GEOTHERMALISM AND GEOTHERMAL POWER PLANT

ROSCA M., COANDĂ S.
University of Oradea, Universităţii no.1, Oradea

Abstract - The paper is structured in six parts with introduction, conclusion and four main parts. In the introduction and the first part is evoked the importance of the topic and current concerns. In the second part are presented the calculation of state parameters. In the third part are presented the calculation of heat flows, the main part of present paper. The final part is about the selection of the main aggregates (equipment) of the cycle, and in the last part of the analysis the conclusions are presented.

INTRODUCTION

Geothermal energy is provided by the flow of heat that is transmitted from the Earth to the surface, amounting to about 0.05 W / mp. This flow of heat can be extracted from rocks – called petro-geothermal, or indirectly, from groundwater heated from rocks - in this case is called geothermal hydro energy.

The use of geothermal water for power generation is achieved through the expansion of the saturated steam in turbines, steam that may be obtained in three ways depending of the exit of geothermal water temperature: right from the geothermal well drilling, at high temperatures, through expansion of the overheated water, OR by water vaporization with a lower pressure or a lower boiling point liquid vaporization in electric centrals with secondary fluid

Though, the yields of geothermal energy used for electric energy production are generally quite low, around 10%, therefore usually is more suitable to use it in many domains for example in waterfall (Lindale diagram), after the production of electric power the remained heat to be used for the heating of the houses or for heat water production like in the stipulated case (see fig.1)
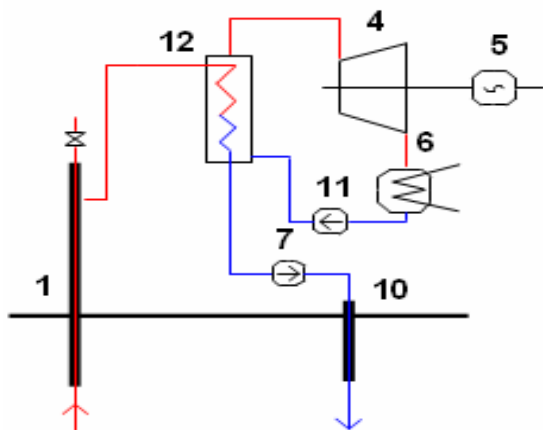

Fig.1. Example of electric energy production with secondary fluid, based on geothermal water

In order to produce electricity to such low temperatures, as it was shown above, used as substitutes for water, various organic substances, so that the Clausius-Rankine cycle gets the name Organic Rankine Cycle (ORC) The difference is given by the used organic substance features .They have a lot of advantages like:

- the organic substances are able to extract more heat than water from the geothermal fluid because at low evaporation temperatures the relation between the latent heat of evaporation and the mass heat capacity is lower, therefore the global efficiency is greater than water since at low vaporization temperatures, the ratio between the latent vaporization heat and the mass heat capacity is lower, therefore the global output – proportionally viewed through the output wattage – is higher.

-at the turbine exit, the organic fluid is in a state of overheated vapours not like the state of water which is to be found as moist saturated vapours. Therefore it appears not the danger of turbine blades erosion, and part of the vapours heat can be recovered by liquid preheating between the exit of the pump and the entry into the evaporator

- the enthalpy drop allows the use of single-stage turbines, with high efficiency and low stresses, so cheaper

- at the same power, the mass flow rate of the organic fluid is higher than that of the water, but the installation dimensions are smaller due to the fact that vapour density is greater and the volume flow is smaller

- we may use under 100 $^{\circ}$C temperatures, using with efficiency the heat from the geothermal liquid.

In our case **we must calculate the characteristics of a power generation system based on a heat flow from a geothermal well drilling water extraction with the following features:** the artesian flow is 30 kg/s, the admission temperature in the geothermal heat exchanger is 85°C, and the exit temperature is 55 $^{\circ}$C, the cooling water temperature is 15 $^{\circ}$C.There are many refrigerants that are well suited in specific cases, in this case using butane (noted also n-butane), or R-600 in the code of the refrigerants with the following features: gas at the normal temperature and pressure wit formula C_4H_{10}, inflammable, colourless; density of 2,48 kg/m^3, at 15°C and 1atm, or 600 kg/m^3-as liquid, at 0°C and 1atm; evaporation temperature of -0,5°C (272,6 °K), molar mass 58,12 g/mol; auto-ignition temperature of 500°C

In the following, we shall describe the Clausius-Rankine cycle and the states of the chosen organic fluid – the butane- passes, as well as the calculation of the heat flow values extracted from the geothermal water..

