point of the working fluid at normal pressure with the exception of air penetration to the working circuit and associated negative processes during condensation is also an important property. This allows a reduction in the initial temperature of the working fluid (the capital intensity of the petrocirculatory system) and a compensation for the reduction of the available heat drop due to its increase in the region of low temperatures. There are also the counter-influencing factors.

Underground heat is used both for heat supply (in 2015, installed capacity in the world was 70.3 GW) and for electricity generation (in 2020, installed capacity reached 16 GW) [2]. We are interested in the electricity production from the deep heat. At the initial stage of petrothermal energy development, when operating not very deep wells, we will have to deal with sources at relatively low temperatures (70-130°C). In this case, the generally accepted approach is to use binary schemes based on the organic Rankine cycle (ORC). There are many candidates for the role of an ORC working fluid with a low boiling point. Therefore, one of the main tasks is to choose the working fluid depending on the numerous requirements and criteria [2,7-12]. Practice shows that only several substances are used for commercial purposes: R134a, R245fa, n-pentane, isobutane, isopentane, and isobutane-isopentane mixture. Utilization of liquids with a low boiling point as the working fluids in electricity production allows more complete utilization of the low potential heat [8-10].

There are quite a few examples of commercial projects with these working fluids. The Swedish company Atlas Copco, commissioned by Turkey and Germany, manufactured 5 turbo-generating units with a unit capacity of up to 22.5 MW using isobutane and n-butane (substances with minimal greenhouse effect on the environment) as the working fluids; the Exergy company (Italy) supplied 11 similar installations with a capacity of up to 12 MW on hydrocarbon to Turkey, Italy and France in 2014-2015.General Electric Energy (USA) has manufactured over 100 power plants using refrigerant R245fa (pentafluoropropane) as a working fluid for generating electricity. The “Ormat”company (USA) supplied more than 200 turbo-generating units with a unit capacity of up to 15 MW, which use C_5H_{12} (pentane) with an initial vapor temperature in front of the turbine of 105 - 180°C, to the foreign and domestic markets. It is necessary to note the advertising nature of commercial information and incompleteness of the submitted data on the published projects with “N/A” marks. This makes it necessary to determine the efficiency of using low-temperature heat carriers in the steam-power organic Rankine cycle (ORC).

The use of natural gases as heat carriers removes issues related to ozone safety and greenhouse effect imposed on refrigerants: working fluids with a low boiling point. The efficiency of the converter plant can be increased by abandoning the ground-based steam generator and the associated energy losses. Ensuring the explosion safety of hydrocarbons is a technically solved task. The disadvantage of this technology is the low efficiency of the heat transfer process (*Table 1*). Table 1 shows that water and n-pentane (R601) have the best heat transfer characteristics and are widely used in practice [2,7,13-15]. However, the use of steam-water coolant in electricity generation is limited by an increase in the final humidity of steam (~12%), and at a temperature of 120°C it returns to the geothermal well (Mutnovskaya geothermal power plant), or is discharged into the nearest reservoir (Pauzhetskaya geothermal power plant).

Table 1

Thermophysical properties of heat-transmitting rocks, material of the heat exchanger wall and heat carriers

Parameter	Rocks [16]	Steel	Water	R601	R245fa
			att = 20 °C		
Density ρ, kg/m^3	2700 (1100 – 5100)	7850	998.2	626.0	1352.2
Heat capacity C_p, kJ /(kg· K)	1.0 (0.5 – 1.2)	0.48	4.183	2.28	1.328
Thermal conductivity λ, W/(m·K)	2.7 (0.2 - 8.2)	58.0	0.599	0.114	0.0915

Another important characteristic is the effect of coolant flow rate in the petrocirculatory system on its service life [16].The flow rate of the working fluid at a given electric power of the turbo-generator depends on the available heat transfer per a turbine.

The purpose of this work is to identify the potential energy options for using the working fluid of the cycle, to establish the relationship between the initial and final parameters of the expansion process of a low-temperature gas coolant in an axial multistage turbine and to identify restrictions on the diverging angle of its last stages [17], determining the efficiency and reliability of operation without changing the turbine blade profile [18-20], which greatly simplifies the process of converting undergound thermal energy into electricity (Fig.1).

