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Efficiency of Geothermal Power Plant Cycles with Different Heat-Carriers

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ABSTRACT

Results of numerical studies of the effectiveness of the thermal scheme of geothermal power plant are presented. Subcritical and supercritical Rankine cycles with different working mediums both pure substances and their mixtures in the temperature range of 70÷200°C are studied. The effect of operating parameters: temperature and vapor pressure in front of and behind the turbine, heat exchange efficiency in the evaporator and condenser and ambient temperature has been found. The dependence of electrical power being generated on the geothermal fluid temperature is defined. The analysis of exergy efficiency of the power plant has been done.

1. INTRODUCTION

There is an annual global increase of fixed power of geothermal power plants (GPP) by 10-20%, in 2015 the capacity is about 12GW (Bertani, 2015). Efficient conversion of geothermal energy into electricity requires the creation of a special power equipment. Conventional power plants working at water vapor (steam turbine at pressure of 20-30 MPa) are not suitable due to the low vapor density and accordingly large turbine size. The use of low-boiling working fluids allows to reduce the size of the turbine. Power plants with ORC binary cycle are widely used in geothermal heat engineering .The GPP heat carrier in the binary cycle receives heat from geothermal water (brine), evaporates, expands in the turbine, condenses and returns to the evaporator by means of the feed pump. The first geothermal power plant with the capacity of 670 kW with a binary cycle was launched in 1967 in the village of Paratunka (USSR) on the Kamchatka Peninsula (Moskvicheva, 1970). R12 Freon was used as the heat-carrier. Temperature of the thermal water is 80°C.The radial turbine with inlet pressure of 1.4 MPa and outlet one of 0.8 MPa was applied. Specific power output was equal to 9.7 kJ/kg. In 1992 a project of the geothermal unit with capacity of 1.5 MW at the R142b coolant with the radial-axial dual-flow turbine for using geothermal water heat with temperature of 165°C was developed. Pressure at the turbine inlet was 2.5 MPa, at the outlet it was 0.4 MPa, calculated specific power output was equal to 41 kJ/kg. In the city Lorderello (Italy) a binary power station with a capacity of 250 kW was developed in 1912.

At present binary stations are widely used. In June 2004 155 geothermal power plants generating 274 MW of energy, that is about 30% of all operating geothermal power plants operated in 16 countries (Povarov, 2000; Labuntsov, 1991).

The analysis of the world's geothermal resources shows that fields with stratum water temperature below 150°C dominate. In Ukraine there are also geothermal fields with temperature of 150–180°C, stocks of which allow to produce about 200 MW of electricity. However, most of the fields are characterized by temperature mode below 130°C. At the temperature of heating heat carrier of 65–150°C it is difficult to build a steam power plant, which can have effective economic parameters. As foreign experience shows the lower geothermal source temperature is the more sophisticated technology is required (DiPippo, 2005).

2. STATE OF THE PROBLEM

The important characteristics of the working medium for the binary plant is a form of a saturated steam curve, shown in the P-H coordinates (pressure-enthalpy), Fig. 1.

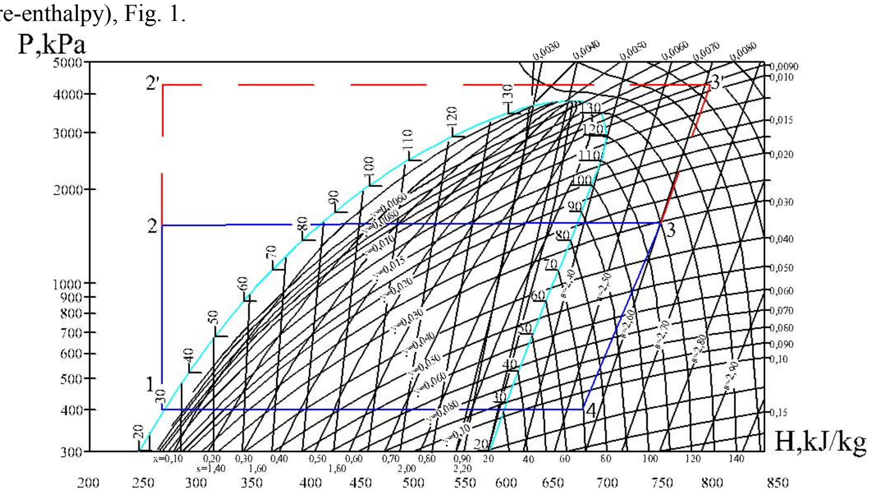

Fig.1: The "pressure-enthalpy"diagram of binary cycle of geothermal power station