2. DESCRIPTION OF THE CLAUSIUS - RANKINE CYCLE

Clausius Rankine cycle is a thermodynamic cycle engine, which produces mechanical work, based on heat input in cycle from a source. On the basis of this cycle occurs over 80% of the electricity produced in the whole world. This cycle may be biphasic or overcritical. In this work I coped with the biphasic cycle, for practical reasons to deal with a lower pressure area
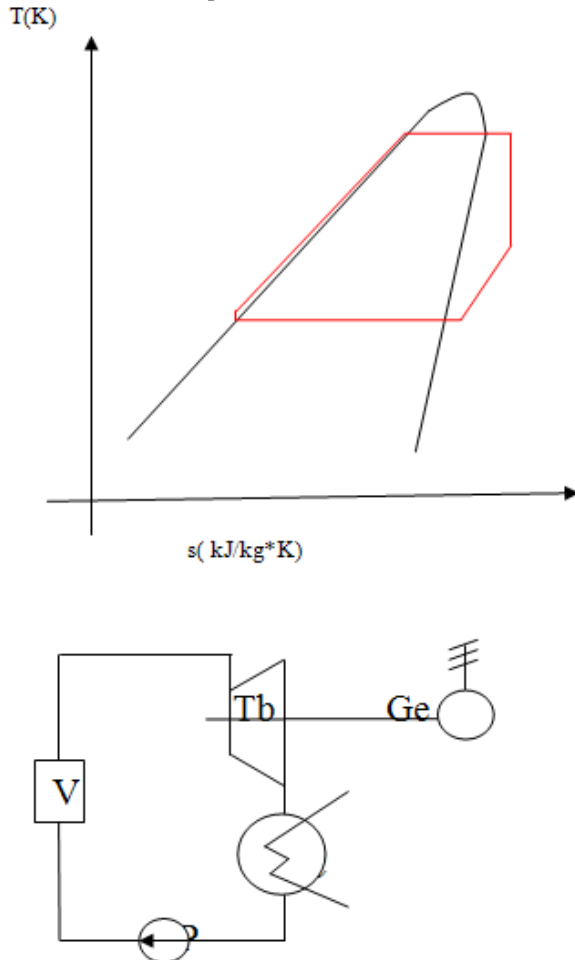

Fig. 2. The basic components and Clausius Rankine cycle to produce EE

In the diagram of the cycle, the coloured red area means the useful cycle mechanical work , the area under the yellow zone, until s axis, means the environment given heat, and the sum of both surfaces gives the cycle introduced heat value:

The four main components of the cycle are: (see draw 2):

- *turbine* - where the steam of the working substance suffers an adiabatic and isentropic expansion, from the state 1, considered with the highest parameters, at the state 2, producing useful mechanical work; *the cooler*- where the steam condenses itself , becoming a liquid , isobaric, from state 2 to state 3, where is to be found as saturate liquid ; *the pump*- where the liquid that leaves the cooler at a little pressure is adiabatic compressed at a great la pressure, passing from status 3 to

status 4; *-evaporator-* where high pressure but low temperature entered liquid heats itself till vaporization temperature, vaporizes until the dry saturated vapour, and then overheat itself till state 1 parameters. At real Clausius-Rankine cycle, differences from the ideal cycle consist in the fact that some processes are not reversible due to the losses, thus the expansion in turbine is not strictly isentropic; the compression in pumps suffers some losses; pressure drops in heat exchangers; etc. The Clausius-Rankine cycle with organic fluid has some differences compared the one with the water, especially given by the fluid nature, which has a vapour curve of different shape, allowing expansion to lower temperatures without the risk given by the turbine blades wear, following the appearance of condensation drops at the end of expansion.

The cycle described below is calculated for the selected organic fluid, the butane and is used without overheating, preferring the expansion direct from the exit state from evaporator, the pressure decrease allowing its quick evaporation

3. THE CALCULATION OF STATE PARAMETERS

The state parameters are measured for each of the four states :

1. at the evaporator exit and at the entry in turbine

2. turbine exit and at the entry in condenser

3. condenser exit and pump entry

4. pump exit and evaporator entry

The four states are characterized by p,t,v,i,s parameters, each with its own state. Some of states own more parameters according to the calculation request:
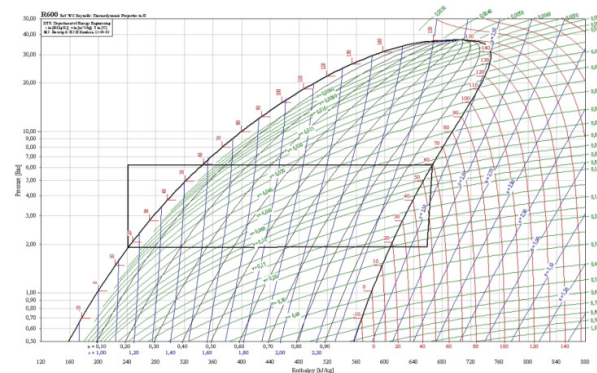

Fig. 3. The p-h diagram for butane is drawn in figure 4 (R-600)

The four transformations are simple transformations, 2 isobar and 2 adiabatic, in the ideal case,as follows:

I.) $1 – 2$ -adiabatic expansion (isentropic) in turbine: $s_1 = s_2$

II.) $2 – 3$–isobaric condensation in condenser: $p_2 = p_3$

III.) $3 – 4$ - adiabatic compression (isentropic) in pump: $s_3 = s_4$

IV.) $4 – 1$. Isobaric heating and evaporation in evaporator: $p_4 = p_1$

There are also isothermal transformations this means at the same temperature where butane the organic fluid suffers also a phase change, it passes from gas to liquid or from liquid to gas . Thus, we can describe the points that represent the respective states, keeping in mind that in the first exchange heater (evaporator), the geothermal water gets in with 85°C and gets out with 55°C, and the secondary fluid gets in with 18.18°C (ideal with 18°C) and gets out with 59.3°C, but in the secondary heat exchanger (the condenser), the cooling water gets in with 15°C and gets out with 27°C , and the butane gets in with the exit of turbine temperature (24.12°C) and gets out with 18°C, The butane will be warmer with a few tenths of degree when compressed in pump (we shall calculate this). The state parameters values were obtained with the help of Coolpack program simulation of butane Clausius-Rankine cycle, at limits of temperature values as described above.

