

Comparison of thermodynamic cycles for power production from low-temperature geothermal heat sources

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ABSTRACT

The performance of different types of organic Rankine cycles (ORCs) and of the Kalina cycle is investigated and optimized for low-temperature (100–150 °C) geothermal heat sources. A variety of configurations is worth considering. The ORCs can be subcritical or transcritical and can have one or more pressure levels. Each cycle can be a simple one, have recuperation or turbine bleeding and up to 80 different working fluids can be used. Comparison of these cycles to each other and to an optimized Kalina cycle concludes that the transcritical and multi-pressure subcritical cycles are the best ones. Exergetic plant efficiencies of above 50% can be achieved when the brine is allowed to cool down as much as possible. A constraint on the brine outlet temperature causes a strong decrease in the mechanical power output of the cycle. Due to the low heat source temperatures, a low condenser temperature and small temperature differences in the heat exchanger are very important.

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1. Introduction

Low-temperature (100–150 °C) geothermal heat sources have a very large potential as a renewable energy source [1,2]. The amount of energy available is huge, but due to the low-temperature, the electricity conversion efficiency is low. For this energy conversion, the use of flash-steam power plants is difficult [3] and therefore two types of binary cycles are proposed in the literature. The first one is the organic Rankine cycle (ORC), which is similar to a steam Rankine cycle, but uses an organic working fluid instead of water. A second type is the so-called Kalina cycle. The Kalina cycle is similar to an ORC, but uses a mixture of water and ammonia with a variable composition as working fluid [4–6]. ORCs are used for different low-temperature heat sources (biomass, geothermal, solar, etc.) [7].

Different configurations of ORCs have been proposed in the literature. Subcritical ORCs with or without recuperator are optimized by Dai et al. [8] for 10 different working fluids. It is shown that R236ea is the best performing working fluid for an inlet temperature of 145 °C and that a recuperator is not always useful. Cayer et al. [9] have optimized a transcritical ORC for a heat-source temperature of 100 °C and a turbine inlet temperature between 80 and 100 °C. CO₂, ethane and R125 are used as possible working fluids. Only the simple ORC configuration with a pump, heat exchan-

ger, turbine and condenser are considered. Kanoglu [10] has investigated multi-pressure cycles. An exergy analysis of an existing power plant, which uses the combination of two subcritical cycles with isopentane, has been performed. Gnutek and Bryszewska-Mazurek [11] have considered a multi-cycle ORC. The power plant consists of four subcritical power cycles, using the heat source in series. Heberle and Brüggemann [12] have concluded that for power generation from low-temperature geothermal heat sources, fluids with a low critical temperature are optimum. Only cycles which use a recuperator have been investigated. Mago et al. [13] have compared simple ORCs with ORCs which use turbine bleeding (called “regenerative” in the paper). It is concluded that the cycle efficiency of the so-called regenerative cycle is higher than that of the simple cycle. It is also stated that reheat ORCs do not perform better than a simple ORC. Different configurations of subcritical ORCs have been compared by Yari [14]. Simple cycles, cycles with recuperation, cycles with turbine bleeding and cycles with recuperation and turbine bleeding are simulated. All these cycles are subcritical ones with a fixed evaporation temperature of 120 °C. It is shown that cycles with recuperation or turbine bleeding are the most promising ones. Subcritical and transcritical cycles with or without recuperation have been investigated by Saleh et al. [15] for many working fluids. For low-temperature geothermal sources, it is shown that fluids with a low critical pressure in the transcritical cycles are optimum. Borsukiewicz-Gozdur and Nowak [16] improved the power output of an ORC by returning a part of the geothermal water after the evaporator to the inlet of the evaporator. The influence of cooling

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Nomenclature

e	specific exergy (kJ/kg)
\dot{E}	exergy (kW)
h	specific enthalpy (kJ/kg)
\dot{I}	irreversibility (kW)
\dot{m}	mass flow [kg/s]
ORC	organic Rankine cycle
p	pressure (bar)
\dot{Q}	heat flow (kW)
s	specific entropy (kJ/kg K)
T	temperature (°C)
\dot{W}	mechanical power (kW)
x	vapor fraction (–)

Sub and superscripts

0	dead state
0	reference
1 – 11	number of the state
brine	brine
cond	condenser
cooling	cooling (installation)
cycle	cycle
en	energetic
ex	exergetic
high	higher

HX	heat exchanger between brine and working fluid
i	number of pressure level
in	inlet
int	intermediate
low	lower
min	minimum
mix	mixing
net	netto
out	outlet
P	pump
plant	plant
recup	recuperator
s	isentropic
T	turbine
wf	working fluid

Greek symbols

Δ	difference
η	(isentropic) efficiency (–)
ϵ_i	improvement of addition of the i th pressure level
ϵ	fraction of mass flow through turbine (Kalina) (–)
ϵ	fraction of mass flow extracted from turbine (turbine bleeding) (–)

medium temperature, working fluid flow rate, etc. on the performance of ORCs is investigated by Nguyen et al. [17].

For low-temperature sources, the Kalina is often mentioned as an alternative for the ORC [4–6]. Although the Kalina cycle is often called to be superior to the ORC [4,6], DiPippo [18] has shown that an existing Kalina cycle has about the same performance as existing ORCs.

As seen from the list above, many thermodynamic cycles for low temperature heat sources have been proposed in the literature. In all of the mentioned references, the number of cycle configurations or the number of fluids is limited and often only one heat source inlet temperature is investigated. Often certain assumptions are made like a fixed evaporation temperature, no superheated cycles, etc. In this paper the most promising of these proposed cycles are simulated, optimized and compared to each other. So, simple ORCs, recuperated ones and ORCs with turbine bleeding are treated. All of these can have one or more pressure levels and every pressure level can be subcritical or transcritical. Up to 80 fluids are investigated as a working fluid. The Kalina cycle is also simulated, optimized and exergetically compared to the different ORCs. In this way, the thermodynamic optimum binary cycle is found for different brine inlet temperatures, condenser temperatures and pinch point temperature differences. The influence of a constraint on the brine outlet temperature on the cycle configuration and performance is also investigated.

2. Definition of different efficiencies

In the literature, different definitions for efficiencies exist. All of these efficiencies are useful, but they often have a different meaning. Therefore, it is important to know how the efficiencies are defined and to use the right efficiency at the right moment.

