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Optimized geothermal binary power cycles

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The aim of this research is to study and recommend optimal Rankine cycles using Isobutane (R600a) and R134a as working fluids for two geothermal binary power machines. The first one (ORC machine A) should be able to generate electricity from low temperature geothermal resources, with profitable operation down to 65°C. The second one (ORC machine B) should be able to cogenerate both heat and power by heat recovery from the cooling water circuit, corresponding to geothermal fluids of 120-150ºC and cooling water supplying a district heating system at 60/80ºC. The main Rankine Cycle parameters and components are modelled, such as the shell and tube condenser and the geothermal plate heat exchanger. The objectives of the optimization are maximizing overall conversion efficiency and minimizing the cost of the plant, which is represented as minimizing the exchangers' surface. Through this research, a set of optimal solutions for ORC machines A and B are obtained that combine maximum plant's efficiency and minimum cost.

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1. Introduction

The main objectives of this research are to widen market perspectives of geothermal Rankine Cycle power generation by developing and demonstrating a unit that can generate electricity from low temperature geothermal resources, with temperature threshold for profitable operation at 65°C, compared with 90-100°C of existing units. In this paper this is called ORC machine A. A second perspective is to develop and demonstrate a Rankine Cycle machine for cogeneration of heat and power by heat recovery from the cooling water circuit. This will lead in cogeneration of heat and power from Rankine Cycle units in present and future geothermal district heating schemes. In this paper this is called ORC machine B. This corresponds to studying and recommending optimal Rankine cycles for the two geothermal binary power machines mentioned above. Isobutane (R600a) and R134a are the working fluids that are examined in this research.

2. Modelling Rankine cycle parameters

The main Rankine cycle parameters are the cooling heat exchanger, the geothermal heat exchanger, the turbine and the pump. In this paper, the cooling heat exchanger and the geothermal heat exchanger are modelled.

2.1 Cooling heat exchanger (condenser)

For this research we have used shell and tube condenser, which is standard practice in geothermal binary power plants. The overall heat-transfer coefficient for the condenser is given by equation (1):

$$U_o = \frac{1}{\frac{A_o}{A_i}\cdot\frac{1}{h_i}+\frac{A_o\cdot\ln\left(r_o/r_i\right)}{2\cdot\pi\cdot k\cdot L}+\frac{1}{h_o}} \tag{1}$$

where:

$$h_o = 0.725\left[\frac{\rho\left(\rho-\rho_v\right)gh_{fg}k_f^{\ 3}}{\mu_f d\left(T_g-T_w\right)}\right]^{0.25} \tag{2}$$

where A_o and A_i represents the out and in surface areas respectively of the inner tubes, L is the length of the tubes, h_i *is the heat transfer coefficient* inside the tubes where the cooling fluid flows, h_o *is the heat transfer coefficient* outside the tubes where the working fluid flows and k *represents the thermal conductivity* of the tube's material.

2.2 Geothermal Heat Exchanger (Evaporator)

The heat exchanger used in this research is of the *plate heat exchanger* (PHE) type with corrugated parallel plates attached to one another and fitted into a casing [1]. The plates could have a corrugation angle of β to the main flow direction but in this analysis we maintained the flow to be parallel to the plates ($\beta = 90$). Plate type exchangers are preferred to shell and tube exchangers, as far as it concerns the geothermal heat transfer, because the geothermal water usually contains dissolved particles or ions (silica SiO_2 or salts such as calcium carbonate $CaCO_3$), which tend to be deposited on the surfaces and cause fouling of the heat exchanger. It is obvious that it is easier to clean them from the plates rather than the tubes, as a plate heat exchanger can be easily dismantled and cleaned either mechanically or chemically.

In order to compute *the heat transfer coefficient of the working fluid*, different heat transfer coefficients are used for each fluid phase regime. This is necessary as its corresponding flow in the geothermal heat exchanger (evaporator) begins as single liquid phase flow $h_{sp/l}$ [equation (4)], then as evaporation starts it becomes two-phase flow h_{tp} [equation (5)], and finally when all liquid has been turned into vapour it becomes single vapour phase flow $h_{sp/g}$ [equation (6)].