3. Research method

To solve the problem, a method of thermal calculation of the process of gaseous n-pentane expansion in the flow part of an axial turbine is used. At that, the available heat drop is divided into expansion stages along the flow path of the turbine taking into account the subsonic outflow of n-pentane from the considered stage. This method allows determination of thermophysical properties, gas-dynamic and structural characteristics of the flow path of a multi-stage turbine for each stage of gas coolant expansion.

a) b)

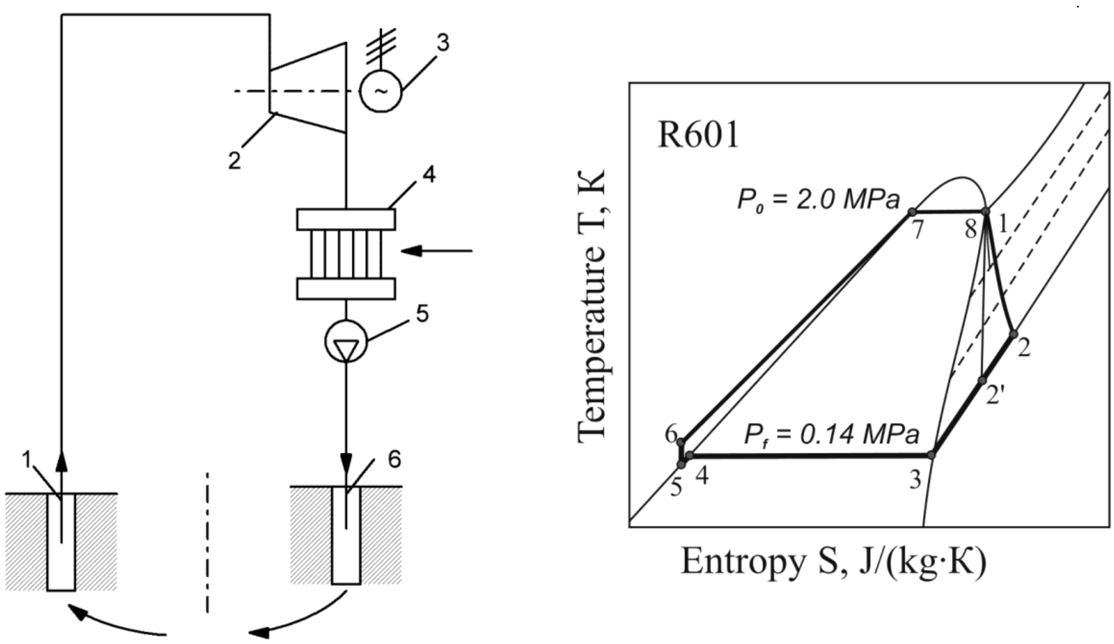


Fig.1. The basic technological scheme of the underground energy complex and the Rankine cycle (ORC) for electricity generation

a) 1 – production well, 2 – turbine, 3 – electric generator, 4 – condenser, 5 – feed pump, 6 – injection well; b) 1 - 2 the process of n-pentane expansion, 2 - 3 – overheating removal, 3 - 4 – condensation, 4 – 5 – condensate supercooling, 5 - 6 – increasing the working fluid pressure, 6 - 7 - 8 - 1 – heating, boiling and overheating of the working fluid in the petrothermal circulation system.

4. Discussion of the results

The presented technological scheme implements the thermodynamic Rankine cycle (ORC), where the processes of heating (6-7), boiling (7-8) and overheating (8-1) of organic liquid are carried out due to the heat of dry hot rocks in an underground power boiler. This solution allows the use of relatively low temperatures of the hot source for generating electricity during n-pentane expansion in the turbine (1-2). The physical properties of working fluids at the critical point are presented in Table 2 [13-15].