2.1. Cycle efficiency

The *cycle efficiency* describes how well a thermodynamic cycle converts heat, which is added to the cycle, to mechanical power.

The *energetic cycle efficiency* η_{en}^{cycle} is defined as the fraction of the net mechanical power output to the heat input:

$$\eta_{en}^{cycle} = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \quad (1)$$

\dot{W}_{net} and \dot{Q}_{in} are the net mechanical power output and the heat input to the cycle, respectively. When the heat source is a geothermal brine, the heat added to the cycle can be written as

$$\dot{Q}_{in} = \dot{m}_{brine} \Delta h_{brine} = \dot{m}_{brine} (h_{in} - h_{out}) \quad (2)$$

with \dot{m}_{brine} the mass flow of brine, Δh_{brine} the specific enthalpy drop of the brine, h_{in} and h_{out} the specific enthalpy of the brine before and after heat is added to the cycle.

Analogous to the energetic cycle efficiency, also an *exergetic cycle efficiency* η_{ex}^{cycle} can be defined:

$$\eta_{ex}^{cycle} = \frac{\dot{W}_{net}}{\dot{m}_{brine} (e_{in} - e_{out})} \quad (3)$$

where e_{in} and e_{out} are the specific flow exergy of the brine before and after heat is added to the cycle, respectively. The specific flow exergy e is defined as:

$$e = (h - h_0) - T_0 (s - s_0), \quad (4)$$

where s is the specific entropy of the brine and the subscript 0 refers to the dead state; so T_0 is the dead-state temperature, usually being the temperature of the environment.

2.2. Plant efficiency

The cycle efficiency of the previous section only describes how efficient a cycle is in converting a certain amount of heat into mechanical power, but it does not tell how efficient the heat source is used. Therefore, both energetic η_{en}^{plant} and exergetic η_{ex}^{plant} plant efficiencies are defined, respectively:

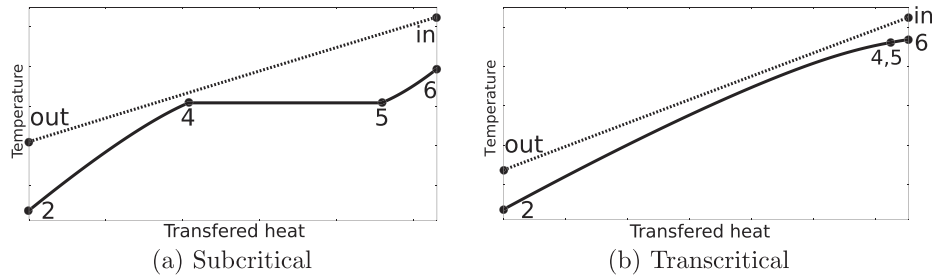


Fig. 1. Temperature-heat diagrams of the heating process in a subcritical (a) and transcritical (b) cycle. The working fluid (–) is heated by the cooling brine (···).

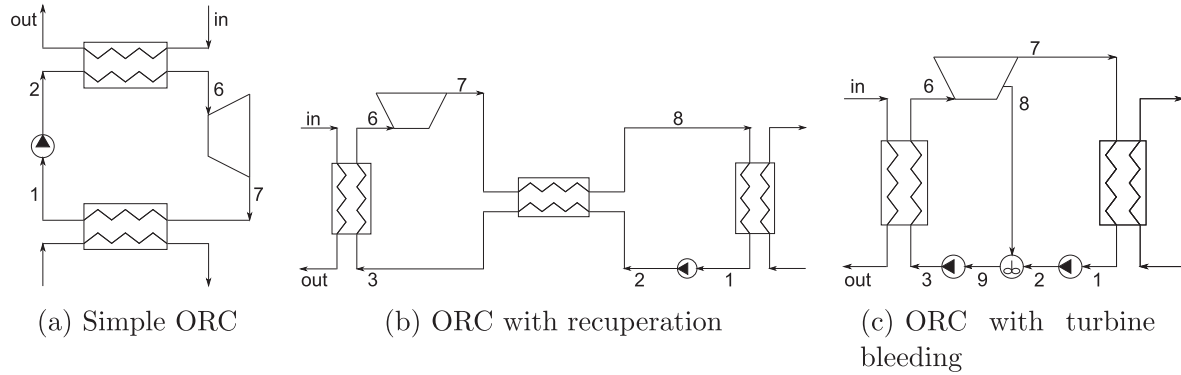


Fig. 2. Different ORC configurations: simple ORC (a), ORC with recuperator (b) and ORC with turbine bleeding (c). The brine in and outlet are given by “in” and “out”, respectively.

$$\eta_{en}^{plant} = \frac{\dot{W}_{net}}{\dot{Q}_{available}} = \frac{\dot{W}_{net}}{\dot{m}_{brine}(h_{in} - h_0)} \quad (5)$$

$$\eta_{ex}^{plant} = \frac{\dot{W}_{net}}{\dot{E}_{available}} = \frac{\dot{W}_{net}}{\dot{m}_{brine}e_{in}} \quad (6)$$

$\dot{Q}_{available}$ and $\dot{E}_{available}$ are the heat and exergy in the heat source, respectively. These plant efficiencies can be rewritten in function of the respective cycle efficiencies:

$$\eta_{en}^{plant} = \frac{\dot{W}_{net}}{\dot{m}_{brine}(h_{in} - h_{out})} \frac{\dot{m}_{brine}(h_{in} - h_{out})}{\dot{m}_{brine}(h_{in} - h_0)} \quad (7)$$

$$= \eta_{en}^{cycle} \eta_{en}^{cooling} \quad (8)$$

$$\eta_{ex}^{plant} = \frac{\dot{W}_{net}}{\dot{m}_{brine}(e_{in} - e_{out})} \frac{\dot{m}_{brine}(e_{in} - e_{out})}{\dot{m}_{brine}e_{in}} \quad (9)$$

$$= \eta_{ex}^{cycle} \eta_{ex}^{cooling} \quad (10)$$

where $\eta_{en}^{cooling}$ and $\eta_{ex}^{cooling}$ show how efficient the geothermal brine is cooled in the energetic and exergetic sense, respectively.

When a geothermal brine is used to produce electricity, the goal is to maximize the net work output per mass unit of brine. So, the fraction $\dot{W}_{net}/\dot{m}_{brine}$ should be maximized. As seen from Eqs. (5) and (6) this is the same as maximizing the energetic or exergetic plant efficiency, because h_{in} , h_0 and e_{in} only depend on the geothermal source and the environment.