The curve of saturated water vapor has a positive slope, and some hydrocarbons and coolants show a negative bias for the saturated vapor line segments. Processes of steam expansion in the turbine in the Rankine cycle are shown by curves 3-4 and 3'-4. For some working mediums the process of steam expansion occurs from the saturated vapor line to the superheating region; the water requires a considerable degree of steam superheating to avoid significant moisture in turbine exhaust vapor (condition 4). The selection of working medium is of great importance for the effective operation of the binary plant. While there are a lot of suitable working mediums, there also occur many choice restrictions which relate to thermodynamic properties, as well as their impact on safety and the environment and their toxicity should be taken into consideration. These issues are determined by the following parameters: flammability, toxicity, possibility of exposure to ozone (ODP) and the probability of impact on global warming (GWP). By optimizing the parameters of the cycle for different working mediums and evaluating the effectiveness of the cycle one can choose a working medium.

Nowadays abroad the creation of turbines operating with non-azeotropic mixtures, boiling and condensation of which occur at varying temperatures is paid great attention to (Kalina cycle). By changing the mixture composition in a wide range, one can get the properties that in set cycle parameters can provide the highest efficiency of power plants. Using water-ammonia mixtures allows to increase the conversion efficiency of geothermal energy and optimize cycle due to changes in properties of the mixture (Vasil'ev, 1996).

3. MODELING TECHNIQUE

The calculated relations for the items of geothermal power plant equipment are given below :

a) capacity of the pump is determined by the relationship:

$$N = \frac{\left[(P_{out} - P_{in}) \cdot m_{f} \right]}{\rho_{f} \cdot \eta} \tag{1}$$

where P_{output}, P_{input} are pressure at the outlet and inlet of the pump; m_f – pumped fluid flow rate; ρ_f –fluid density ; η – adiabatic efficiency of the pump.

To calculate the fluid temperature at the outlet of the pump the following ratio is used:

$$N = m_{f} \cdot \left[h_{f} \cdot (P_{out}, T_{out}) - h_{f} \cdot (P_{in}, T_{in}) \right] \tag{2}$$

where h – enthalpy of the fluid at a given pressure P and temperature T.

b) calculation of the turbines is made by the adiabatic model. The flow pressure and temperature at the turbine inlet, pressure at the turbine outlet, working fluid flow rate and efficiency of the turbine are used as the initial data.

Enthalpy of the ideal expansion process is determined by the condition of adiabatic expansion of the flow:

$$h_{id.output} = h(P_{out}, S_{in}) \tag{3}$$

where S_{input} – specific enthalpy of the flow at turbine input defined by known temperature and pressure.

Then the power being generated in the ideal turbine at the adiabatic expansion process is:

$$N_{id} = \left[h_{id.out} - h(P_{in}, T_{in}) \right] \cdot m_{s} \tag{4}$$

where m_s – vapor flow rate through the turbine.

Net power from the shaft of the turbine is:

$$N_{real} = N_{id} \cdot \eta \tag{5}$$

where η is adiabatic efficiency of the turbine.

Enthalpy at the turbine outlet in the real process (including losses) is defined as:

$$h(P_{out}, S_{out}) = h(P_{in}, T_{in}) - (N_{real} / m_{s}) \tag{6}$$

which makes it possible to calculate the temperature of steam at the turbine outlet.

c) countercurrent heat exchanger. For determining the parameters in the counterflow heat exchanger in our formulation of the problem (i.e. excluding hydraulic losses in the flowing part and heat losses into the ambient air) the system of three equations is used:

$$\begin{cases} Q = m_{hot} \cdot [h(T_{hot.in}) - h(T_{hot.out})] \\ Q = m_{cold} \cdot [h(T_{cold.in}) - h(T_{cold.out})] \\ Q = k \cdot F \cdot \Delta T \end{cases} \tag{7}$$

where m_{hot}, m_{cold} are mass flow rate of the hot and cold heat carriers; $T_{hot.in}$, $T_{hot.out}$, $T_{cold.out}$, $T_{cold.in}$ are temperatures at the inlet and outlet of the cold and hot heat carriers, respectively; F is heat exchange surface area; k –is an average heat transfer coefficient of the heat exchanger, taken from the references; ΔT is an average value of the temperature difference of the heat exchanger.