A comparative analysis of the critical characteristics of the working fluids presented in Table 2 shows that n-pentane is more preferable, and it is accepted for further research because it has a higher available enthalpy difference in the process of energy conversion. It is noted in the comparative analysis of implementations of the ORC cycles using R245fa, n-pentane, and their zeotropic mixtures with different concentrations as the working fluids that R245fa freon has low toxicity and does not ignite [21]. However, its use in a cycle is less economically efficient.

Table 2

Physical properties. Temperature of heat carriers at the critical point and under normal conditions

Parameter	C_5H_{12} n-pentane	C_4H_{10} isobutane	R245fa pentafluoropropane
P_{cr}, bar	33.74	36.47	36.5
T_{cr}, ^oC	196.62	134.98	154.05
T_b at 760 mm Hg T_{cr},^oC	36.07	- 11.73	15.1

The need for this study relates to the task of maximizing the use of the available heat transfer on the turbine in the presence of restrictions on the efficiency of the working fluid expansion caused by the influence of its volumetric flow [23].

An axial, multistage turbine with a capacity of 4000 kW with full gas supply along the entire circumference of the nozzle grid is considered. The choice of an axial multistage turbine is conditioned by its greater energy efficiency as compared to a radial type turbine due to the beneficial use of part of the internal losses during the working fluid expansion in the intermediate stages of the turbine. At that, the rise in the cost of a multi-stage turbine is not a determining factor in the cost of an underground energy system. To achieve this goal, the process of n-pentane expansion in the turbine was studied taking into account internal energy losses per the expansion stages [18-20].

The thermodynamic domain of changes in the parameters of n-pentane during thermal energy conversion is presented in Table 3.

Table 3

Changes in the parameters of n-pentane (R601) during expansion in an axial turbine

Parameter	Initial parameters before the nozzle grid of the first stage (1)	Final parameters of the last stage of the turbine (2)
Pressure P, bar	20	1.4
Temperature T, ^oC	170	45.7
Enthalpy h, kJ/kg	1136.73	1005.5
Density ρ, kg/m^3	59.62	3.9

Figure 2 shows the expansion process in the turbine stage of the cycle, the methodological provisions of which are set out in [18-20]. However, depending on the task, there are the features of calculation process modeling, which are influenced by the properties of the working fluid that determine the quantitative values of the design characteristics.

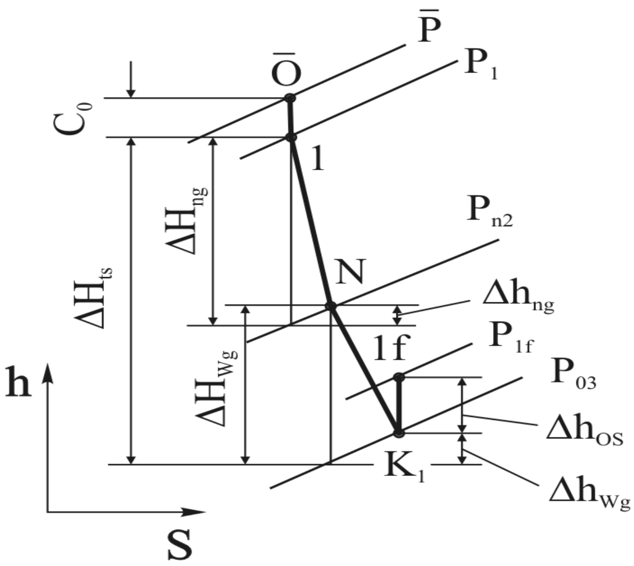


Fig.2. Calculation of the turbine stage in the h-s diagram

ΔH_{ts}, ΔH_{ng}, ΔH_{wg} - adiabatics of the enthalpy difference and operation processes ($\bar{O}$ - 1 , 1 – N , - N - K_1) in the stage, nozzle and working grids, respectively.

To evaluate the efficiency of each of the turbine stages, a relative coefficient of blade efficiency, which characterizes the perfection degree of energy conversion, is adopted. At the same time, energy losses in the nozzle grid Δh_{ng}, on the turbine blades Δh_{tb} and with the output velocity are taken into account Δh_{ov}. In addition to these energy losses, there are a number of other losses associated with the design of turbine elements that are not considered here (working fluid leakages, its friction against the surface of the disk, bandages, etc.).