Eqs. (8) and (10) show that the heat source should be cooled down as far as possible (i.e., $\eta_{en}^{cooling} \rightarrow 1$) and the heat added to the cycle should be efficiently converted to electricity, to have a high plant efficiency and thus a high power output per mass unit of brine.

In this paper, different configurations of Organic Rankine Cycles (ORCs) and the Kalina cycle will be described and optimized for maximum plant efficiency.

3. Organic Rankine cycle

In this section, different types of ORCs are described.

3.1. Subcritical and transcritical ORCs

A distinction has to be made between subcritical, transcritical and supercritical cycles. In a subcritical cycle, the pressure is always below the critical pressure and in a supercritical cycle, the pressure is always above the critical pressure. In a transcritical cycle, the low pressure is below the critical pressure, and the high pressure above the critical one.¹ In this paper, only subcritical and transcritical cycles are investigated.

Fig. 1 shows the temperature-heat diagram of the heating process in a subcritical and a transcritical cycle. In the subcritical cycle, the working fluid is heated in the economizer (2 → 4), in the evaporator (4 → 5) and in the superheater (5 → 6). Due to the evaporation at constant temperature in the subcritical cycle, large temperature differences between the brine cooling curve and the working fluid heating curve exist. These temperature differences induce the creation of high irreversibilities. In the transcritical cycle, the temperature differences between the brine cooling curve and the working fluid heating curve are smaller. So, the irreversibilities created during the heat exchange of the transcritical case are smaller and because of the better fit between the curves, the brine can cool down more than in the subcritical case.

3.2. ORC configurations

Three ORC configurations are investigated, as shown in Fig. 2. These three configurations can be subcritical or transcritical,

¹ The so-called “supercritical steam cycle” is actually a transcritical cycle according to this definition.

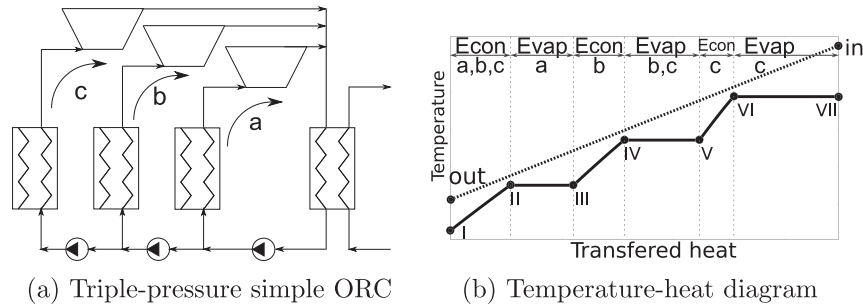


Fig. 3. Scheme and temperature-heat diagram of a triple-pressure, subcritical cycle without superheating. The working fluid (—) is heated by the cooling brine (···). The three pressure levels are called *a*, *b* and *c*. The economizers and evaporators are denoted by “Econ” and “Evap”, respectively.

depending on the maximum pressure in the cycle. In the simple ORC (Fig. 2a), the working fluid is pumped from the low to the high pressure ($1 \rightarrow 2$), heated by the brine ($2 \rightarrow 6$), expanded in the turbine ($6 \rightarrow 7$) and cooled in the condenser ($7 \rightarrow 1$). Often, state 7 is still at a high temperature and a recuperator can be used as shown in Fig. 2b. This configuration will improve the cycle efficiency, because less heat is needed for the same power output. Another method to improve the cycle efficiency is the use of turbine bleeding as depicted in Fig. 2c. Part of the working fluid (state 8) is extracted from the turbine at an intermediate pressure and is mixed with state 2 to form state 9. Then the working fluid is pumped to the high pressure ($9 \rightarrow 3$) and the rest of the cycle is analogous to the simple ORC.

3.3. Number of pressure levels

The cycles shown in Fig. 2 have only one high pressure level (except for the steam-bleeding case, which has an intermediate bleeding pressure). It is possible to combine two or more of these cycles. Fig. 3a shows a simple ORC with three pressure levels, but other configurations are also possible. The brine, which is not shown on the figure, is split into parts to heat the different cycles. This splitting is done in such a way that the hot brine first heats the working fluid at the highest temperature. Fig. 3b shows the temperature-heat diagram of a three-pressure level, subcritical cycle without superheating. The three sub-cycles are called *a*, *b* and *c* and $p_a < p_b < p_c$. It is seen that the temperature difference between the brine cooling curve and the working fluid heating curve is lower than for a subcritical cycle with one pressure level. So, efficiency improvements can be obtained by applying more than one pressure level.

3.4. Modeling of the ORCs

For the modeling of the ORCs, it is assumed that the condenser temperature T_{cond} , the isentropic pump efficiency η_p and isentropic turbine efficiency η_T are known. Further assumptions are that state 1 is saturated liquid and that the heat exchangers are ideal (no pressure drop and heat loss). It is allowed that the working fluid at the inlet of the turbine (state 6) is superheated for all types of fluids (even for the so-called dry² ones).

For the cycle with recuperation, it is assumed that as much heat as possible is recuperated, taking into account a minimum temperature difference in the recuperator ΔT_{min}^{recup} . For the cycle with turbine bleeding, it is assumed that state 9 is saturated liquid. The mass flow at the inlet of the turbine is given by \dot{m}_{wf} . The fraction ε is extracted at the intermediate pressure and the other part $(1 - \varepsilon)$ expands to the condenser pressure.

