Optimized geothermal binary power cycles

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ABSTRACT

This study has been carried out for the LOW-BIN (Efficient Low Temperature Geothermal Binary Power) project, which is supported by the European Commission FP6 program. Its aim 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 of the exchangers' surface. Through this study, a set of optimal solutions for ORC machines A and B are obtained, that combine maximum plant's efficiency (6.7-7.4 %) and minimum cost. Each optimal solution corresponds to an optimal Rankine Cycle and every parameter of the cycle is defined.

1. INTRODUCTION

The main objectives of the LOW-BIN project are to:

1. 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 study this will be called ORC machine A.
2. 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 study this will be called ORC machine B.

The ORC plants manufacturer TURBODEN is the industrial partner to the LOW-BIN project. Overall research and demonstration is supported by the European Commission DG-TREN, through its 6th framework for support programme.

This paper presents the research carried out in CRES until now towards the above objectives. This corresponds to studying and recommending optimal Rankine cycles for the two geothermal binary power machines mentioned above, e.g. ORC machines A and B. Based on our research presented during the ENGINE workshop held in Strasbourg in mid September 2006, we selected for optimization purposes water cooled machines due to the corresponding higher conversion efficiency and lower costs [4].

Isobutane (R600a) and R134a have been selected as working fluids due to the following reasons:

- They are widely used with excellent results in the heat pumps and cooling/refrigeration industry, which involves inverse Rankine cycle machines.
- Both isobutane (R600a) and R134a are available in the market.
- The necessary parts for the corresponding Rankine cycle machine are also available in the market.
- CARRIER, the multinational air-conditioning manufacturer has lately developed a low cost 200 kWe geothermal binary power unit using R134a as working fluid. The plant was installed in the "Chena" geothermal field in Alaska, USA, utilizing 74°C water. Two units have been installed, which commenced operation in August 2006 and December 2006 respectively.

Then the main Rankine Cycle parameters and components have been defined. The Rankine cycle plant is schematically presented in Figure 1.

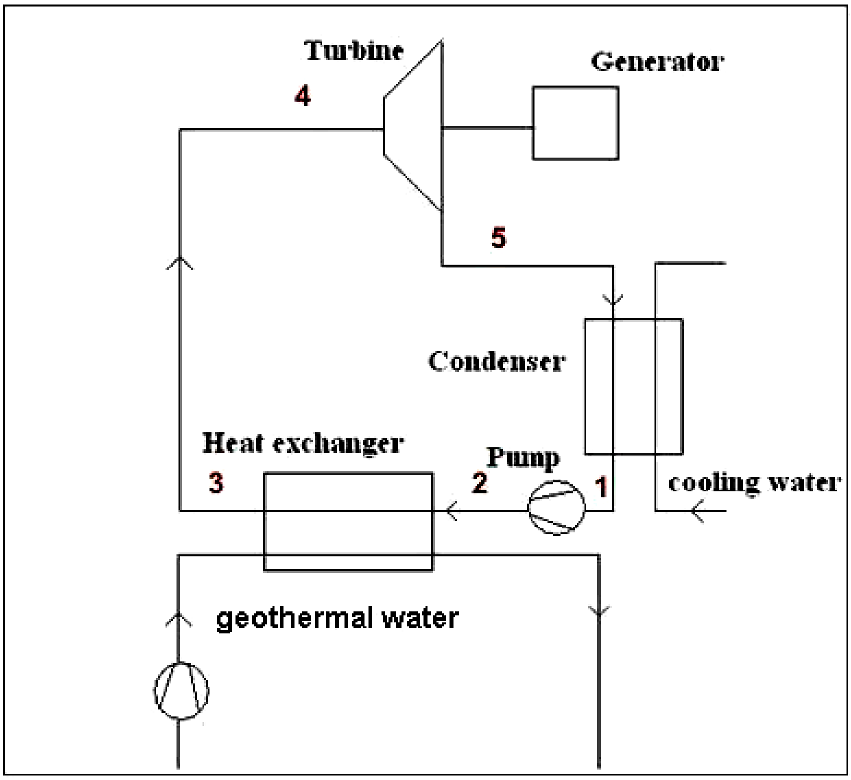

Figure 1: *Binary plant layout.*

2. MODELLING RANKINE CYCLE PARAMETERS

2.1. Cooling heat exchanger (condenser)

For this study 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):

$$U_o = \frac{1}{\frac{A_o}{A_i}\frac{1}{h_i} + \frac{A_o \ln(r_o / r_i)}{2\pi kL} + \frac{1}{h_o}} \tag{1}$$

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.