The parameters are presented below.

– 1 - at the exit of evaporator and at the entry turbine, is determined by the following parameters:

-temperature t_1=59.3°C

-gas entropy s_1= 2.44973 kJ/(kg K)

-fluid title x_1 =1

- gas enthalpy h_1=668.492 kJ/kg

-gas specific volume v_1=0.0646785 m^3/kg, taken from tables,

-pressure p_1=6.2798 bar, the pressure where the butane vaporizes at 59.3 °C, taken from tables

-c_{pb}= 2.519 kJ/(kg K)-specific jet mass at average temperature, average value of 38,74°C (between 18.179 and 59.3 °C);

- 2 - at the exit of turbine and at the entry in condenser is defined by the following parameters:

-temperature t_{2i}=24.12°C

-pressure p_{2i}= 1.9493 bar, taken from tables,

-gas specific volume v_{2i}= 0.204945 m^3/kg, also taken from tables,

- gas enthalpy h_{2i}=621.006 kJ/kg

-pressure p_1=6.2798 bar, the pressure where the butane vaporizes at 59.3 °C, taken from tables

-c_{pb}= 2.519 kJ/(kg K)-specific jet mass at average temperature, average value of 38,74°C (between 18.179 and 59.3 °C)

- entropy gas s_{2i}= 2,41284 kJ/(kg K)

- fluid state –gas- dried vapour

- latent heat of condensation at 18°C r_2 = 365.01 kJ/kg

- 3 - at the exit of condenser and at the entry in pump, is defined by the following parameters:

-temperature t_3=18°C

-pressure p_3= 1.9493 bar, from tables, for x_3 =0 -specific

volume liquid v_3= 0.001721 m^3/kg, from tables,

-pressure p_1=6.2798 bar, the pressure where the butane vaporizes at 59.3 °C, taken from tables

-c_{pb}= 2.519 kJ/(kg K)-specific jet mass at average temperature, average value of 38,74°C (between 18.179 and 59.3 °C)

-liquid enthalpy h_3 =242.290 kJ/kg

-entropy liquid s_3= 1.14936 kJ/(kg K)

- fluid title x_3 =0, saturated liquid

- 4 - at the exit of pump and at the entry in evaporator is defined by the following parameters:

-temperature t_{4i}=18.179°C

-pressure p_{4i}= 6.2798 bar, the pressure where the butane vaporizes 59.3 °C, from tables,

- specific volume liquid v_{4i}= 0.00172 m^3/kg, taken from tables-liquid enthalpy h_{4i}=243.035 kJ/kg

-liquid enthalpy h_{4i}=243.035 kJ/kg

-liquid entropy s_{4i}=1.1494 kJ/(kg K)

- the fluid state – sub-cooled liquid

- latent heat of evaporation at 59.3°C,r_1 =321.87 kJ/kg.

In the **real** cycle, there are some differences which modify the efficiency and the size of the equipments, if the cycle remains between the same temperature values.

4. THE CALCULATION OF HEAT FLOWS

The calculation of the values of heat flows circulated through this cycle, it will be taken into account mainly by the flow of heat introduced by geothermal water

$Q_{ag} = m_{ag} c_p \Delta t = 30 \cdot 4.185 \cdot (85-55) = 3766.5$ kW

Considering the heat exchanger without great losses, and c_{pb}= 2.519 kJ/(kg K), at average value of 38.74°C (between 18.179 and 59.3 °C), the heat taken by butane in evaporator is :

$Q_{vb} = 3766.5 = m_b c_{pb} \Delta t + m_b r_1$ or

$m_b = 3766.5$ / $(c_{pb} \Delta t + r_1)$ = 3766.5 / [2.519 ·(59.3 -18.179) +321.87] = 8.853 kg/s

If we take in consideration the energy balance for the mentioned cycle we may write foe every kg of fluid, **the ideal cycle**:

I) The adiabatic and isentropic transformation between the states 1 and 2, the expansion in turbine:

- q_{12} = 0 (adiabatic), Δs_{12i} = 0, $s_1 = s_{2i}$ = 2.44973 kJ/(kg K) (isentropic),

- the useful mechanical work: $l_{12i} = h_1 – h_{2i}$ = 668.492 – 621.006 = 47.486 kJ/kg, so for every kg of fluid we obtain 47.486 kJ/kg, so totally :$L_{12i} = m_b \cdot l_{12i}$ = 8.85266 · 47.4856 = 420.374 kW

- pressure difference - turbine pressure drop::

$\Delta p_{12i}=p_2 – p_1$=1.9493 -6.2798 = - 4.3305 bar, expansion

-temperature difference: $\Delta t_{12i}= t_{2i}– t_1$ =24.1212 – 59.304= - 35.183°C cooling

-volume difference: $\Delta v_{12i}=v_{2i} – v_1$=0.204945 – 0.064678 = 0.14027 m^3/kg volume increase

-aggregation state -gas, even at the end of expansion;

II) The isobaric transformation between states 2 and 3, the cooling and condensation in condenser:

$p_2 = p_3$=1.9493 bar, taken from tables,

$\Delta s_{23i} = s_3 - s_{2i}$ = 1.14945 – 2.41284 = - 1.26349 kJ/(kg ·K),entropy gain