The *overall heat-transfer coefficient* is given by equation (7):

$$U_{sp/l} = \frac{1}{\frac{1}{h_{sp/l}} + \frac{\Delta x}{ktit} + \frac{1}{h_{gw}}} \tag{4}$$

$$U_{tp} = \frac{1}{\frac{1}{h_{tp}} + \frac{\Delta x}{ktit} + \frac{1}{h_{gw}}} \tag{5}$$

$$U_{sp/g} = \frac{1}{\frac{1}{h_{sp/g}} + \frac{\Delta x}{ktit} + \frac{1}{h_{gw}}} \tag{6}$$

$$U_{total} = \frac{1}{\frac{1}{U_{sp/l}} + \frac{1}{U_{tp}} + \frac{1}{U_{sp/g}}} \tag{7}$$

where Δx represents the thickness of the plate, h_{gw} *is the heat transfer coefficient* of the ground water and k represents the *thermal conductivity* of the plate's material, (titanium in this research). The heat transfer coefficient of the single phase flow is given by equation (8) and the heat transfer coefficient of the two-phase flow by equation (9) where D_e is the diameter (taken as twice the mean plate spacing in PHEs), k_l the working fluid's thermal conductivity for the liquid phase, h_{fg} is the latent heat, p is the working fluid's pressure in the inlet of the heat exchanger, and β is the plate corrugation inclination angle:

$$h_{gw,sp/g-l} = \frac{Nu \cdot k}{L_p} \tag{8}$$

$$h_{tp} = C \cdot \left(\frac{k_l}{D_e}\right) \cdot \left[\frac{\mathrm{Re}_l^2 \cdot h_{fg}}{L_P}\right]^{0.4124} \cdot \left(\frac{p}{p_{cr}}\right)^{0.12} \cdot \left(\frac{65}{\beta}\right)^{0.35} \tag{9}$$

3. Rankine cycle optimization for different working fluids

The optimization tool used is called EASY (Evolutionary Algorithm System developed by the Parallel CFD and Optimization Unit of the Laboratory of Thermal Turbomachines, National Technical University of Athens) [2-3]. The objectives of the optimization are the mmaximization of the overall net conversion efficiency of the plant and the minimization of the cost of the plant. Since the cost of the heat exchanger and the condenser constitute a major part of the plant cost, for our optimizing purposes the plant cost can be substituted by the above two costs. So the new goal is to minimize the cost of the heat exchanger and the condenser that is proportional to the minimization of their surface.

The pressure of the liquid working fluid at the pump outlet, the hot ground water mass flow rate, the mass flow rate of the working fluid in the cycle, the temperature difference of the ground water in the heat exchanger, the temperature difference of the cooling water in the condenser constitute the variables of the optimization.

The electrical power of the plant is 200 kW$_e$ and it is defined as a constraint in this optimization. This indicates that each solution (each optimal Rankine cycle) has to respect this constraint (195 kW is the minimum accepted value and 205 kW the maximum accepted value).

4. Results

4.1 Temperature threshold at 65°C (ORC machine A)

4.1.1 R134a

For the ORC machine A the results of the optimization are plotted in Fig. 1 in the case of R134a used as working fluid. A representative solution and heat exchanger geometry are presented in Tables 1 and 2, respectively.

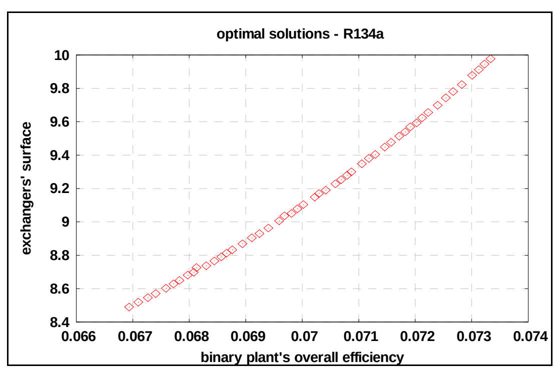

Fig. 1. Rankine cycle optimization - optimal solutions for R134a.

Table 1. A representative solution for R134a.

Parameter	Value
p_2 (kPa)	1199
m_{gr} (kg/sec)	51.2
m_{R134a} (kg/sec)	17.5
ΔT_H (°C)	18.6
ΔT_C (°C)	7.5
R134a pump power (kW)	13.4
Cooling water flow (kg/sec)	116
Surface of the condenser (m^2)	5.5
Surface of the heat exchanger (m^2)	4.0
Total H.E. surface (m^2)	9.5
Net conversion efficiency	7.16
Net Electrical Power (kW)	202

Table 2. Typical features and dimensions of heat exchangers for R134a.

P.H.E. - plate heat exchanger		Shell and tube condenser	
Length of the plate (m)	0.8	Diameter of the tube (cm)	1.3
Width of the plate (m)	0.3	Total length of the tubes (m)	136
Number of plates	17	Number of tubes	29
Total thickness (m)	0.04	Length of the condenser (m)	5

Each point of Fig. 1, which is called the Pareto front, represents an optimal solution that respects the constraints of the optimization. Each solution is represented by two numbers that constitute the objectives of the optimization, the heat transfer surface of the exchangers (geothermal heat exchanger and R134a condenser) and the overall conversion efficiency of the binary plant. Additionally, each solution results from a different combination of the optimization variables and corresponds to an optimal Rankine cycle. The Pareto front supplied us with 50 optimal solutions and the selection of a solution depends on which of the two objectives we want to give priority. A representative solution has been selected in order to observe the values of several important parameters of the optimized Rankine cycle, which are presented in Table 1.