The heat transfer coefficient outside the tubes is based on the following formula for laminar film condensation on horizontal tubes:

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

where ρ and ρ_v represent the density of working fluid in liquid and vapour forms respectively, h_{fg} is the latent heat, μ_f is the dynamic viscosity of the working fluid, k_f is the thermal conductivity of the working fluid, d the outside diameter of the tube, T_g is the saturation temperature of the fluid condensate and T_w the temperature of the tube's wall.

The heat transfer coefficient inside the tubes for turbulent flow is based on the following formulas :

$$\begin{aligned} h_i &= \frac{Nuk}{D} \\ Nu &= 0.023\,\mathrm{Re}^{0.8}\,\mathrm{Pr}^{0.4} \\ \mathrm{Re} &= \frac{u \cdot D \cdot \rho}{\mu} \end{aligned} \tag{3}$$

where D stands for the outside diameter of the tube, k for the cooling water's thermal conductivity, Nu is the Nusselt number, Pr the Prantl number, and Re the Reynolds number calculated for the ρ, μ cooling water's properties inside the tube.

2.2. Geothermal Heat Exchanger (Evaporator)

The heat exchanger used in this study is of the *plate heat exchanger* (PHE) type with corrugated parallel plates attached to one another and fitted into a casing. 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 (β= 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.

The ***overall heat-transfer coefficient***:

$$U_o = \frac{1}{\frac{1}{h_{wf}} + \frac{\Delta x}{ktit} + \frac{1}{h_{gw}}} \tag{4}$$

where *Δx* represents the thickness of the plate, h_{wf} *is the heat transfer coefficient* of the working fluid, 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 study).

The heat transfer coefficient of the ground water for turbulent flow is based on the following formulas:

$$\begin{aligned}
&\mathrm{Re} = \frac{u \cdot L_p \cdot \rho}{\mu} \\
&c_f = \frac{0.074}{\mathrm{Re}^{0.2}} - \frac{1050}{\mathrm{Re}} \\
&St = \frac{c_f}{2} \cdot \frac{1}{\mathrm{Pr}^{\frac{2}{3}}} \\
&Nu = St \cdot \mathrm{Re} \cdot \mathrm{Pr} \\
&h_{gw} = \frac{Nu \cdot k}{L_p}
\end{aligned} \tag{5}$$

where L_p is the length of the plate, *Re* the Reynolds number calculated for the *ρ, μ* geothermal water properties, c_f the friction coefficient for the flat plate for Reynolds numbers as $Re_{crit} < Re_L < 10^7$, *Nu* is the Nusselt number, *Pr* the Prantl number, *k* the geothermal water thermal conductivity.

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, then as evaporation starts it becomes two-phase flow, and finally when all liquid has been turned into vapour it becomes single vapour phase flow.

The ***heat transfer coefficient for two phase flow*** is based on the following formula which was presented by Ayub [1] in an extensive literature review for Plate Heat Exchangers (PHEs):

$$h_{tp} = C\left(\frac{k_l}{D_e}\right)\left[\frac{\mathrm{Re}_l^{\,2}\, h_{fg}}{L_P}\right]^{0.4124}\left(\frac{p}{p_{cr}}\right)^{0.12}\left(\frac{65}{\beta}\right)^{0.35} \qquad (6)$$

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. This correlation is not dimensionless and the values are based on English units, so in order to use it we converted it into SI units.