3.4.1. Pump and turbine

The relationship between the states before (state 1) and after (state 2) the pump are given by:

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_p} \quad (11)$$

with state 1 at the low pressure p_{low} , state 2 at the high pressure p_{high} and state 2s at p_{high} and the same entropy as state 1. The mechanical power needed to drive the pump \dot{W}_p is:

$$\dot{W}_p = \dot{m}_{wf}(h_2 - h_1) \quad (12)$$

with \dot{m}_{wf} the mass flow of the working fluid. The irreversibility created in this pump, is:

$$\dot{I}_p = (\dot{m}_{wf}e_1 + \dot{W}_p) - (\dot{m}_{wf}e_2) = \dot{m}_{wf}T_0(s_2 - s_1) \quad (13)$$

For the second pump in the cycle with turbine bleeding, the equations are analogous:

$$\dot{W}_p = (1 - \varepsilon)\dot{m}_{wf}(h_2 - h_1) + \dot{m}_{wf}(h_3 - h_9) \quad (14)$$

$$\dot{I}_p = (1 - \varepsilon)\dot{m}_{wf}T_0(s_2 - s_1) + \dot{m}_{wf}T_0(s_3 - s_9) \quad (15)$$

For the turbine, the relationship between the states at the turbine inlet (state 6) and at the turbine outlet (state 7) is given by:

$$h_7 = h_6 - (h_6 - h_{7s})\eta_T \quad (16)$$

with state 6 at p_{high} , state 7 at p_{low} and state 7s at p_{low} and the same entropy as state 6. The turbine mechanical power and irreversibility generated in the turbine are:

$$\dot{W}_T = \dot{m}_{wf}(h_6 - h_7) \quad (17)$$

$$\dot{I}_T = \dot{m}_{wf}T_0(s_7 - s_6) \quad (18)$$

For the cycle with turbine bleeding, these equations are:

$$\dot{W}_T = \dot{m}_{wf}(h_6 - h_8) + (1 - \varepsilon)\dot{m}_{wf}(h_8 - h_7) \quad (19)$$

$$\dot{I}_T = \dot{m}_{wf}T_0(s_8 - s_6) + (1 - \varepsilon)\dot{m}_{wf}T_0(s_7 - s_8) \quad (20)$$

In this paper, the net mechanical power output is given by:

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_p \quad (21)$$

So, the power needed for cooling and other auxiliary power is not taken into account.

3.4.2. Heat exchangers

The heat exchangers are assumed to be ideal; so no pressure drop is induced and no heat is lost to the environment. The energy balance for the heat exchanger HX between the brine and working fluid is given by:

$$\dot{m}_{brine}(h_{in} - h_{out}) = \dot{m}_{wf}(h_6 - h_3) \quad (22)$$

A fixed pinch point temperature difference ΔT_{min}^{HX} is assumed in this heat exchanger. In the recuperator, the energy balance is:

² A dry fluid has a saturated vapor line with a positive slope in the temperature-entropy diagram.

$$(h_7 - h_8) = (h_3 - h_2) \quad (23)$$

and the minimum temperature difference is given by $\Delta T_{\min}^{\text{recup}}$. The pinch point temperature differences $\Delta T_{\min}^{\text{HX}}$ and $\Delta T_{\min}^{\text{recup}}$ are generally not the same. The energy equation for the condenser in the simple or recuperated cycle is given by

$$\dot{m}_{\text{wf}}(h_x - h_1) = \dot{m}_{\text{cooling}}(h_{\text{out}}^{\text{cooling}} - h_{\text{in}}^{\text{cooling}}) \quad (24)$$

with $h_x = h_7$ in the simple cycle and $h_x = h_8$ in the cycle with recuperator. The pinch point temperature difference in the condenser is $\Delta T_{\min}^{\text{cond}}$, a mass flow \dot{m}_{cooling} of cooling water is used and $h_{\text{in}}^{\text{cooling}}$ & $h_{\text{out}}^{\text{cooling}}$ are the enthalpy of the cooling water before and after the condenser, respectively.

The irreversibilities created in the heat exchanger HX and recuperator are:

$$\begin{aligned} \dot{I}_{\text{HX}} &= (\dot{m}_{\text{brine}}e_{\text{in}} + \dot{m}_{\text{wf}}e_3) - (\dot{m}_{\text{brine}}e_{\text{out}} + \dot{m}_{\text{wf}}e_6) \\ &= \dot{m}_{\text{brine}}T_0(s_{\text{out}} - s_{\text{in}}) + \dot{m}_{\text{wf}}T_0(s_6 - s_3) \end{aligned} \quad (25)$$

$$\begin{aligned} \dot{I}_{\text{recup}} &= \dot{m}_{\text{wf}}[(e_2 + e_7) - (e_3 + e_8)] \\ &= \dot{m}_{\text{wf}}T_0[(s_3 - s_2) + (s_8 - s_7)] \end{aligned} \quad (26)$$

The irreversibilities created in the cooling installation are given by

$$\dot{I}_{\text{cooling}} = \dot{m}_{\text{wf}}(e_x - e_1) \quad (27)$$

with $e_x = e_7$ for the simple cycle and $e_x = e_8$ for the cycle with recuperator. So, in the cooling installation, all the exergy is lost to the environment. For the cycle with turbine bleeding, the irreversibility creation is:

$$\dot{I}_{\text{cool}} = (1 - \varepsilon)\dot{m}_{\text{wf}}(e_7 - e_1) \quad (28)$$

3.4.3. Mixing

In the cycle with turbine bleeding, two flows are mixed. The energy equation for the mixing process is given by:

$$\varepsilon h_8 + (1 - \varepsilon)h_2 = h_9 \quad (29)$$

and the irreversibility created is:

$$\begin{aligned} \dot{I}_{\text{mix}} &= \dot{m}_{\text{wf}}[(1 - \varepsilon)e_2 + \varepsilon e_8] - e_9 \\ &= \dot{m}_{\text{wf}}T_0(s_9 - [(1 - \varepsilon)s_2 + \varepsilon s_8]) \end{aligned} \quad (30)$$

3.5. Optimization

As mentioned in Section 2, the objective of the optimization is to maximize the plant efficiency. The decision variables for a simple and a recuperated ORC are the pressure and the temperature at the turbine inlet (state 6) of every pressure level. This state can be saturated or superheated vapor. The boundaries for the tempera-

ture of state 6 are: $T_{\text{cond}} \leq T_6 \leq T_{\text{in}} - \Delta T_{\min}^{\text{HX}}$ and the pressure should be higher than the condenser pressure. For a cycle with turbine bleeding, the bleeding pressures of every pressure level are added as decision variables. This pressure should be higher than the condenser pressure and lower than the maximum pressure of the cycle. It is also limited from above by the critical pressure of the fluid. In every optimization step, all the states of the cycle can be determined by the value of the decision variables at that moment. The mass flow of working fluid is calculated by Eq. (22) and by the value of $\Delta T_{\min}^{\text{HX}}$.