Mathematical modeling of the process of n-pentane expansion in a turbine includes an independent separation of the available heat drop by individual stages. At that, the initial parameters (pressure and temperature) of n-pentane at the inlet to the nozzle grid of the first stage are calculated with the zero velocity of the working flow after its throttling in the control valves (C_o = 0). For all subsequent stages of an axial multistage turbine (C_o = 1), energy losses with an output velocity Δh_{ov} are beneficially used, increasing the enthalpy and temperature of the working fluid at a reduced pressure of the expansion process.

Table 4 shows the design characteristics and energy efficiency of the first stage (h_1) of n-pentane expansion in an axial turbine.

Table 4
Stage characteristics

Characteristics	Value
Adiabatic adiabatic of the enthalpy difference h_1, kJ/kg	27.17
Reactivity of the stage	0.5
Adiabatic (fictitious) gas velocity in a stage C_{af}, m/s	233.1
Circumferential gas velocity along the average stage diameter U, m/s	109.56
The angle of the working flow output from the nozzle gridα_1, degrees	11
Average diameter of the stage d_a, m	0.697
Working blade height h_{wb}, m	0.0188
Absolute velocity of gas at the nozzle grid outlet C_{1t} , m/s	156.6
Relative velocity of gas at the working grid outlet W_2, m/s	158.9
Absolute velocity gas at the working grid outlet C_2 , m/s	58.0
Energy losses in the nozzle grid Δh_{ng}, kJ/kg	0.756
Energy losses on the working blades Δh_{wb}, kJ/kg	1.441
Energy loss with output velocity Δh_{ov}, kJ/kg	1.682
Relative blade efficiency of the stage η_{rb}	0.857

The coolant flow rate at a given electric power of the generator was calculated by expression

$$D = N_g/(\Delta h_a \cdot \eta_{ri} \cdot \eta_m \cdot \eta_g) , \quad (1)$$

here D is the flow rate of gas per turbine, kg/s; N_g is the power of electric generator, kW; Δh_a is available heat transfer to the turbine, kJ/kg; η_{ri} is the relative internal efficiency of the turbine; η_m, η_g are the efficiency of the mechanical and electric generators, respectively.

The relative blade efficiency of the stage is determined by dependence

$$\eta_{rb} = 1 - \sum \Delta h_i / \Delta h_{ai,} \quad (2)$$

where $\sum \Delta h_i$ is the sum of energy losses in the flow part of the turbine (Δh_{ng}, Δh_{wg}, Δh_{ov}).

An increase in triggered heat transfer (the process takes place in the area of superheated gas) is accompanied by a decrease in the working fluid density (Table 3), an increase in the volume flow of the gas coolant ($D \cdot v_{1t}$) and an increase in the working blade height of the last expansion stage, which leads to a decrease in the ratio of the average diameter of the blade row of stage d_a to the length of the working blade h_{wb} $\theta = d_a/l_{wb}$ [17-20, 23-28]. It should be noted that when analyzing the diverging angle of the last stages of the turbine, specialists use ratio θ and ratio 1/θ, whose defining characteristic is the working blade length.

$$h_{ng} = D \cdot v_{1t} / (\mu_1 \cdot \pi \cdot d_{a1} \cdot C_{1t} \cdot \sin \alpha_1 \cdot e), \quad (3)$$

where h_{ng} is the height of the nozzle apparatus, m; v_{1t} is the specific volume of gas behind the nozzle grid, m^3/kg; α_1 the angle of gas outlet from the nozzle; e is the part of the impeller