The ***heat transfer coefficients for single phase flow*** (liquid, gas) is based on the following formulas:

$$\begin{aligned}
&\mathrm{Re} = \frac{u \cdot L \cdot \rho}{\mu} \\
&c_f = \frac{0.074}{\mathrm{Re}^{0.2}} - \frac{1050}{\mathrm{Re}} \\
&St = \frac{c_f}{2} \cdot \frac{1}{\mathrm{Pr}^{\frac{2}{3}}} \\
&Nu = St \cdot \mathrm{Re} \cdot \mathrm{Pr} \\
&h_{sp/l,g} = \frac{Nu \cdot k}{L}
\end{aligned} \qquad (7)$$

So we calculate ***three overall heat-transfer coefficients,*** one for every phase and we come up with the total coefficient :

$$U_{sp/l} = \frac{1}{\frac{1}{h_{sp/l}} + \frac{\Delta x}{ktit} + \frac{1}{h_{gw}}}, U_{tp} = \frac{1}{\frac{1}{h_{tp}} + \frac{\Delta x}{ktit} + \frac{1}{h_{gw}}}, U_{sp/g} = \frac{1}{\frac{1}{h_{sp/g}} + \frac{\Delta x}{ktit} + \frac{1}{h_{gw}}}$$

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

3. RANKINE CYCLE OPTIMIZATION FOR DIFFERENT WORKING FLUIDS

The optimization tool used is code EASY (Evolutionary Algorithm System by National Technical University of Athens), http://velos0.ltt.mech.ntua.gr/EASY. EASY is a generic optimization tool developed by the Parallel CFD and Optimization Unit of the Laboratory of Thermal Turbomachines. EASY is based on Evolutionary Algorithms and Artificial Intelligence and is capable to solve either single or multi-objective optimization problems, with or without constraints. It exploits Multi-level and/or Distributed exploration on the field of variables and can be efficiently combined with other deterministic optimization methods. It operates in parallel on Unix or Windows clusters and employs user friendly graphical interface. A detailed overview of the algorithmic features of EASY and its additional capabilities can be found in publications, such as [2] and [3].

In order to optimize the Rankine cycle of a typical geothermal binary plant, the optimization's objectives, the variables, the variables' limits and the constraints of the optimization have to be defined.

Objectives of the optimization

- Maximization of the overall net conversion efficiency of the plant:

$$\eta_{cycle} = \frac{W_{turbine} - N_{pump}}{Q_{heatexch}} = \frac{(h_4 - h_5)\cdot m_{wf} - N_{pump}}{(h_3 - h_2)\cdot m_{wf}}$$

- 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 cost plant can be substituted by their cost. So the new goal is to minimize the cost of the heat exchanger and the condenser which is proportional to their surface → Minimization of the exchangers' surface.

Variables of the optimization

1. the pressure of the liquid working fluid at the pump outlet, p_2
2. the hot ground water mass flow rate, m_{gr}
3. the mass flow rate of the working fluid in the cycle, m_{wf}
4. the temperature difference of the ground water in the heat exchanger, ΔT_H
5. the temperature difference of the cooling water in the condenser, ΔT_C

Constraints of the optimization

In the framework of the LOW-BIN project, the electrical power of the plant is at 200kW$_e$ and it is defined as a constraint in this optimization. This indicates that each solution (each optimal Rankine cycle) has to respect this constraint (195kW is the minimum accepted value and 205kW the maximum accepted value). As mentioned above, in order to optimize the cycle, the working fluids that were selected, are R134a and Isobutane (R600a).

4. TEMPERATURE THRESHOLD AT 65°C (ORC MACHINE A)

4.1. R134a

For the ORC machine A, the limits of the optimization variables are shown in table 1, while the results of the optimization are plotted in figure 2 in the case of R134a used as working fluid. A representative solution and heat exchanger geometry are presented in tables 2 and 3 respectively.