4. Kalina cycle

The scheme of a typical Kalina cycle, which always contains a recuperator, is shown in Fig. 4a. State 1, which has an intermediate concentration of ammonia x_{int} , is pumped to the high pressure (1 → 2), heated in a recuperator (2 → 3) and heated by the brine (3 → 5). State 5 is in the two-phase region and the vapor part (state 6 with a high concentration of ammonia x_{high}) is split from the liquid part (state 9 with a low concentration of ammonia x_{low}) in the separator. State 6 is expanded in the turbine (6 → 7), mixed with state 9 (7 + 9 → 11), cooled in the recuperator (11 → 8) and cooled in the condenser (8 → 1). Often a second recuperator is used to exchange heat between states 9 and 3, but in this paper this heat exchanger is omitted because the mass flow around the turbine is small.

Fig. 4b show the corresponding temperature-heat diagram of the Kalina cycle of Fig. 4a. Because a mixture of fluids is used, the evaporation at constant pressure does not take place at a constant temperature. This “temperature glide” decreases the temperature difference in the heat exchange between brine and working fluid; so less irreversibilities are created in the Kalina cycle than in a subcritical ORC. The temperature-glide at the condenser pressure allows a strong internal heat recuperation.

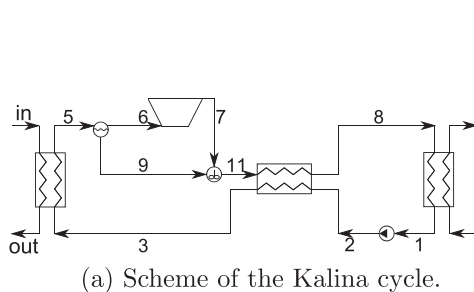
4.1. Modeling of the Kalina cycle

The only component which has not yet been described in the section of the ORCs is the separator. In this component, the two-phase fluid is separated into its liquid and vapor part. An amount \dot{m}_{wf} of working fluid is heated in the heat exchanger, the fraction ε of this mass flow is expanded in the turbine:

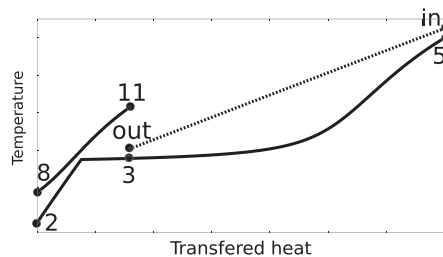
$$\varepsilon = \frac{x_{\text{int}} - x_{\text{low}}}{x_{\text{high}} - x_{\text{low}}} \quad (31)$$

4.2. Optimization

The objective is to maximize the plant efficiency. The decision variables for the Kalina cycle are the temperature and pressure at the turbine inlet (state 6) and the intermediate concentration of



(a) Scheme of the Kalina cycle.



(b) Temperature-heat diagram of the Kalina cycle.

Fig. 4. Scheme and temperature-heat diagram of the Kalina cycle. The working fluid (–) is heated by the cooling brine (···).

ammonia x_{int} . The limits for the temperature and pressure are analogous to the ones for the ORCs. The concentration is limited by: $0 < x_{int} < 0.96$. The upper limit is chosen to avoid numerical problems in the calculation of the fluid properties (Section 5.1).

5. Results and discussion

In this section, the results of the comparison between the ORC configurations of Figs. 2 and 3 and the Kalina cycle of Fig. 4 for the different cases are given. All the simulations have been performed with our own developed code, written in Python. The thermodynamic properties are obtained from RefProp [19], which is connected to Python by using F2PY [20]. The SLSQP optimization subroutine from Scipy [21] is used as the optimizer.

5.1. Input parameters

Table 1 shows the parameters which are used in the remainder of this paper, unless denoted otherwise. The brine inlet temperature and pressure are given by T_{in} and p_{in} , respectively. The cooling water temperature is taken equal to T_0 .

For some fluids available in RefProp, only supercritical cycles are possible and they are therefore skipped. All other fluids are used in the calculations to obtain the thermodynamic optimum cycle configuration. Only thermodynamic properties of the fluids are taken into account. Other properties like ozone depletion potential, global warming potential, and flammability are not taken into account. For the Kalina cycle, a maximum ammonia concentration of 96% is used, because for higher concentrations RefProp is not always able to find the thermodynamic properties of the mixture.

5.2. Validation of the models

The model for the subcritical, single-pressure ORC and the optimization routine are validated with the data published by Dai et al. [8]. The optimization is performed for the same working conditions and for various fluids. The comparison shows very good agreement, as seen from Table 2.

Subcritical cycles without superheating are optimized by Yari [14]. Simple cycles, recuperated cycles and cycles with turbine bleeding with a fixed evaporation temperature of 120 °C are treated. The results are given in Table 2. From this table, it is concluded that the model for recuperation and turbine bleeding are validated.

Table 1
Input parameters.

T_{in}	125 °C	η_p	80%
p_{in}	5 bar	η_T	85%
T_{cond}	25 °C	ΔT_{min}^{HX}	5 °C
T_0	15 °C	ΔT_{min}^{recup}	5 °C
p_0	1 bar	ΔT_{min}^{cond}	5 °C

Table 2
Validation of the single-pressure ORC-model with data published in the literature.

Working fluid	T_6 (°C)	p_{high} (bar)	T_{out} (°C)	η_{en}^{cycle} (%)	Specific net power output (kJ/kg-wf)	Type	Source
R236ea	87.73	12.0	59.59	11.53	22.71	Subcritical, simple	[8]
	87.28	11.9	59.57	11.48	22.57		Present
R113	120	6.81	≈105	14.45	–	Subcritical, recuperated	[14]
	120	6.81	105.83	14.44	25.37		Present
R113	120	6.81	≈110	14.34	–	Subcritical, turbine bleeding	[14]
	120	6.81	112.51	14.24	22.42		Present
CO ₂	100	119	–	8.4	20.1	Transcritical, simple	[9]
	100	118.8	25.60	8.43	20.08		Present

Cayer et al. [9] have maximized the specific net power output per unit of working fluid for a transcritical cycle with CO₂ as working fluid. As seen from Table 2, it can be concluded that the present model works fine for transcritical cycles.

The model of the Kalina cycle is validated by comparison to Ogriseck [5].

5.3. Net power output

In this section, two cases are investigated. In the first case, there is no constraint on the brine outlet temperature. In the second case, a minimum brine outlet temperature of 75 °C is used. This is done to simulate cases where scaling (fouling) has to be avoided or when the brine is used for heating or cooling (absorption or adsorption cooling).