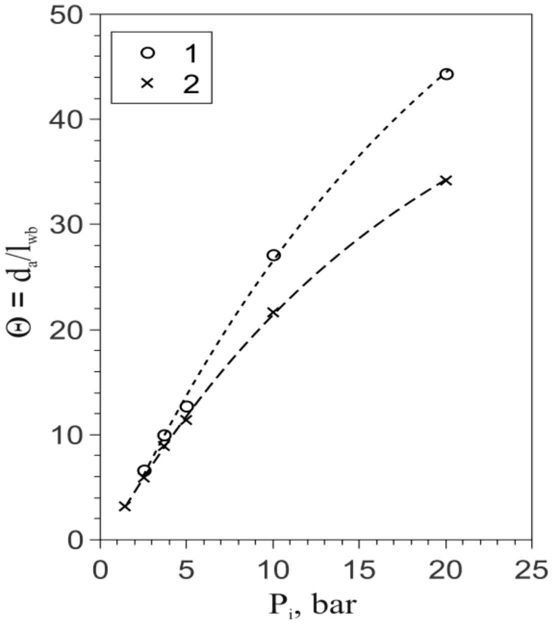


Fig.3. Influence of pressure P and flow rate D of n-pentane during its expansion on the diverging angle.
1. D = 43.2 kg/s (P_f = 0.25 MPa; θ = 5.9), 2. D = 38.5 kg/s (P_f =0.25 MPa; θ = 6.6).

circumference through which n-pentane is fed, C_{1t} is the theoretical flow rate of the working fluid from the nozzle grid, m/s;

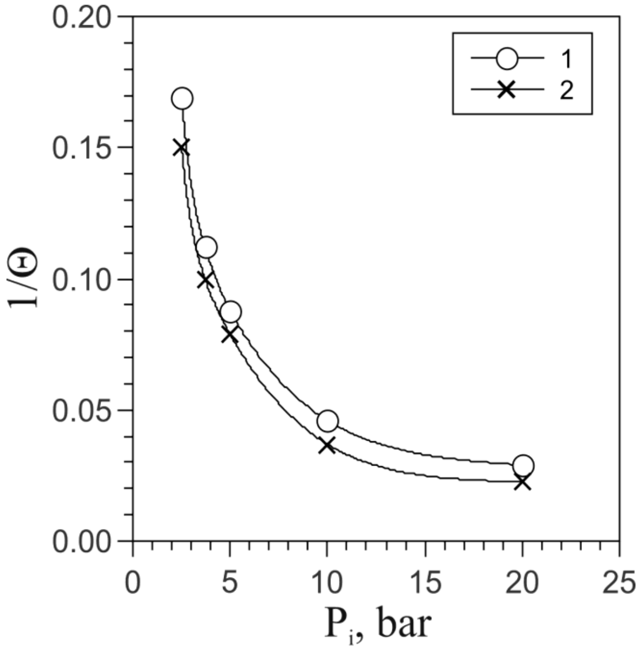
$$C_{1t} = (2 \cdot \eta_{ri} + x \cdot C_o)^{0.5}, \quad (4)$$

were x is a ratio of the circumferential velocity to the equivalent velocity of the adiabatic outflow.

The height of the working blade (h_{wb}, m) is assumed to be equal to the increase in the size of the nozzle height to exclude working fluid leaks through the axial and radial gaps between the nozzle and working grids (at the root and on the periphery) [20].

Figure 3 shows a change in θ of the stages of n-pentane expansion depending on the working fluid pressure in front of the stage. The main attention in the analysis of the expansion process is associated with an increase in the efficiency of thermal energy conversion. Consideration of the axial turbine allowed reduction in the working fluid flow due to a higher relative internal efficiency as compared to the radial one. The relative blade efficiency of the intermediate stages of n-pentane expansion is 86 %.

According to Fig. 4, in the area of low pressures close to atmospheric, there is a sharp increase in the height of working blades of the stage. This reduces operational reliability of the turbine and the need to change the profile of the working grid. Despite the possibility of a further increase in the triggered heat transfer (the process takes place in the area of superheated gas), to preserve the original profile of the blades without involving additional technologies, a restriction has been introduced on the diverging angle of the working grid of the last expansion stage θ = 2.5– 3.0 [23].



.

Fig.4. Variation 1/θ of the n-pentane diverging angle depending on pressure.