To find the mean value of the temperature difference of the heat exchanger the following relation is used:

$$\Delta T = \frac{\left[\left(T_{hot.in} - T_{cold.out} \right) - \left(T_{hot.out} - T_{cold.in} \right) \right]}{\ln \dfrac{T_{hot.in} - T_{cold.out}}{T_{hot.out} - T_{cold.in}}} \qquad (8)$$

Due to the complexity of heat transfer processes taking place in a heat exchanger in order to improve the accuracy by setting the appropriate value k the heat exchanger is conventionally divided into sections with phase transitions and without them, as well as providing the difference in heat exchange temperature at the pinch point (point of minimum temperature difference).

The presence of the regenerative heat exchanger in which the heat of geothermal water at the evaporator outlet (or turbine steam) is given for heating the low-temperature heat carrier in front of the evaporator reduces the thermal load both of the evaporator and condenser. Rational use of the geothermal energy fluid requires the development of optimal thermal circuit and determination of the optimal cycle parameters of the power plant. In general, optimal technical and economic parameters of two-circuit geothermal power station can be obtained on the basis of the analysis and optimization of the combined thermodynamic, technological, geological, hydrological and other influencing parameters. As the results of the performed research show (DiPippo, 2005; Moskvicheva, 1970; Vasil'ev, 1996) maximum net electrical power is obtained by increasing the flow rate of geothermal water, circulating in the geothermal circulation system, and optimizing the thermodynamic cycle parameters of the secondary circuit. The process of steam expansion in the turbine was regarded as an adiabatic one. The calculation of steam expansion in a polytropic process is mainly associated with the technical perfection of the technological equipment (Vasil'ev, 1996).

4. RESULTS OF THE RESEARCH

As it was mentioned earlier, nowadays environmentally ozone-savety substances of a new generation – freons of ethane, propane and butane series appeared. The prospect of their use as a working medium of the converter cycle of geothermal energy into electricity has been studied in this work.

Numerical studies of Rankine cycle of geothermal power plant with a binary cycle with different working media (R13, R13b1, R22, R114, R134a, R142b, R143a, R152a, R218, R318, R600a/R141b, R600a/R161, R600a/R602) have been done .

The calculations of thermodynamic parameters of the Rankine cycle have been performed under the following conditions:

- adiabatic efficiency of the turbine (0,70–0,80);
- efficiency of pump – 0,75–0,80;
- ambient temperature 15; 20; 25°C.
- ΔT_{min} – 5, 10, 15K underrecuperation in the regenerative heat exchanger and evaporator (minimum temperature difference between geothermal water and a working medium).

Pressure at the outlet of the turbine is determined by the condition of the saturated state of the working medium. Turbine inlet pressure is determined by the value of the temperature and at the evaporator outlet it is selected as maximal as possible at which the vapor flow at the turbine outlet will be a single phase one.

The value of specific electrical power for different working media and their mixtures are shown in Table 1.

Table1.The value of the specific electric power

Heating medium	Parameter					
	P_e, kPa	t_e, °C	t_n, °C	N,kW/(kg/s)	m, (kg/s)	COPc, %
*R13b1	5000	-	40.64	34.10	3.56	0.087
R22	2500	61.16	55.28	24.73	1.35	0.073
*R22	5000	-	39.0	43.37	1.90	0.010
R114	866	76.88	70.54	23.40	1.38	0.089
R134a	3800	97.65	71.88	27.99	1.10	0.019
*R134a	5000	-	38.52	43.87	1.95	0.011
R142b	1263	75.22	70.18	24.44	0.94	0.092
R143a	3500	69.56	41.70	29.89	1.60	0.077
*R143a	5000	-	40.42	35.95	1.79	0.091
R152a	2175	77.03	69.16	26.02	0.73	0.097
R218	5000	-	40.79	32.76	3.11	0.084
R318	1500	84.97	69.48	22.79	1.51	0.086
isobutane	1400	81.98	75.86	22.35	0.50	0.010

Note: * is a supercritical cycle; the heat carrier in the secondary circuit $t_{g.w.}$=130°C; t_a=15°C

Parameters of supercritical cycle are shown in Table 2.