5.3.1. Brine cooling temperature without limit

5.3.1.1. Single pressure. Fig. 5 shows the exergetic plant and energetic cycle efficiency for the 20 fluids with the highest power output in a single pressure level cycle for the simple configuration. The fluids are ordered from the best (R227ea) to the worst fluid (C5F12) of the 20 (in terms of exergetic plant efficiency). As mentioned in Section 2, it is the plant efficiency that should be maximized. In Fig. 5 it is chosen to plot the *exergetic* plant efficiency, because this characteristic allows an easier comparison between power plants with different brine inlet temperatures or dead-state temperatures (e.g. Section 5.4). The energetic cycle efficiency is added to the figure, because this value shows that maximizing the plant efficiency is not equal to maximizing the cycle efficiency. It is also a characteristic that is often used in the literature and is added as a reference.

Although the exergetic plant efficiency decreases monotonically, the energetic cycle efficiency does not. It is seen from Eqs. (8) and (10) that the plant efficiency is the product of the cycle efficiency and the brine cooling efficiency. This means that the cycle with R115 can cool the brine further down than a cycle with R125, because both have about the same cycle efficiency, but a different plant efficiency. The other way around is also possible, as seen with R41 and R32.

In this figure, only cycles with a simple configuration are shown, because the model shows that it is not useful to use a recuperator or turbine bleeding. Recuperation and turbine bleeding can increase the cycle efficiency, but not the plant efficiency in this case. When an internal heat exchanger is used, the amount of heat transferred between the brine and the working fluid is decreased and the brine cools down less. The cycle efficiency increases, but the brine cooling efficiency decreases. The net result is zero.

The best cycles are all transcritical ones and have critical temperatures below 80 °C. The only exception is R227ea, which has a critical temperature of 101.75 °C. In the obtained cycle with R227ea, the expansion occurs partially through the two-phase region. This has no consequences on the performance of the turbine if the outlet state of the turbine is superheated [22,23]. Only

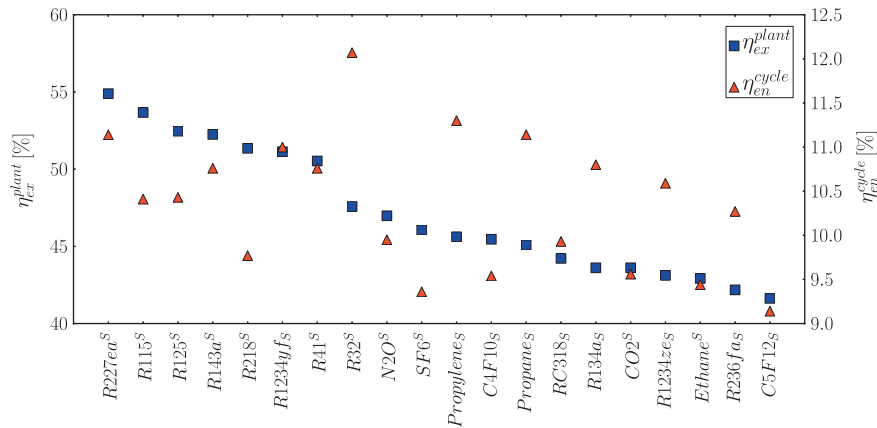


Fig. 5. Exergetic plant efficiency (\square) and energetic cycle efficiency (\triangle) of the 20 fluids with the highest mechanical power output in a single pressure level cycle for the simple (S) configuration. Subcritical cycles are denoted with a subscript S and transcritical cycles with a superscript S.

R236fa and C5f12 have critical temperatures equal to or higher than the brine inlet temperature.

R245fa is often used as the working fluid in ORCs. An optimization with this fluid leads to a subcritical cycle with an exergetic plant efficiency of 39.7% and an energetic cycle efficiency of 10.4%. The power output of this cycle does not belong to the top-20 and is therefore not mentioned in Fig. 5.

All pure fluids condense at a constant temperature, so that the cooling water temperature increases from 15 °C to about 20 °C ($T_{cond} - \Delta T_{min}^{cond}$). In the Kalina cycle, the minimum temperature in the condenser is assumed to be 21 °C instead of 25 °C, so that the cooling water heats up from 15 to about 20 °C. Even with this assumption, the exergetic plant efficiency of the Kalina cycle is only 41.5%. So, the Kalina cycle is even thermodynamically outperformed by simple subcritical ORCs. The obtained Kalina cycle, is not a “real” Kalina cycle, because the working fluid before the separator (state 5) is saturated vapor. So, a separator is not needed and a recuperated subcritical ORC with a mixture of water and ammonia with a constant concentration is obtained.

In the Kalina cycle, the temperature glide at the condenser pressure allows a high amount of recuperation. As shown in Fig. 4b, the pinch point is somewhere in the evaporator and not at the end of the economizer. This allows a higher mass flow of working fluid per unit brine and a higher power output.

Fig. 6 shows the net mechanical power production and irreversibilities as a fraction of the incoming brine exergy for four opti-

mized cycles. The first one is a transcritical cycle with R227ea, the second one is a subcritical cycle with propylene, the third one is the Kalina cycle and the fourth one is a subcritical cycle with R245fa, which serves as a reference. The cycle with R227ea performs the best because of the low irreversibilities created in the heat exchange between brine and working fluid and because of the high brine cooling efficiency. The pressure ratio in this cycle is high, so the irreversibilities created in the pump and turbine are high too. The combination of the high brine cooling efficiency and the average cycle efficiency causes relatively high irreversibilities in the cooling installation.

The subcritical cycle with propylene performs better than the Kalina cycle because of the higher brine cooling efficiency and the lower irreversibilities created in the high-pressure heat exchange (I_{HX} and I_{recup}). In the Kalina cycle, the lower irreversibilities generated in the condenser (temperature glide), turbine and pump cannot compensate this. The cycle with R245fa produces less power than the other cycles because of the very high irreversibilities created in the heat exchanger. This is a consequence of the high heat of vaporization of R245fa. To cool the brine sufficiently down, the optimum evaporation temperature should be low and a high amount of irreversibilities is created in the heat exchanger.