$q_{23i} = h_3-h_{2i} = 242.29 - 621.006 = -378.716$ kJ/kg, so at every kg of secondary fluid it has to be extracted 367.865 kJ/kg of heat, so totally:

$Q_{23i} = m_b \cdot q_{23i} = 8.85266 \cdot (-367.864) = -3352.646$ kW

- the mechanical work during this process is 0 , $l_{23} = 0$

- difference of temperature: $t_{23i} = (t_3 - t_{2i}) = (18 - 24.1212) = -6.1212°C$ cooling

- difference of volume: $\Delta v_{2i3} = v_3 - v_{2i} = 0.0017215 - 0.2049450 = -0.203224$ m^3/kg volume compression

-aggregation state–from gas, through cooling it becomes at transformation end a liquid with title x= 0;

III) adiabatic transformation, compression in pump between states 3 and 4:

adiabatic= $q_{34} = 0$,

$\Delta s_{34i} = 0$, $s_3 = s_{4i} = 1.481564$ kJ/(kg K)-isentropic

- the consumed mechanical work will be:

$l_{34i} = h_3 - h_{4i} = 242.29 - 243.035 = -0.745$ kJ/kg

so for all the butane quantity, the consumed mechanical work in pump in ideal case will be :

$L_{34i} = m_b \cdot l_{34i} = 8.85266 \cdot (-0.74518) = -6.597$ kW

- difference of pressure - pump pressure growth:

$\Delta p_{34i} = p_{4i} - p_3 = 6.2798 - 1.9493 = 4.3305$ bar, compression

- temperature difference: $\Delta t_{34i} = t_{4i} - t_3 = 18,1785 - 18 = 0,179°C$

- volume difference : $\Delta v_{34i} = v_{4i} - v_3 = 0.00172002 - 0.00172149 = -0.0000015$ m^3/kg Compression of the volume, negligible;

-aggregation state -liquid;

IV) isobaric transformation, heating and evaporation in the first heat exchanger (the evaporator), between states 1 and 4:

$p_4 = p_1 = 6.2798$ bar, given by the butane evaporation pressure at 59.3 °C, taken from the tables,

$\Delta s_{14i} = s_1 - s_{4i} = 2.44973 - 1.14936 = 1.300373$ kJ/ (kg ·K),

the mechanical work during this process is 0 , $l_{41} = 0$,

$q_{41i} = h_1 - h_{4i} = 668.492 - 243.035 = 425.456$ kJ/kg so for every kg of fluid the system gets 425.456 kJ/kg of heat, totally:

$Q_{41i} = m_b \cdot q_{41i} = 8.85266 \cdot 425.456 = 3766.42$ kW

-temperature difference: $\Delta t_{4i1} = t_1 - t_{4i} = 59.3 - 18.1785 = 41.128$ °C heating

-volume difference: $\Delta v_{4i1} = v_1 - v_{4i} = 0.0646785 - 0.00172 = 0.06296$ m^3/kg volume expansion;

-aggregation state - liquid at the beginning of transformation it becomes saturate vapours with title x = 1 at the heating end.

The thermodynamic efficiency of the cycle, concerning mechanical work production will be:

$$\eta_t = \frac{E_u}{E_c} = \frac{L_c}{Q_{41}} = \frac{L_{12}+L_{34}}{Q_{41}} = \frac{l_{12}+l_{34}}{q_{41}} = \frac{(h_1-h_2)+(h_3-h_4)}{h_1-h_4} = 1 - \frac{h_2-h_3}{h_1-h_4}$$

with E_u – effective energy and E_c –consumed energy,so:

$\eta_{ti} = 1 - (621.006 - 242.290) / (668.492 - 243.035) = 1 - 378.716/425.457 = 0.1099$ or rounded **11%**.

If we take in consideration the energy of the cycle, we may write for every kg of fluid the **real cycle**

Fluid flow: $Q_{vb} = 3766.5 = m_b \cdot c_{pfmed} \Delta t + m_b r_1$

Or $m_b = 3766.5 / (c_{pfmed} \Delta t + r_1) = 3766.5 / [2.518945 \cdot (59.3-18.257) + 321.887] = 8.857$ kg/s.

For every kg of fluid, **the real cycle:**

I)adiabatic and isentropic transformation theory between states 1 and 2, the expansion in turbine

in the real transformation: $q_{12} = 0$ (adiabatic), we may consider it adiabatic, due to the fact that the exterior heat transfer for every kilo of butane is negligible in the expansion short time .The transformation is no more isentropic due to the fact that it is irreversible, so we will have a loss of entropy:

$(s_{2r} - s_{2i}) = 2.47353 - 2.44971 = 0.02382$ kJ/(kg K).