Table 1. *Upper and lower limits of the optimization variables for R134a.*

Variable	Lower limit	Upper Limit
p_2 (kPa)	750	1200
m_{gr} (kg/sec)	45	55
m_{R134a} (kg/sec)	10	20
ΔT_H (°C)	10	30
ΔT_C (°C)	7.5	12.5

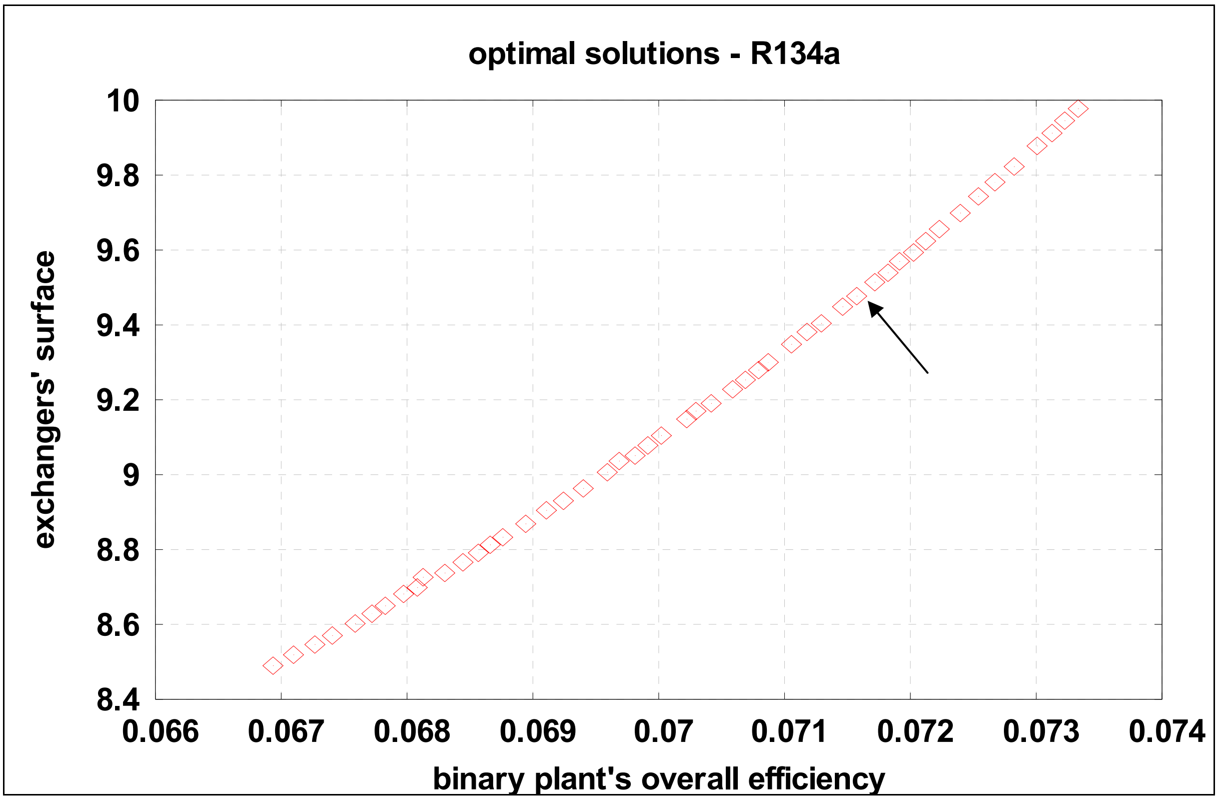

Figure 2: *Rankine cycle optimization- optimal solutions for R134a.*

Each point of the above chart, which is called the Pareto front, represents an optimal solution that respects the constraints of the optimization. Each solution is represented by two numbers which 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 resulted 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 (pointed with the arrow in the figure) has been selected in order to observe the values of several important parameters of the optimized Rankine cycle, which are presented in table 2.

Table 2. *A representative solution for R134a.*

Parameter	Value	Range
p_2 (kPa)	1199	750 - 1200
m_{gr} (kg/sec)	51.2	45 - 55
m_{R134a} (kg/sec)	17.5	10 - 20
ΔT_H (°C)	18.6	10 - 30
ΔT_C (°C)	7.5	7,5 – 12,5
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	

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 study are presented in table 3.