1. D = 43.2 kg/s (P_f = 0.25MPa; θ = 5.9), 2. D = 38.5 kg/s (P_f =0.25MPa; θ = 6.6).

In addition to the gas-dynamic and thermodynamic characteristics of expansion process along the flow part of the turbine, the unit power of the energy generator has a great influence on installation efficiency, whose implementation determines the technical and economic feasibility of the energy system [29]. The single power of the turbo-generator affects not only the occurrence of a critical regime during expansion. The ultimate diverging angle of the last stages (and the power of installation) can be increased by increasing the final pressure (an optimization factor that has a positive effect), using a two-flow turbine where the total gas flow is divided into two symmetrical flows, each directed to its own flow part. However, the fundamental basis for choosing a unit power of the generator is the thermodynamic options for generating electricity, which are determined by the properties of the working fluid used for this purpose. The issue of achieving the upper temperature of n-pentane in the steam-power cycle is a complex circuit-parametric problem of the entire petrothermal energy system.

5. Conclusions

1. Studies have shown that n-pentane is an effective working fluid for generating electricity when implementing the low-potential heat of dry hot rocks of the Earth with a temperature of less than 170°C.

2. The results obtained allow a constructive study of the elements of the complex, design and creation of an underground pilot-industrial renewable energy source.

3. Computational and methodological studies have been carried out to establish the relationship between the initial and final parameters of low-temperature n-pentane and the possibilities of its use for generating electric energy in an axial multistage turbine are shown.

4. An estimate of the ratio of the average diameters of the turbine stages to the height of the working blades along its flow part is given.

5. At a fixed electrical power, the effect of pressure and flow rate of n-pentane on the diverging angle of expansion stages is shown. A satisfactory profile of the flow part of a jet-type axial multistage turbine with subsonic flow velocities of the working fluid is obtained.

Nomenclature

ΔH – adiabatic heat transitions (kJ/kg)
Δh – energy losses (kJ/kg)
d – average diameter of the stage (m)
U – circumferential velocity at the average diameter of the stage (m/s)
D – working fluid consumption (kg/s)
C – working fluid speed (m/s)
l – height (length) (m)

Indexes

0 – initial value of the parameter
1 – first stage
i –the current value of the parameter
f – the final value of the parameter
cr – the critical point
ng – nozzle grid
wg – working grid
os – output speed

Greek letters

α – the direction of the working fluid in the turbine stage α
θ – fidelity of the working grid
μ – working fluid speed coefficient
η – energy efficiency of the process

REFERENCES

[1] DiPippo, R. Geothermal Power Plants. Principles, Applications, and Case Studies (2005) Elsevier, Oxford, England, 450 p.

[2] Alekseenko S.V., "Geothermal energy technology", Breakthrough ideas in energy for the next 10 years, Association on development of the international research and projects in the field of energy // Global energy, 2022, pp. 135 – 155.

[3] Gnatus N.A. Petrothermal energy of Russia. Prospects of development and development // - M.: INP RAS. 2013, 27 p. (in Russian).

[4] Alkhasov A.B. Geothermal energy: problems, resources, technologies // - M.: Fizmatlitgiz, 2008, 376 p.

[5] The world's first truly scalable form of Green baseload power demonstrated by Eavor // - M.: INP RAS. 2013, 27 p.

[6] Zaurbekov S.S., Mintsaev M.S., Labazanov M.M., Cherkasov S.V., Butuzo V.V. Results of design development for the pilot geothermal plant at the Khankala deposit in Chechen Republic // Ecology, Environment and Conservation, 2015, т. 21, pp. 134-139.

[7] Proceedings World Geothermal Congress 2020 //Reykjavik, Iceland, - 2020.

[8] Song V.,Wang J., Zhou E. Thermodynamic analysis of a transcritical CO_2 power cycle driven by solar energy with liquefied natural gas as its heat sink//Appl.Energy , 2012, 92, pp.194-203.

[9] Shengjun Z., Huaixin W., Tao G. Performance comparison and parametric optimization of subcritical organic Rankine cycle (ORC) and transcritical power cycle system for low-temperature geothermal power generation //Appl.Energy , 2011 , 88(8), pp. 2802- 2810.