$l_{12r} = l_{12i} \cdot \eta_{tb} = 47.465 \cdot 0.85 = 40.345$ kJ/kg, considering an average efficiency of 85% of the turbine,

$l_{12r} = h_1 - h_{2r}$, then $h_{2r} = h_1 - l_{12r} = 668.475 - 40.346 = 628.129$ kJ/kg, that is in case of real expansion of every kilo of fluid we attain the state 2 with $h_{2r} = 628.129$ kJ/kg,, so per total :

$L_{12r} = m_b \cdot l_{12r} = 8.857 \cdot 40.346 = 357.346$ kW every second, so with 63.03 kW less (the real mechanical work is 85% from the ideal one) than in ideal case

The state 2_r parameters will be obtained with the help of the diagrams or of the Coolpack program where we may introduce h_{2r} and p_{2r} we shall obtain also the other parameters

$t_{2r} = 28.111$ °C, $p_{2r} = p_{2i} = p_3 = 1.9493$ bar;, $v_{2r} = 0.2083383$ mc/kg,

$s_{2r} = 2.47353$ kJ/(kg ·K), $h_{2r} = 628.13$ kJ/kg

- the loss of entropy is $\Delta s_{12r} = s_{2r} - s_1 = 2.47353 - 2.44971 = 0.02382$ kJ/(kg ·K),

- the pressure difference – the loss of the pressure on turbine: $\Delta p_{12r} = p_{2r} - p_1 = 1.9493 - 6.278 = -4.33$ bar , expansion,

-- the difference of temperature: $\Delta t_{12r} = t_{2r} - t_1 = 28.111 - 59.292 = -31.181$ °C cooling

- the volume difference : $\Delta v_{12r} = v_{2r} - v_1 = 0.20834 - 0.064698 = 0.1436$ m^3/kg volume growth;

II) the isobaric transformation between states 2_r and 3, cooling and condensing in condenser:

$p_{2r} = p_3 = 1.9493$ bar, $\Delta s_{2r3} = s_3 - s_{2r} = 1.14936 - 2.47353 = -1.3242$ kJ/(kg ·K),

$q_{2r3} = h_3 - h_{2r} = 242.29 - 628.13 = -385.84$ kJ/kg, for every kilo of secondary fluid we must extract the heat of 385.84 kJ/kg, so per total:

$Q_{2r3} = m_b \cdot q_{2r3} = 8.857 \cdot (-385.84) = -3417.4$ kW , a plus of 64.76 kW of heating which has to be removed from the process, which allows us to dimension a greater heat exchanger surface, when the butane parameters are to be kept in state 3 when it leaves the condenser;

the mechanical work during this process is 0, $l_{23} = 0$;

-the difference of temperature: $\Delta t_{23r} = (t_3 - t_{2r}) = (18 - 28.111) = -10.111$ °C cooling

-the difference of volume: $\Delta v_{2r3} = v_3 - v_{2r} = 0.0017215 - 0.20834 = -0.20662$ m^3/kg volume compression

III) Adiabatic transformation, compression in pump between states 3 and 4:

-adiabatic $= q_{34} = 0$, we may consider it adiabatic , due to the fact that the exterior heat transfer for every kilo of butane is negligible in the pump compression short time

$\Delta p_{34r} = p_{4r} - p_3 = 6.278 - 1.9493 = 4.33$ bar, pump pressure, for which the consumed mechanical work is:

$l_{34i} = h_{3i} - h_{4\ i} = 242.29 - 243.035 = - 0.745$ kJ/kg, supposing the pump efficiency of approximately 80% we shall have:

$l_{34r} = l_{34i}\cdot(100/80) = - 0.745\cdot(100/80) = -0.9314$ kJ/kg real, so

$L_{34r} = m_b \cdot l_{34r} = 8.857 \cdot(- 0.9313) = - 8.25$ kW, with 1.653 kW more than the ideal cycle (difference gives the pump efficiency)

$l_{34r} = h_3 - h_{4\ r}$ then $h_{4r} = h_3 - l_{34r} = 242.29 + 0.9314 = 243.221$ kJ/kg, taking this value and the one of the pressure $p_{4r} = p_1 = 6.2798$ bar, given by the butane evaporation pressure at 52.292 °C, taken from tables, we will obtain also from tables the values of the other parameters which describe state 4_r :

$t_{4r} = 18.257$ °C, $p_{4r} = p_{4i} = p_1 = 6.278$ bar, $v_{4r} = 0.0017203$ mc/kg, $s_{4r} = 1.145$ kJ/ (kg ·K), $h_{4\ r} = 243.221$ kJ/kg,

$\Delta s_{34r} = s_{4r} - s_3 = 1.145 - 1.1494 = 0.0044$ kJ/(kg ·K), entropy increase

-temperature difference $\Delta t_{34r} = t_{4r} - t_3 = 0.292$ °C, heating in pump

-volume difference $\Delta v_{34r} = v_{4r} - v_3 = 0.00172028 - 0.001721491 = -0.00000121$ m^3/kg, compression in pump

IV) isobaric transformation, heating and evaporation in the first heat exchanger, (evaporator), between states 4 and 1:

$p_4 = p_{1.} = 6.278$ bar, the mechanical work during this process is 0 $l_{41} = 0$,

$\Delta s_{4r1} = s_{4r} - s_1 = 1.149996 - 2.449714 = -1.29972$ kJ/(kg ·K), the increase of entropy through temperature growth and change of the phase,

$q_{4r1} = h_{4r} - h_1 - = 243.221 - 668.475 = - 425.253$ kJ/kg this means for every kilo of fluid the system gets from outside with the geothermal water 425.253 kJ/kg of heating so overall: $Q_{14r} = m_b \cdot q_{41r} = 8.857 \cdot 425.253 = 3766.5$ kW, needed heating covered by the heating flow introduced in system with the geothermal water as calculated at the beginning of this chapter:

$Q_{ag} = m_{ag} c_p \Delta t = 30 \cdot 4.185 \cdot 30 = 3766.5$ kW

-the difference of temperature: $\Delta t_{4r1} = t_1 - t_{4r} = 59.232 - 18.257 = 41.035$°C heating