Table 3. *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

4.2. Isobutane R600a

For the ORC machine A, the limits of the optimization variables are shown in table 4, while the results of the optimization are plotted in figure 3 in the case of R600a used as working fluid. A representative solution and heat exchanger geometry are presented in tables 5 and 6 respectively.

Table 4. *Upper and lower limits of the optimization variables for R600a.*

Variable	Lower limit	Upper Limit
p_2 (kPa)	390	620
m_{gr} (kg/sec)	45	55
m_{isob} (kg/sec)	10	20
ΔT_H (°C)	10	30
ΔT_C (°C)	7.5	12.5

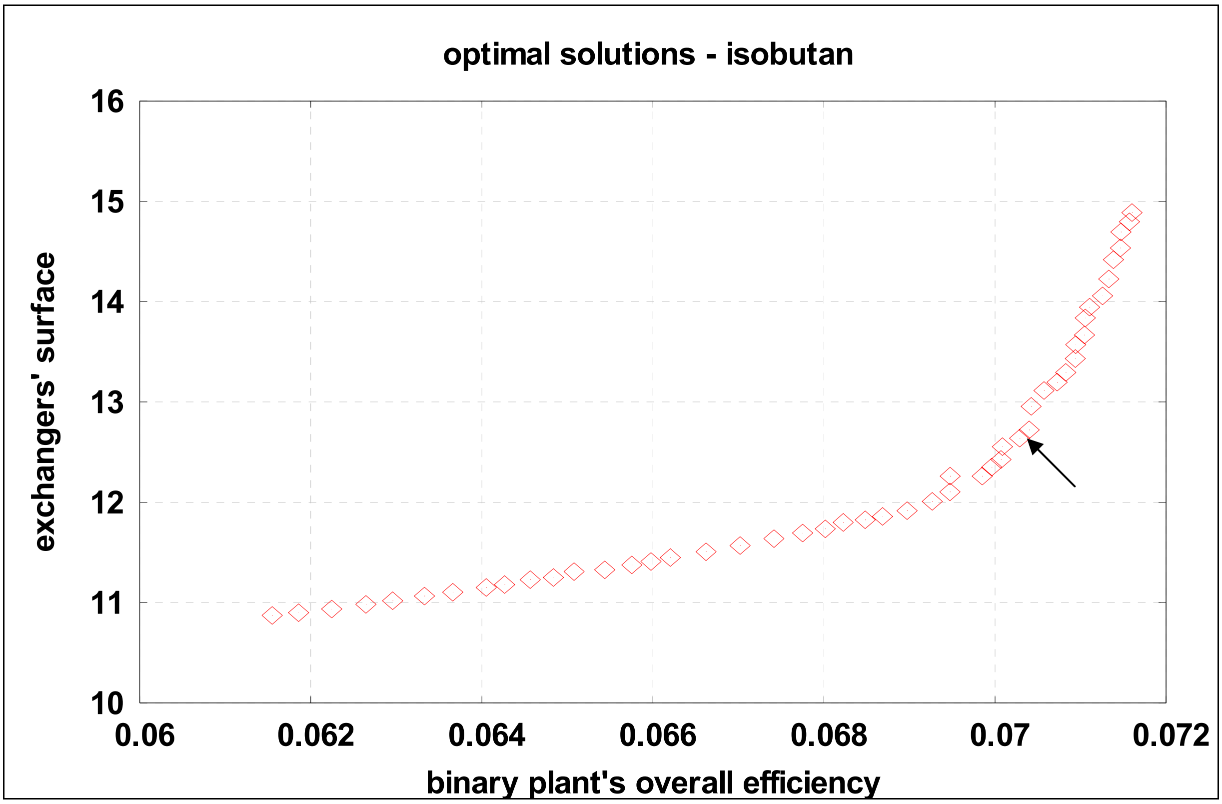

Figure 3: *Rankine cycle optimization- optimal solutions for R600a.*

Table 5. *A representative solution for R600a.*

Parameter	Value	Range
P_2 (kPa)	619	390 - 620
m_{gr} (kg/sec)	46	45 - 55
m_{isob} (kg/sec)	10.2	10 - 20