Table 2.Parameters of supercritical cycle (R13+R13b)

Heating medium	Parameters		
	N,kW/(kg/s)	m, (kg/s)	COPc, %
80%R13 and 20%R13b	17,00	3,02	4,6
50%R13 and 50%R13b	22,57	3,13	6,04
20%R13 and 80%R13b	28,48	3,30	7,58
10%R13 and 90%R13b	30,57	3,36	8,14

Analysis of supercritical cycles for high temperature geothermal fluid ($t_{g.w.}$=180°C) has shown that a supercritical cycle like the triangular cycle with initial pressure of 5,0 MPa is the most effective one in terms of getting the maximum electric power (Alkhasov, 2001). Electric power being generated by the turbine is increased by 11% compared with subcritical cycle (initial pressure of 3,4 MPa) Density of the working fluid in front of the turbine is 1,7 times higher than in the subcritical cycle, thereby reducing the overall dimensions of the turbine plant. R13b1 coolant is proposed to be used in the secondary circuit at supercritical parameters for a geothermal plant with a temperature of geothermal water 120°C (Alkhasov, 2001). Using R142a, R134b coolants does not allow to expand the range of using geothermal fluid with temperature below 15–165°C.

Some results of numeral investigations of a GeoPS working cycle having different working mediums are given in table 3.

Table 3. Main parameters of GeoPS working cycle

Working medium	*t_3,ºC*	*t_4,ºC*	*P_3, kPa*	*P_4, kPa*	*ρ_3, kg/m^3*	*W_{3-4}, kW·s/kg*	*COP, %*
Subcritical cycle							
R600a/R602	197	114.9	3850	197	116	64.0	13.99
R600a/R161	197	132.7	3600	489	61.0	85.3	13.10
R600a/R141b	197	140.5	2500	280	49.5	74.5	12.89
	197	124.6	3700	280	82.8	92.7	14.48
Supercritical cycle							
R600a/R602	197	76.3	6000	197	278.6	92.2	13.64
R600a/R161	197	106.0	6000	489	119.8	108.3	15.08
R600a/R141b	197	95.3	6000	280	177.0	108.4	15.55

Thermal properties of Freon are shown in (Prausnitz, 1986; Smith, 1987; Sandler, 1989; Van Ness, 1982).

Generation of electric power of geothermal power plant increases with the increase of the temperature of geothermal fluid (Fig. 2.).

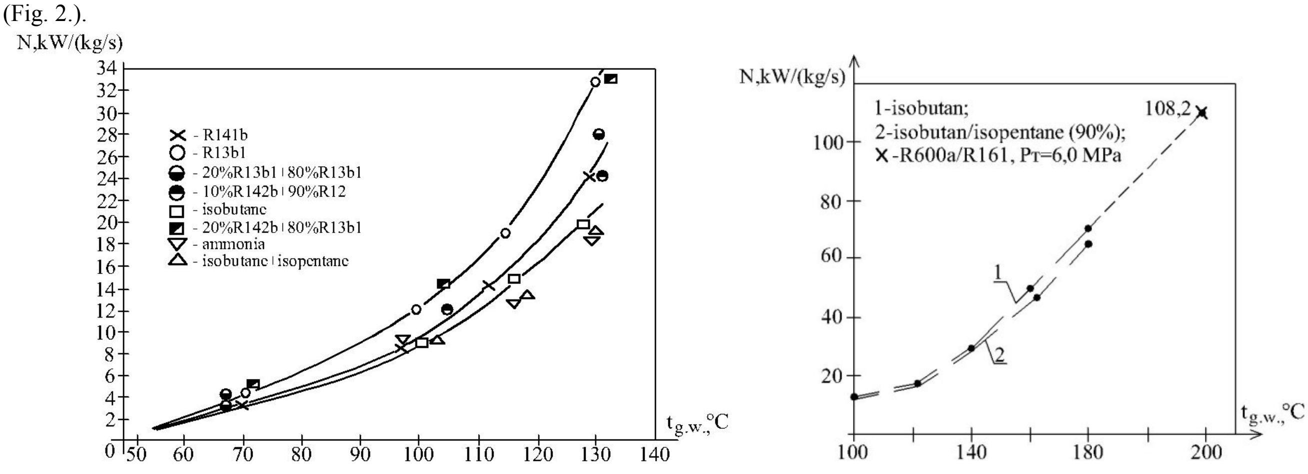

Fig. 2: Specific electric power of geothermal power station in the function of the geothermal fluid temperature