It is obvious that the outlet pressure of the pump, p_2 (which is indicated by the temperature of the ground water), achieves the value of the upper limit in order to take total advantage of the ground heat and maximize the electrical power and proportionally the overall efficiency of the plant. In order to get an idea about the dimensions of the exchangers, according to the surface of the optimal solution, typical dimensions, used in our research, are presented in Table 2.

4.1.2 Comparison between R134a and Isobutane

Comparison of the optimal solutions for the ORC machine A for the working fluids R134a and isobutene (R600a) is shown in Fig. 2, while a comparison of the key variables corresponding to the optimal solutions selected above is shown in Table 3.

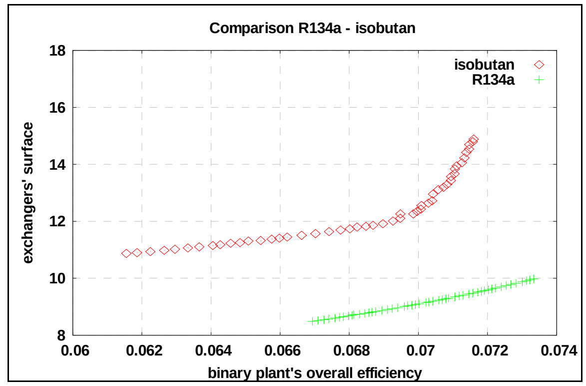

Fig. 2. Rankine cycle optimization- optimal solutions for both R134a and R600a.

Table 3. Rankine cycle variables for selected optimal solutions for R134a and R600a.

Variable	Isobutane	R134a
P_2 (kPa)	619	1199
m_{gr} (kg/sec)	46	51.2
$m_{working\ fluid}$ (kg/sec)	10.2	17.5
ΔT_H (°C)	21.8	18.6
ΔT_C (°C)	7.5	7.5
pump power (kW)	3.9	13.4
Cooling water flow (kg/sec)	119	116
Surface of the condenser (m^2)	7.0	5.5
Surface of heat exchanger (m^2)	5.7	4.0
Total H.E. surface (m^2)	12.7	9.5
Net conversion efficiency	7.04	7.16

By comparing the optimal solutions between Isobutane and R134a in Table 3, it becomes evident that the surface of the heat exchangers' needed for R134a is less than the one for Isobutane when the plant's efficiency is around 7% in both cases. On the other hand however, the geothermal water flow rate, the working fluid mass flow rate and necessary auxiliary pumping power are higher in the case of R134a than in R600a. As the vapor density of R600a is more than 3 times less than the one of R134a the necessary turbine volume for R600a should be around 2 times higher than the one of R134a, further increasing the cost difference between the two machines.

4.2 Temperature threshold at 120°C (ORC Machine B)

In ORC machine B, the optimization concerns of a Rankine Cycle for cogeneration of heat and power by heat recovery from the cooling water circuit since the geothermal fluids is of 120ºC and the cooling water supplies a district heating system at 60/80 ºC. In this analysis, the variables are four since the temperature difference of the cooling fluid in the condenser, ΔT_C, is stable at $\Delta T_{cond.}$=20 ºC. The optimal solutions are presented in Fig. 3, while the key cycle variables for one optimal solution are shown in Table 4, together with the ones corresponding to the ORC machine A.

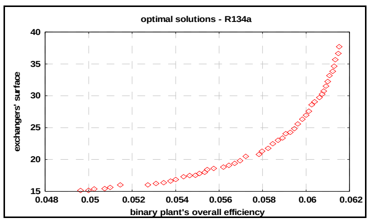

Fig. 3. Rankine cycle optimization- optimal solutions for ORC machine B.

Table 4. Rankine cycle variables for selected optimal solutions for ORC machine B and R134a as working fluid..