One of the advantages of the Kalina cycle would be the temperature glide. This temperature glide causes a low creation of irreversibilities in the condenser, but not in the high-pressure heat exchanger. Some simple subcritical ORCs even outperform the

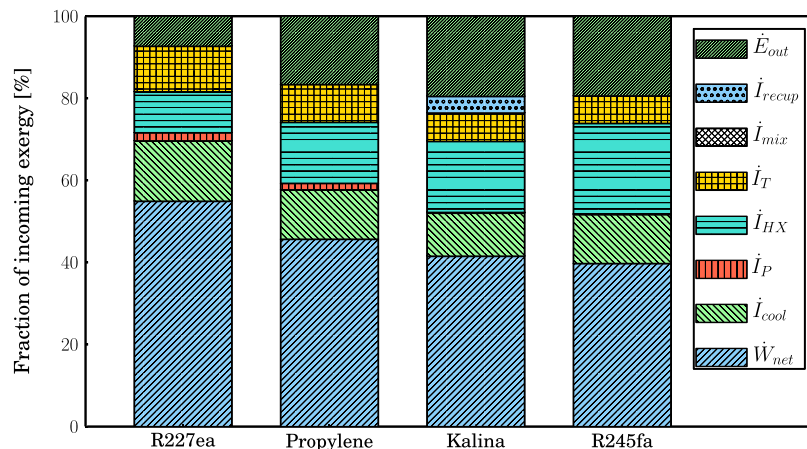


Fig. 6. Net mechanical power production and irreversibilities created, as fraction of the incoming brine exergy for the optimum transcritical cycle with R227ea, the optimum subcritical cycle with propylene, the optimum Kalina cycle and the optimum cycle with R245fa as a reference.

Kalina cycle considered in this context. This comparison is only thermodynamical, so no economic conclusions are made.

5.3.1.2. Dual pressure. It is also possible to use more than one pressure level. The exergetic plant and energetic cycle efficiency for the 20 fluids with the highest power output in a dual-pressure level cycle are shown in Fig. 7. Again, only the simple configuration is shown because of the same reason as before.

The cycles with the highest mechanical power output are almost all a combination of a subcritical cycle with a transcritical one, but the addition of the subcritical part does not improve the plant efficiency very much (e.g. for R227ea an increase of 3% is obtained by using two pressure levels instead of one). For subcritical cycles, the effect is much stronger. The dual-pressure level cycle with R134a produces almost 20% more mechanical power than the single-pressure level. Because of the strong improvement of the subcritical cycles and the poor improvement of the transcritical cycles, the difference between the best and the worst fluid/cycle becomes smaller. The best 15 fluids have an exergetic efficiency above 50%. So, it can be concluded that subcritical cycles with two pressure levels are almost as efficient as transcritical cycles.

For the single-pressure cycle in Fig. 5, many fluids with a low critical temperature like R41, N₂O, SF₆, CO₂ an Ethane perform well. Their optimum cycle is a transcritical one, with a pressure much above their critical pressure. Due to this high pressure, the

irreversibilities created in the pump and the turbine are high. When two pressure levels are allowed (Fig. 7), most of these fluids are not among the 20 best performing fluids and are replaced by fluids with a higher critical temperature like R22, C4F10, R124, R12, R236ea and Isobutane. Their critical temperature ranges from 110 to 140 °C and the both pressure levels are subcritical.

For the single-pressure level cycles (Fig. 5), the energetic cycle efficiency fluctuates strongly. For the dual-pressure level cycles, the energetic cycle efficiency is more constant and is about 11%. So, the extra pressure level does not only reduce the difference in plant efficiency, but also in cycle efficiency.

5.3.1.3. Triple and multi-pressure. Fig. 8 shows the efficiencies when a third pressure level is added. The exergetic plant efficiency improves a bit for the subcritical cycles, but the effect is much less than the addition of the second pressure level. The difference in performance between the fluids has become smaller and the energetic cycle efficiency is about 11%. Addition of more pressure levels will improve the power plant performance, but every extra pressure level will have less effect. By addition of this extra pressure level, fluids with higher critical temperatures (e.g. R152a, R245fa, etc.) are performing relatively better, replacing fluids with lower critical pressures (e.g. R41, R218, etc.). Fig. 9 shows the exergetic efficiency of a multi-pressure cycle with isobutane for different pressure levels. The improvement of the addition of an extra pres-

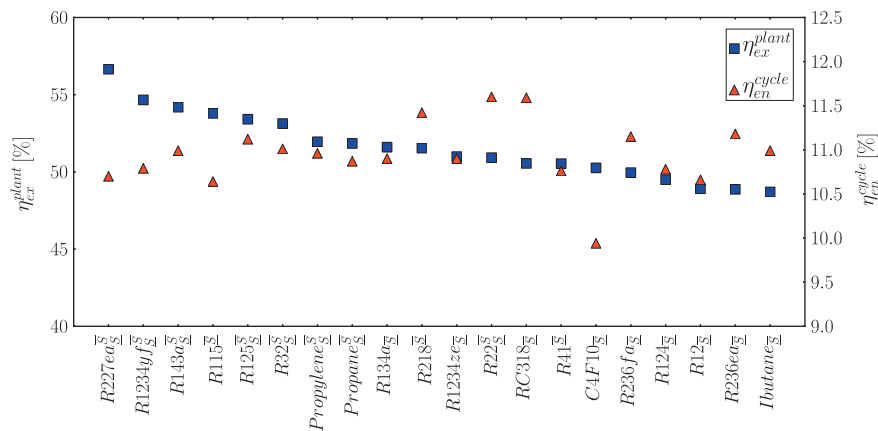


Fig. 7. Exergetic plant efficiency (\square) and energetic cycle efficiency (\triangle) of the 20 fluids with the highest mechanical power output in a dual-pressure level cycle for the simple configuration. Subcritical cycles are denoted with a subscript S and transcritical cycles with a superscript S. The low-pressure sub-cycle is underlined, the high-pressure sub-cycle has an overbar.