- volume difference: $\Delta v_{4r1} = v_{4r} - v_1 = 0.0647 - 0.00172 = 0.06298$ m^3/kg volume expansion;

The cycle thermodynamic efficiency concerning mechanical work production will be:

$$\eta = \frac{E_u}{E_c} = \frac{L_t}{Q_{41}} = \frac{L_{12} + L_{34}}{Q_{41}} = \frac{l_{12} + l_{34}}{q_{41}} = \frac{(h_1 - h_2) + (h_3 - h_4)}{h_1 - h_4} = 1 - \frac{h_2 - h_3}{h_1 - h_4}$$

with E_u – effective energy and E_c – consumed energy, so: $\eta_{tr} = 1 - (628.13 - 242.29)/ (668.475 - 243.221) = 0.0927$ or **9.27%** , a loss in efficiency of approximately **1.79 %**

as related to the ideal case, including pressure ratio of approximately 0.3105.

$\eta_{ti} = 11\%$, $\eta_{tr} = 9.27\%$ then $\Delta\eta(i\text{-}r) = - 1.78\%$.

The ideal cycle has been drawn in - Schema T-s for butane -R600. **It was synthetically presented, in tables** the ideal and real cycles states and transformations during the EE output process with butane in the table below:

No state	Enthalpy h(kJ/kg)	Entropy s(kJ/kgK)	Pressure p (bar)	Temperature t(°C)	Specific volume v (mc/kg)
1	668.475	2.4497	6.278	59.292	0.064698
Δ(1-2$_i$)	l_{12i}= 47.486	0(isentropic)	-8,618	Δt_{12i}=-52,98	Δv_{12i}=0.20630
Δ(1-2$_r$)	l_{12r}= 40.345	Δs_{12r} =0.0335	-8,618	Δt_{12r}=-47,20	Δv_{12r}=0.08648
Δ1-2(i-r)	Δl(i-r)= -7.141	Δs(i-r)= 0.0351	0	Δt(i-r)=5.78	Δv(i-r)=-0.11982
2$_i$	621.006	2.4497	1.9493	24.12	0.20495
2$_r$	628.129	2.47353	1.9493	28.111	0.20834
Δ2(i-r)	2$_r$ -2$_i$ =7.123	2$_r$ -2$_i$ =0.02383	2$_r$ -2$_i$ =0	2$_r$ -2$_i$ = 3.991	2$_r$ -2$_i$ =-0.0339
Δ(2$_i$-3)	q_{23i}= -378.716	Δs_{23i} = -1.3275	0(isobar)	Δt_{23i}= -6.12	Δv_{23i}= -0.203223
Δ(2$_r$-3)	q_{23r}= -385.839	Δs_{23r} = -1.3242	0	Δt_{23r}= -10.111	Δv_{23r}=-0.20662
Δ2-3(i-r)	Δq_{23}(i-r)= -7.123	Δs =- 0.0033	0	Δt(i-r)=3,991	Δv(i-r)=0.003397
3	242.29	1.14936	1.9493	18	0.0017215
Δ(3-4$_i$)	l_{34i}= - 0.687	0(isentropic)	6.278	Δt_{34i}=0	Δv_{34i}=-0.000003
Δ(3-4$_r$)	l_{34r}=-0.916	Δs_{34r} =0.0335	6.278	Δt_{34r}=0.087	Δv_{34r}=-0.000010
Δ3-4(i-r)	Δl_{34}(i-r)=-0.229	Δs(i-r)=0.0335	0	Δt(i-r)=0.087	Δv(i-r)=-0.0000052
4$_i$	243.035	1.14936	6.278	18.1685	0.00172
4$_r$	243.221	1.149996	6.278	18.2565	0.0018874
Δ4(i-r)	4$_r$ -4$_i$ =9.186	4$_r$ -4$_i$ =0.00064	4$_r$ -4$_i$ =0	4$_r$ -4$_i$ =0.088	4$_r$ -4$_i$ =0.00016
Δ(4$_i$-1)	q_{4i1}=425.44	Δs_{4i1} =1.30034	0(isobar)	Δt_{4i1}=-41.1235	Δv_{4i1}= -0.06298
Δ(4$_r$-1)	q_{4r1}=425.25	Δs_{4r1} =1.299704	0	Δt_{4r1}=-41.0355	Δv_{4r1}=-0.062811
Δ4-1(i-r)	Δq_{41}(i-r)=-0.19	Δs(i-r)=0.00064	0	Δt(i-r)=-0.088	Δv(i-r)= -0.00017

5. SELECTION OF THE MAIN AGGREGATES (EQUIPMENT) OF THE CYCLE

I Vaporizers selection: The selection of the vaporizers has been performed by specialized software: **HEX calc – DANFOSS HEAT EXCHENGER CALCULATION TOOLS.** In order to establish the data input in the selection software of heat exchangers it has been considered the parameters of real state of the circulated cooling agent. The installed capacity for the group of 2 paralleled vaporizers has been calculated based on the primary agent flow – the geothermal water at a temperature of 85ºC available:

$Q_{nec.bat.SCP\text{-}VAPORIZ}$. = (m_{ag} $c_p \Delta t$)/η_{expl}. = [30 · 4.185 · (85-55)]/0.95 = 3,964 kW~4,000 kW = 4MW; therefore, it will be selected 2 SCPs of 2 MW

each. The output of 95% (5% loss) mirrors, but also covers the real working.