[10] Chao He, Chao Liu, Hong Gao, Hui Xie, Yourong Li, Shuangying, Jinliang Xu. The optimal evaporation temperature and working fluids for subcritical organic Rankine cycle // Energy, 38 (2012), pp. 138-143.

[11] Emilie Sauret, Andrew S. Rowlands. Candidate radial-inflow turbines and high-density working fluids for geothermal power systes // Energy, 36 (2011), pp. 4460-4467.

[12] Emiliano Casati. New concepts for organic Rankine cycle power systems. Milan, 2014, 249 p.

[13] http://NIST Chemistry WebBook, SRD 69.Thermophysical Properties of 1,1,13,3-Pentafluoropropan (R245fa).

[14] Standard reference data. Isobutane liquid and gaseous // M.–Standartinform, 2018.

[15] N-pentane. Thermodynamic properties in the triple point temperature range up to 700 K at pressures up to 100 MPa. GSSSD 291-2013, (2013), 54 p. (in Russian).

[16] Alekseenko S.V., Borodulin V.A., Gnatus N. A., Nizovtsev M.I., Smirnova N.N. Problems and prospects of development of petrothermal energy (review) // Thermophysics and Aeromechanics, 23 (2016), pp.1-16.

[17] Zakharov A.V., Penza G.G. Critical modes of operation of the last stages of powerful steam turbines with blades of maximum length // Energetika, 2013, vol.1, pp. 41-44.

[18] Shcheglyaev A.V. Steam turbines // - M.: Energiya, 1976, 368 p. (in Russian).

[19] Vawra M.H. Aero-thermodynamics and flow in turbomachines. N.Y.-London,1960, 609 p.

[20] Kirillov I.I., Ivanov V.A., Kirillov A.I. Steam turbines and steam turbine installations // - L.: - Mashinostroenie, 1978, 278 p., (in Russian).

[21] Nicola G.I. Vapor Pressure and Gas Phase P-V-T Data for 1.1.1.3.3-pentafluoropropane

(R245fa) //Journal of Chemical and Engineering Data, 2001, vol.46, No.6, p.1619-1622.
[22] Van Long Le, Abdelhamid Kheiri, Michel Feidt, Sandrine Pelloux-Prayer. Thermodinamic and economic optimizations of a waste heat to power plant driven by a subcritical ORC (Organic Rankine Cycle) using pure or zeotropic working fluid //Energy, 78 (2014), pp. 622- 638.
[23] Alekseenko S.V. and Ogurechnikov L.A. Concerning utilization of heat from low-potential Sources // Thermal science, 2019, vol.23, No. – 6B, pp. 4023-4030.
[24] Kirillov I.I., Nosovitsky A.I., Shpenser G.G. Some design features of the last stages of steam turbines // Thermal power engineering, 1981, No. 9, pp. 34-36, (in Russian).
[25] Vawra M.H. Aero-thermodynamics and flow in turbomachines // Journal of royal Aeronautical society, 1961,vol.65, pp. 144-145.
[26] Deitch M.E., Gubarev A.V., Filippov G.A., Wang Zhong-qi. A new method for profiling guide grids of steps with a small d/l // Heat power engineering, 1962, No. 8, pp. 42-47.
[27] Deich M.E., Zaryankin A.E., Filippov G.A., Zatsepin M.F. Methods of increasing the efficiency of steps with low blade heights // Heat power engineering, MAIK “Science/Interperiodics". - M. : 1960, No. 2, pp. 18-24, (in Russian).
[28] Delhaye J.M., Jiot M., Riethmuller M.L. Thermohydraulics of two-phase systems for Industrial design and nuclear enginsering. –N.Y.:McGraw-Hill Book Co., 1981, 525 pp.
[29] Alekseenko S.V. and Ogurechnikov L.A. Heat exchange in steam generator with low-boiling point fluid // Thermal science, 2021, vol.25, No. 5A, pp. 3569-3578.

Submitted: 14.7.2023.
Revised: 15.9.2023.
Accepted: 26.9.2023.