ΔT_H (°C)	21.8	10 - 30
ΔT_C (°C)	7.5	7,5 – 12,5
pump power (kW)	3.9	
Cooling water flow (kg/sec)	119	
Surface of the condenser (m^2)	7.0	
Surface of the heat exchanger (m^2)	5.7	
Total H.E. surface (m^2)	12.7	
Net conversion efficiency	7.04	
Net electrical Power (kW)	~ 200	

Table 6. *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)	171
Number of plates	29	Number of tubes	35
Total thickness (m)	0.07	Length of the condenser (m)	5

4.3. 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 figure 4, while a comparison of the key variables corresponding to the optimal solutions selected above is shown in table 7.

By comparing the optimal solutions between Isobutane and R134a, 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.

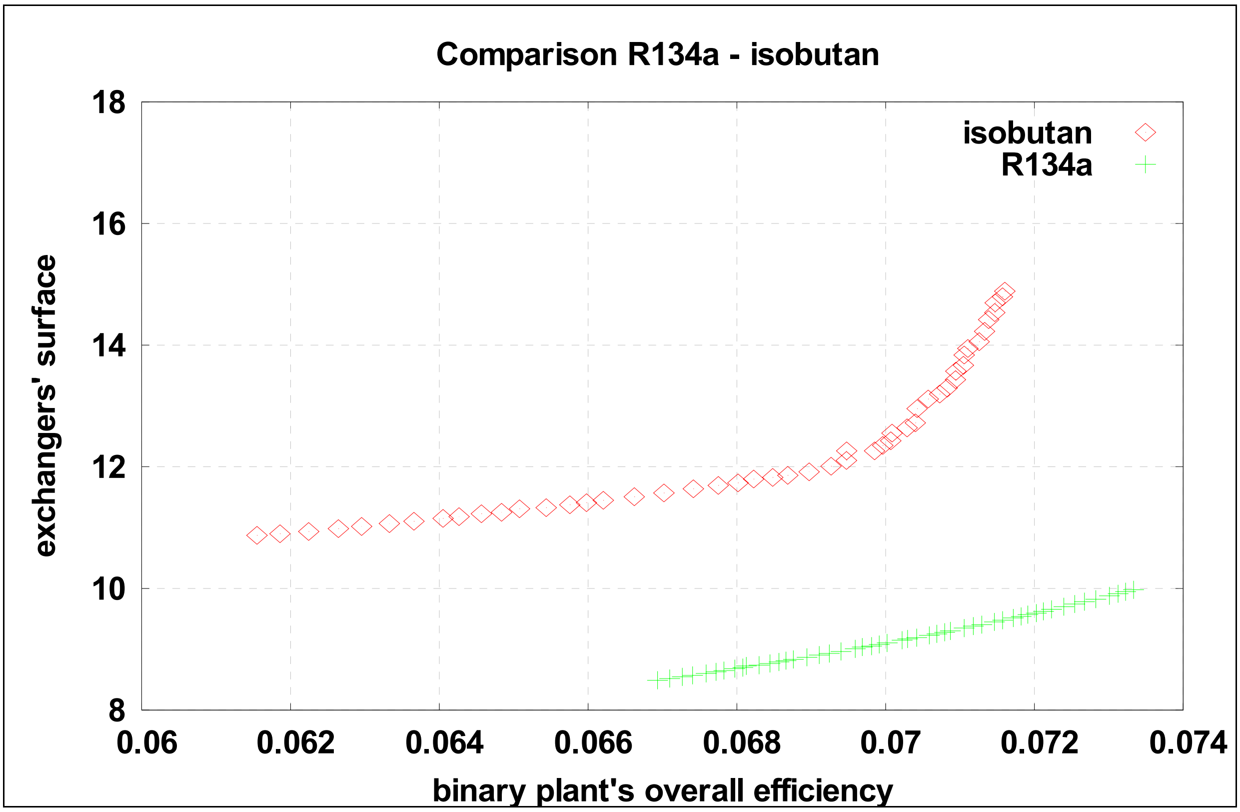

Figure 4: *Rankine cycle optimization- optimal solutions for both R134a and R600a.*