It has been selected two identical paralleled vaporizers (battery) of SCP- DANFOSS type: XG 50-1 with 80 plates each with an installed thermal capacity of 2 MW/item (2000 kW/item). We show within the Annex 7 an excerpt of the vaporizers selection program process.

The exit temperature of the geothermal water of the SCPs during real working is lower than the temperature initially considered (30.2°C as for 55°C) since we introduced firmly the parameters on the secondary agent. The fact that the geothermal water is disposed at such a temperature it fails to influence the thermodynamic cycle parameters of the secondary agent and therefore its energetic performances.

II) SELECTION OF CONDENSERS The selection of condensers has been carried out with the same software as for the vaporizers. The installed capacity for the group of 4 paralleled condensers has been calculated based on the secondary agent flow (Freon R600) established within the real conditions of the cycle:

$Q_{2r3} = m_b \cdot q_{2r3} = 8.857 \cdot (-385.839) = - 3417.38$ kW

Difference of temperature: $\Delta t_{23r} = (t_3 - t_{2r}) = (18 - 28.111) = -10.111°C$ cooling

$\mathbf{Q_{nec.bat.SCP-CONDENS}}$. = (m_{ag} $c_p\Delta t)/\eta_{expl}$. = $[8.857 \cdot 38.16 \cdot (28.111-18)]/0.95 = 3417.38$ kW $/0.95 = 3597.2$ kW~3,500 kW = 3.5 MW;

thereupon, it will be selected 4 SCPSs of 0.875 MW each. The output of 95% (5% loss) mirrors but also covers the real working.

The exit temperature of the cooling water (secondary agent) from the SCPs during real working is higher than the temperature initially considered (24.9°C as for 24°C). It has been selected 4 (4 items) identical paralleled (battery) condensers of SCP- DANFOSS type: **XG 50-1 with 180 plates** each (but with great extension possibilities, as appropriate) with thermal installed capacity of 0.875 MW/item (875 kW/item).

III) Selection of pumps for circulating the primary agent for vaporizers (geothermal water)

The circulation pumps will be selected with varable speed (then during operation with variable flows and pressures based on the thermodynamic cycle requirements). The pumps will be selected by specialized software made by a specialized producer. We firmly specified the input parameters in the selection which in our case are as follows:

- pumped agent: GEOTHERMAL WATER AT 85°C;

- pumps flow: 31 l/s = 111.6 mc/h;

- pumping height (bar or in this case meter of liquid column = H2O column meter)

$P_{nec} = \Delta p_{SCP-PRIMARY\ VAPORIZER}$ + 1.25 $\Delta p_{LINEAR\ (PATH\ SCP-PT)} = 2.05$ m H2O column

Selection of the pumps or of the pumping group provided with variable speed pumps was carried out with specialized software: **WinCAPS** made by the provider GRUNFOS.

As a result of the software operation, it recommends a double pump of:

TPED 150-130/4-S -F-A BAQE type (in fact the equipment includes two pumps with variable speed and frequency converter). We show few features of the pump obtained following the selection program):

Product name TPED 150-130/4-S

Phases	U [V]	P2	Backwater	p max	Q real	H real
3	380	2x 7.5	DN 150	16	112	2.05

IV) SELECTION OF THE CIRCULATING PUMPS A R600 –n – BUTANE

The circulation pumps of the R 600 – n butane AGENT differs as selection as related to the previous case by the fact that the circulated liquid is not the water, but the butane, and upon selection the physical parameters need to be determined (kinetic tack, density at the pumping temperature:18-19°C) of the circulated agent.

The pumps (the pumping group) are selected with the same specialized software; the input parameters are as follows:

- pumped agent: R 600 – n – butane at 18-19°C;

- pumps flow: 54.90mc/h ; (8.857Kg/s) at the density of 581.13 kg/mc at the temperature of 18.5°C;

- the pumping height (bar or in this case meter liquid column = n-butane meter column) which is determined as follows :

P_{nec} = $(p_{VAPORIZER} - p_{CONDENSER}) +$ $\Delta p_{SCP-\ SECONDARY\ VAPORIZER}$ + $\Delta p_{SCP\ -\ PRIMARY\ CONDENSER}$ $+\Delta p_{LINEAR(PATH)}$ $+\Delta p_{LOCAL(PATH)}$ = $P_{nec} = 4.75$ bar;

We specify that we set for selection a pumping group (module) with four pumps with INOX steel body and runner with variable speed. As a result of the software operation, it recommends a pumping group of:

Hydro MPC-E 4 CRE32-3 50 Hz type

The final outcome with explanation of the parameters (choose/selection of geothermal water circulating pumps – primary vaporizers):

Product name Hydro MPC-E 4 CRE32-3 50 Hz type

Phases	U [V]	P2[kW]	Backwater	p max [bar]
3	380	4x5.5	DN 150	16

Q real[mc/h]	H real[m]	Energy [kWh/an]	EUR/an]
54.9	47.5	57735	7910

V) Selection of the circulating pumps of the cooling water (secondary agent) of condensers