Variable	120ºC	65ºC
P_2 (kPa)	3499	1199
m_{gr} (kg/sec)	52	51.2
m_{R134a} (kg/sec)	35	17.5
ΔT_H (°C)	26	18.6
ΔT_C (°C)	20	7.5
Cooling Temperature (°C)	60	10
Condensing Temperature (°C)	80	30
R134a pump power (kW)	58	13.4
cooling water flow (kg/sec)	66	116
Surface of the condenser (m^2)	22.0	5.5
Surface of heat exchanger (m^2)	2.0	4.0
Total H.E. surface (m^2)	24.0	9.5
Net conversion efficiency	5.93	7.16
Net electrical Power (kW)	207	202

By comparing ORC machines A to B for R134a, it is observed that there is a significant difference in the mass flow rate of the working fluid and the pump power (14kW to 60kW). On the other hand, the cooling fluid flow needed for ORC machine B is much less than ORC machine A and this is due to the temperature difference of the cooling fluid in the condenser which is $\Delta T_{C.}$=20 ºC for ORC machine B when $\Delta T_{C.}$= 7.5 ºC for ORC machine A. It is also evident that when the ground water reaches 120 ºC, the surface of the geothermal heat exchanger is less, due to the higher temperature difference between the geothermal water and the R134a. We can also observe that a major difference exists in the value of the condenser's surface that is attributed to the extremely small temperature difference between the condensing temperature and the cooling water outlet temperature.

4.3 Comparison with existing binary machines optimized for 100 ºC geothermal water

In order to examine the feasibility of the two ORC machines, one last optimization run was performed using a standard ORC plant for geothermal water supply of 100°C, but using R134a as refrigerant, in order to obtain comparable results. The comparison of all three machines is shown in Fig. 4 and Table 5.

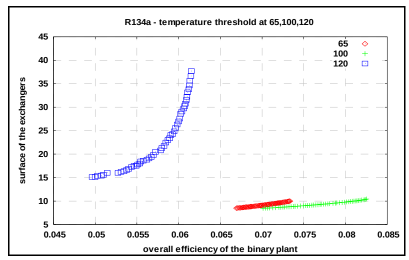

Fig. 4. Rankine cycle optimization- optimal solutions for R134a machines.

Table 5. Rankine cycle variables for selected optimal solutions for three ORC machines (A, B and Standard) with R134a as working fluid.

Variable	Heat & power cogeneration 120 ºC	Power generation 65 ºC	Standard binary power plant, 100ºC
P_2 (kPa)	3499	1199	1552
m_{gr} (kg/sec)	52	51.2	45
m_{R134a} (kg/sec)	35	17.5	17.8
ΔT_H (°C)	26	18.6	20.0
ΔT_C (°C)	20	7.5	7.5
Cooling Temp (°C)	60	10	10
Condensing Temp (°C)	80	30	27
R134a pump power (kW)	58	13.4	18.5
cooling water flow (kg/sec)	66	116	110
ondenser surface (m^2)	22.0	5.5	4.6
Surface of the PHE (m^2)	2.0	4.0	5.4
Total H.E. surface (m^2)	24.0	9.5	10
Net conversion efficiency	5.93	7.16	7.7
Net electrical Power (kW)	207	202	204

By comparing the optimum Rankine cycles of 65 °C to standard binary machines of 100°C we come up with the following conclusions:

- As far as it concerns the net conversion efficiency, the efficiency of the 65°C binary cycle (6.7-7.3%) is a little less than this of the 100°C binary cycle (7.0-8.1%), which is predictable since the temperature of the geothermal water is lower. This observation shows that even by using geothermal water of 65°C, the conversion efficiency remains at the same levels as in binary units of 100°C.
- As far as it concerns the cost of the plant, by comparing the total surface of the heat exchangers, the supply of the working fluid and the hot ground water supply, it is obvious that there is no significant difference which shows that the Rankine cycles of 65°C do not contribute to the increase of the plant's cost.

By comparing ORC machine B (cogeneration of heat and power of 120°C geothermal water) to standards binary machines optimized for 100°C geothermal fluid supply, we come up with the following conclusions:

- As far as it concerns the net conversion efficiency, the efficiency of the 120°C binary cycle (5.0-6.1%) is less than the one of the 100°C binary cycle (7.0-8.1%).
- As far as it concerns the cost of the plant, by comparing the total surface of the exchangers, the supply of the working fluid and the hot ground water supply, it is obvious that there is a difference which shows that the Rankine cycles of 120 °C contribute to a remarkable increase of the plant's cost (almost the twofold cost). The above cost is compensated by the reduction of capital costs per unit of energy delivered due to the use of the same installations (production wells, injection wells, fluid transmission pipelines etc).

5. Conclusions

The main goal of this research was to study and recommend the optimal Rankine cycles for the two geothermal binary power machines. After comparing the optimum Rankine cycles of 65°C and 120°C to standard binary machines of 100°C we come up with the conclusion that the 65°C binary cycle has very good perspective compared with the binary units of 100°C, since the conversion efficiency and the cost of the plant is comparable in both units, taking into account the advantage of the low temperature of the geothermal water. As far as it concerns the 120°C binary cycle, the efficiency is at a lower level than that of the 100°C binary units, as expected to be, but the gain from the cogeneration of heat and power counterbalances the situation.

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