Table 7. *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 the 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

5. TEMPERATURE THRESHOLD AT 120°C (ORC MACHINE B)

As we have described before, 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 variables limits are shown in table 8, while the the optimal solutions are presented in figure 5, while the key cycle variables for one optimal solution are shown in table 9, together with the ones corresponding to the ORC machine A.

Table 8. *Upper and lower limits of the optimization variables for ORC machine B.*

Variable	Lower limit	Upper Limit
p_2 (kPa)	3100	3500
m_{gr} (kg/sec)	45	70
m_{R134a} (kg/sec)	25	40
ΔT_H (°C)	15	30

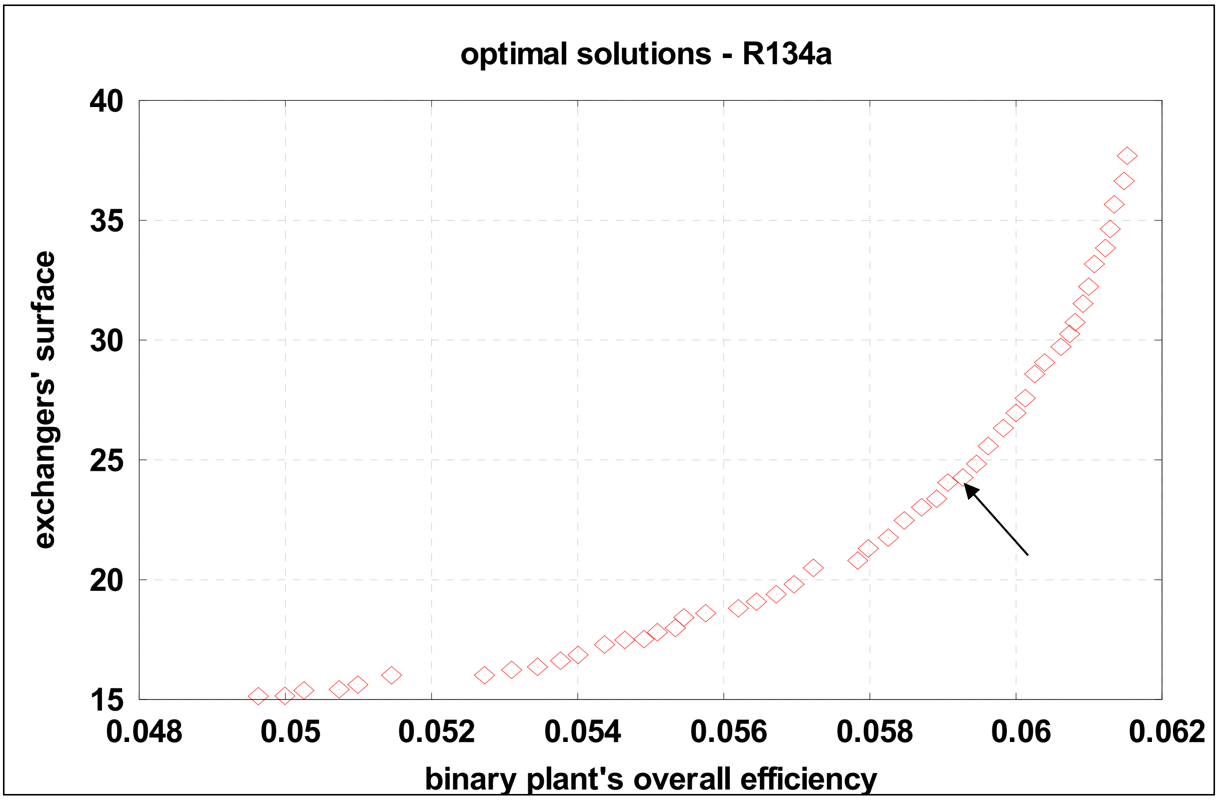

Figure 5: *Rankine cycle optimization- optimal solutions for ORC machine B.*

Table 9. *Rankine cycle variables for selected optimal solutions for ORC machine B and R134a as working fluid.*

Variable	120°C	65°C
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 the 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 which is attributed to the extremely small temperature difference between the condensing temperature and the cooling water outlet temperature.