The circulating pumps of CODENSERS COOLING WATER – SECONDARY AGENT – are chosen in the same manner as the previous case, the only difference being the nature of the pumped fluid: clean conventional water at 15 °C and the fact that the pumping group needs to be placed near the source – surface water flow (and not near the condensers), residing a difference in geodesic quota which will appear in the total need of pressure that the pumping group needs to achieve. The pumps (the pumping group) are selected with the same specialized software; the selection input parameters are as follows:

- pumped agent: clean conventional water at 15 °C;

- pumps flow: 324.4 mc/h;[4x875 kW/4.185(24.3-15)]/0.998=90.11 l/s

- pumping height (bar or in this case meter liquid column = H2O meter column) = 1.88 bar;

a result of the software operation, it recommends a pumping group of: Hydro MPC-E 6 CRE 45-2-2 type (In fact the setting includes six paralleled pumps with variable speed with frequency converter included). We show here few features of the resulted pump while operating the selection software:

FINAL OUTCOME WITH EXPLANATION OF PARAMETERS (CHOOSE/SELECTION OF CONDENSERS COOLING WATER CIRCULATING PUMPS):

Product name	Hydro MPC-E 6 CRE45-2-2			
Phases	**U [V]**	**P2[kW]**	**Backwater**	**p max [bar]**
3	380	6x7.5	DN 200	16
Q real[mc/h]		**H real[m]**		**Energy [kWh/an]**
324		18.8		243.528

VI)   Selection of turbine and generator:

The selection of this equipment is different and more difficult than the case of the equipment previously chosen, as the providers' tender is limited because of the complexity of the equipment and a strict specialization of the activity. Practically, in order to use such an aggregate it will be drawn a design theme for a certain producer who based on the requested parameters within the theme will design a dedicated equipment for the wanted application. For the hereby didactic project we found at the manufacturer: KAWASACHI – A SETTING MADE OF A GAS TURBINE CLUTCHED TO AN ELECTRIC GENERATOR: **GROUP GAS TURBINE (VAPORS R600)-GENERATOR POWER SUPPLY – KAWASACHI type; - encased;**

INSTALLED THERMAL CAPACITY. TURBINE :**Ptt =0.5 MW(500 kW)**;

OUTPUT   ENERGY turbina: **PGE.=0.4 MW(400kW)**.

6.CONCLUSIONS

Within the process of power output through geothermal water (primary source), selection of the working fluid (Freon) represents one of the most relevant features, taking into account the temperature limitations entailed by the relatively low temperature of the hot source and especially the small gap between the temperatures. Provided that we consider the fact that the working fluid needs to fulfil certain requirements – as those of toxicity, safety, initial price, maintenance price, chemical balance, corrosive action etc. – the palette of working fluids narrows consistently.

Thus, of the substance remained to consider, upon the next step of the selection it may be taken into account for separation the extension output, the working pressures etc.

The output differences between the ideal and the real case, given by the practical limitations of thermal installations used and the irreversibility of transformations, will mirror in the final production of electric energy.

The situation of the primary heat exchanger dimensioning presents a feature, the working fluid warming in the pump, assisting the vaporization process, by diminishing, on a very small scale, the heat requirement, unlike the ideal case, where there is no heating going on, the process operating as a pre-warming of the working fluid.

Alike, part of the heat conceded to the cool source may be reused to pre-warm the butane at the exit of the pump, thus diminishing the required heat for butane vaporization and helping the increase of output and butane quantity, thus implicitly the final quantity of electric energy produced and delivered, and eventually a faster recovery of investments and profit increase.

In the drawings below is described a variant of geothermal power plant location and a trim line of her
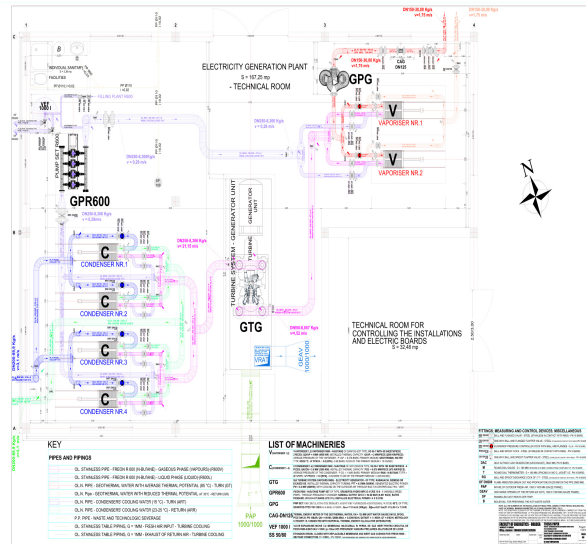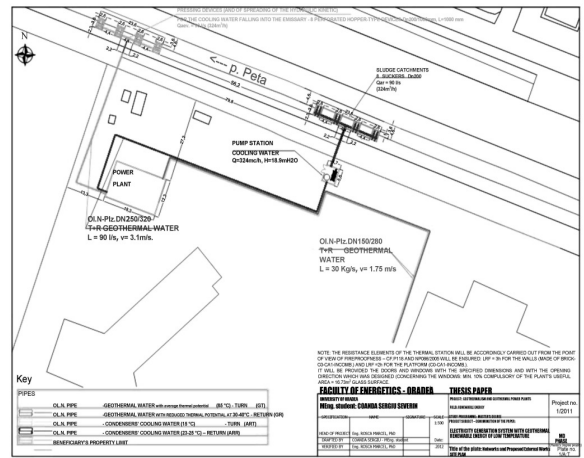
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