The value of the specific electric power at the temperature of geothermal fluid of 70°C for the being examined working media - Freons and their mixtures is about 3.2-3.3 kW/(kg/s), at temperature of 130°C it is (29.8-31.3) kW/(kg/s), at temperature of 200°C it is (64.0–108.4) kW/(kg/s). The results of calculations show that pressure values(P_e) and vapor temperature (t_e) of working medium in front of the turbine, flow rate of the working medium (m), the value of the minimum temperature difference (Δt_{min}), ambient temperature ($t_{a.}$) and other parameters influence obtaining the maximum specific electrical power.

The effect of minimum temperature difference is the most significant one. Thus, the decrease of ΔT_{min} from (10-15K) to (5-7K) allows to increase the production of electricity by 20-25%. The increase of ΔT_{min} leads to lowering evaporation pressure and temperature, which substantially reduces the efficiency and specific electrical power (N). Numerical results show that optimum temperature of evaporation of the working medium corresponds to each value of geothermal fluid temperature (Fig.3).

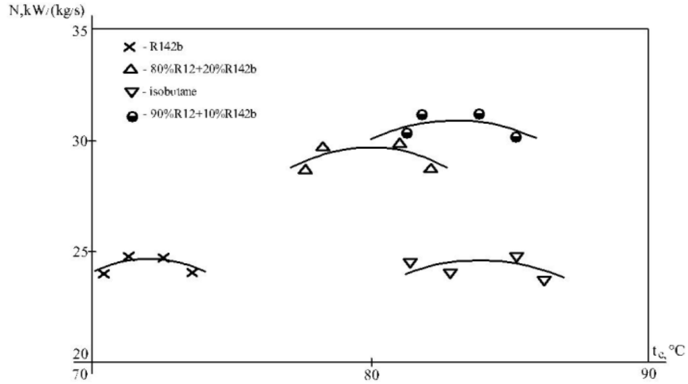

Fig.3: Specific electric power v.s. thermal evaporation

Therefore, when designing a geothermal system of heat- and power supply the search of optimal parameters of the binary power plant for a particular working medium should be done in each case.

Numerical calculation results show the promising use of environmentally safety Freons and their mixtures as a working medium of geothermal power stations. At geothermal fluid temperature of 70–130°C the specific electric power at the turbine shaft having Freon mixtures is 29-31 kW/(kg/s), which is by 10-12% higher than for the pure substance cycles 22-24 kW/(kg/s). Further improvement of thermodynamic cycle efficiency is possible by increasing cycle parameters to supercritical ones (for example, for R13b1 coolant specific electrical power of 34.0 kW/(kg/s) is received, and for R22-it is 3.37 kW/(kg/s), for R134a- it is 43. 87 kW/(kg/s) at temperature of geothermal fluid of 130°C).

To use mixtures of isobutane and isopentane as the working medium of the secondary circuit of a binary geothermal plant at supercritical parameters is offered in the works (Alkhasov, 2001; Ram, 1988). Using mixtures at supercritical parameters allows to control the thermodynamic parameters of the working medium changing the composition of the mixture, the nature of the geothermal fluid cooling curve being the same as the curve of heating the low temperature substance that allows to provide minimal value of the temperature difference between them. The present paper studies (R13b1+R142b),(R13b1+R13) Freon mixtures .Parameter calculation results of the supercritical cycle for Freon mixture (80%R131b+20%R142b), the temperature of the geothermal fluid being 130°, are shown in Table 2.

For variants of mixture (50% R13b1 + 50% R142b), (20% R13b1 + 80% R142b) the supercritical mode at pressure of 5.0 MPa is not realized due to the increase in the critical temperature and pressure of the mixture, expansion of steam in the turbine being completed in the two-phase region. Numerical results show that the specific electric power of the turbine is 34 kW/(kg/s).