6. COMPARISON WITH EXISTING BINARY MACHINES OPTIMIZED FOR 100 °C 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 figure 6 and table 10.

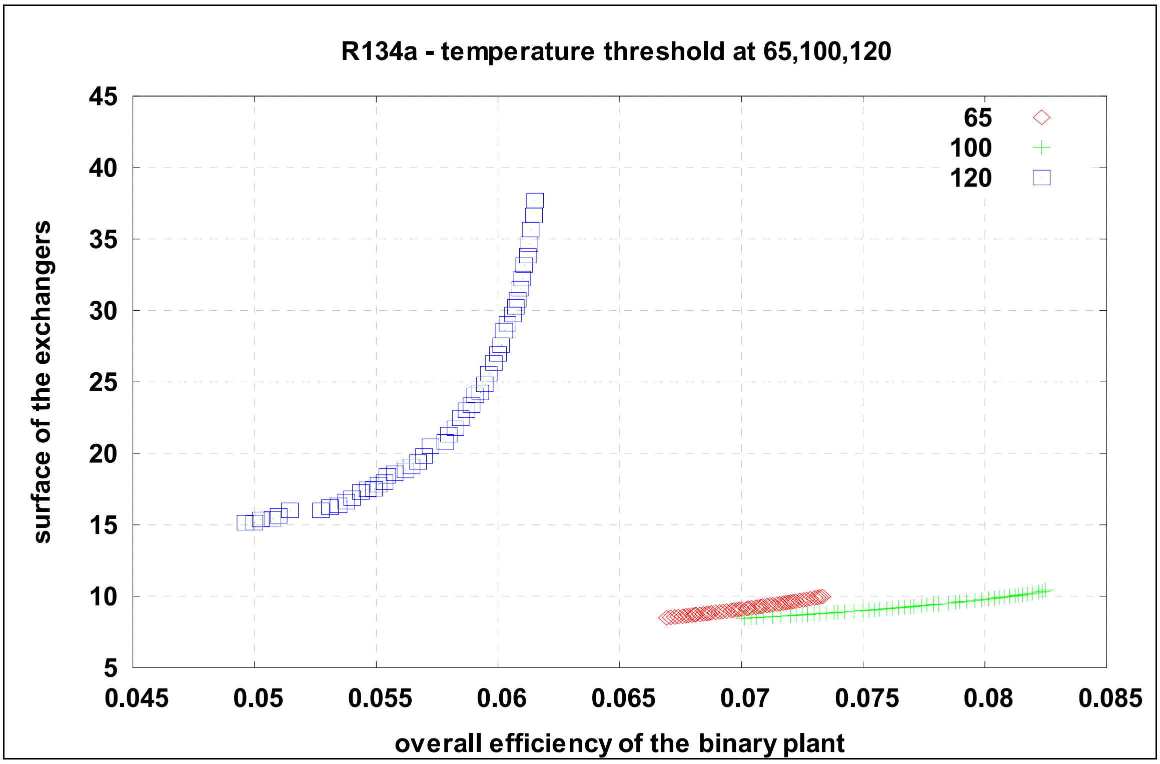

Figure 6: *Rankine cycle optimization- optimal solutions for R134a machines.*

Table 10. *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°C	Power generation 65°C	Standard binary power plant, 100°C
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
Condenser surface (m²)	22.0	5.5	4.6
Surface of the PHE (m²)	2.0	4.0	5.4
Total H.E. surface (m²)	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 don't 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).

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