The exergy efficiency analysis of geothermal power plant cycles is carried out on the basis of the first and second laws of thermodynamics for open stationary systems. Exergy values allow to define the maximum work (power) which can be obtained from the working medium with specified thermodynamic parameters in relation to the environment.

The utilization ratio of the power plant cycle, being defined as the ratio of the actual net power of the station to the maximum theoretical energy which can be obtained from the geothermal heat carrier is the following:

$$\eta_u = \frac{W}{m_{g.l}\left[\left(h_{g.l.} - h_o\right) - T_o\left(s_{g.l} - s_o\right)\right]} \tag{9}$$

where W - is power station capacity, kW; T_o – the temperature of environment, °C; $m_{g.l.}$ – the mass flow rate of geothermal liquid, kg/s; $h_{g.l}$, $S_{g.l}$–enthalpy and entropy of geothermal liquid of working parameters; h_o, S_o - enthalpy and entropy of geothermal liquid of temperature of environment.

The analysis of the results shows that the exergetic efficiency (gross) of the turbine is 92,99-95,16% for the various working mediums at the geothermal fluid temperature of 70°C, and the relative change in the turbine exergy is 3,24-9,37%. Thermal efficiency of the turbine is 3,45-3,90%. With increasing the temperature of the geothermal fluid up to 130°C, thermodynamic efficiency characteristics vary. Thus, the exergetic efficiency (gross) of the turbine is reduced to the values of 85,69-93,06%. The coefficient of utilization of geothermal power station cycle at a temperature of geothermal fluid of 70°C is about 50% and at temperature of 130°C it is 28,50-54,88%.

The most effective working media of geothermal power plant cycles at temperature of geothermal fluid of 130°C are R134a, R22, R143a, R218, R13b1, mixture (80% R13b1 + 20% R142b). Analyzing the thermodynamic efficiency of the power- and- heat exchangable geothermal power station equipment, it can be seen that the work of fluid compression in the pump for the subcritical cycle is from 0,84 to 1,83 kJ/kg, for the supercritical cycle it is from 2,20 to 3,59 kJ/kg. Pump capacity is increased from 0,92 to 7,82 kW. Specific exergy losses in the pump range from 1,33 to 13,03 kJ/kg and exergetic efficiency ranges from 84,50 to 93,75%. Exergetic losses in the pre-heater and evaporator increase and the value of exergetic efficiency (gross) decreases to 31,0-43,4%. Increased exergy losses of the working substance are also observed in the condenser due to significant difference in temperature (t_c–t_a). Relative change in exergy in the condenser is from 11,9 to 22,5%.

4. CONCLUSIONS

Numerical study results of thermodynamic efficiency of binary geothermal power plants have shown that generation of specific electric power of 34,0-43,3kW/(kg·s) is provided in cycles with R13b1, R22, R134a coolants at temperature of the geothermal fluid of 130°C. The influence of vapor pressure and temperature in front of the turbine, flow rate of the working medium, the minimum temperature difference value (Δt_{min}) in the heat exchange equipment (evaporator, condenser) is shown. The use of coolant mixtures provides the increase in generation of specific electric power by 10-12% and more. The analysis of the thermodynamic cycle and equipment efficiency of the geothermal power plant shows that at the temperature of the geothermal fluid ($t_{g.w.}$=130°C) thermal efficiency ranges from 7,58 to 10,96%, the coefficient of cycle utilization is from 28,50 to 54,88, at the temperature of geothermal fluid $t_{g.w}$ =200°C thermal efficiency is 15-16% for R600a/R141b, R600a/R161b mixtures, and the utilization coefficient increases up to 78,0-82,2%. Higher losses of exergy are shown to occur in the heat exchange equipment - evaporator, preheater, condenser. The total value of the exergy losses in the heat exchange equipment at Δt_{min} = 10K ranges from 45 to 55%.

Thus, in the cycles examined the thermal fluid temperature at the turbine outlet is from 63,2 to 86,2°C for R22 and R134a, from 28,1 to 91,1°C for R318, from 80 to 85°C for R600a/R141b mixtures that indicates the possibility of utilization η-coefficient increase in case of deeper decrease of geothermal fluid temperature by means of using cascading power unit in the turbine (Redko,2010).

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