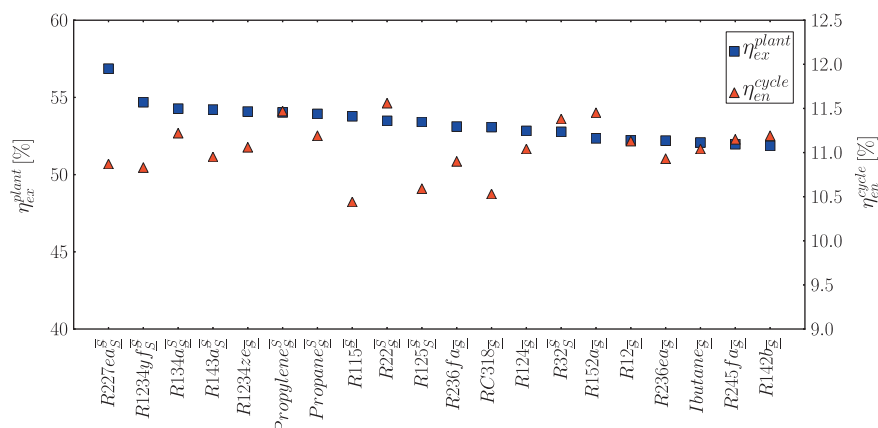


Fig. 8. Exergetic plant efficiency (\square) and energetic cycle efficiency (\triangle) of the 20 fluids with the highest mechanical power output in a triple-pressure level cycle for the simple configuration. Subcritical cycles are denoted with a subscript S and transcritical cycles with a superscript S. The low-pressure sub-cycle is underlined, the intermediate pressure sub-cycle is lined through and the high-pressure sub-cycle is overlined.

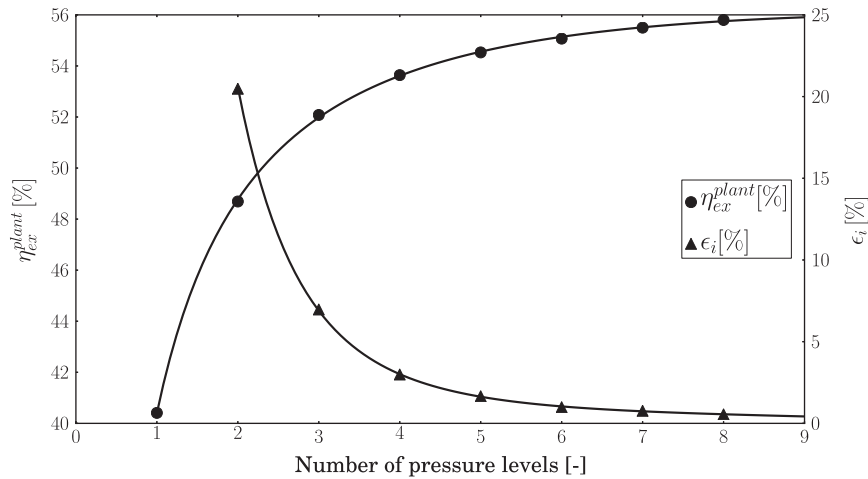


Fig. 9. Exergetic plant efficiency (◦) of a multi-pressure simple cycle with isobutane in function of the number of pressure levels. The improvement of an extra pressure level is shown (△).

sure level is shown on the right hand side. This improvement is defined as:

$$\epsilon_i = \frac{\dot{W}_{net}^i - \dot{W}_{net}^{i-1}}{\dot{W}_{net}^{i-1}} \quad (32)$$

with ϵ_i the improvement of addition of the i th pressure level and \dot{W}_{net}^i the net power output of a cycle with i pressure levels. From the figure, it is seen that the addition of an extra pressure level is thermodynamically useful, but the effect decreases with increasing number of pressure levels.

Fig. 10 shows the net mechanical power production and irreversibilities created in the single, dual and triple-pressure cycle with isobutane. The addition of extra pressure levels improves the brine cooling efficiency and decreases the irreversibility created in the high-pressure heat exchanger. Because of the higher pumping power and turbine power, also more irreversibilities are created in these components.

5.3.2. Constraint on brine outlet temperature

Fig. 11 shows the exergetic plant and energetic cycle efficiencies in the case when the minimum brine outlet temperature is limited to 75 °C. Again the 20 fluids with the highest mechanical power output in a single pressure level cycle are given. In this case, the cycles with recuperation or turbine bleeding can have higher power

outputs than the simple one. This follows from Eq. (10). The brine cooling is limited to a temperature which is higher than the brine cooling temperature in the case without constraint. So, in every cycle the brine will be cooled to 75 °C and the brine cooling efficiency is fixed. Recuperation and turbine bleeding can improve the cycle efficiency; so in the limited case, they can improve the plant efficiency. This also explains the strong correlation between the cycle and plant efficiency in Fig. 11. The often used R245fa, which can be seen as a reference case, performs best in a subcritical, recuperated cycle, with an exergetic plant efficiency of 37.4% and an energetic cycle efficiency of 12.6% (not in the figure).

As seen in Fig. 11, recuperation is always useful, except for R22 and R32. These are very wet fluids, so they would need very high superheating to be able to recuperate. The gain in power output because of the recuperation is apparently lower than the loss in power output due to the extra superheating.

The transcritical cycles are not as superior as in the unconstrained case. For recuperation or turbine bleeding, superheating before the turbine is often needed. The maximum temperature of the working fluid is limited by the temperature of the brine. So, when superheating is used, the maximum pressure has to decrease. Therefore, only for fluids with a low critical pressure, transcritical cycles with recuperation or turbine bleeding are the optimum. Fluids with very low critical temperatures like R41, N₂O and CO₂, which perform well in the unconstrained case with one pressure le-

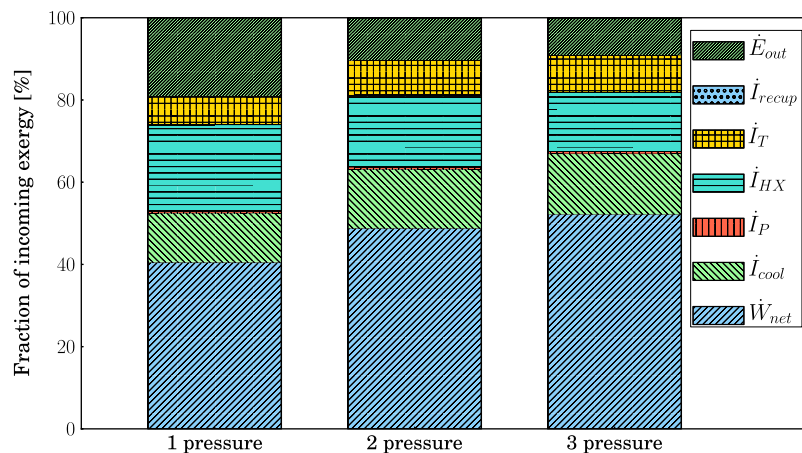


Fig. 10. Net mechanical power production and irreversibilities created, as fraction of the incoming brine exergy for the optimum single, dual and triple-pressure cycle with isobutane.

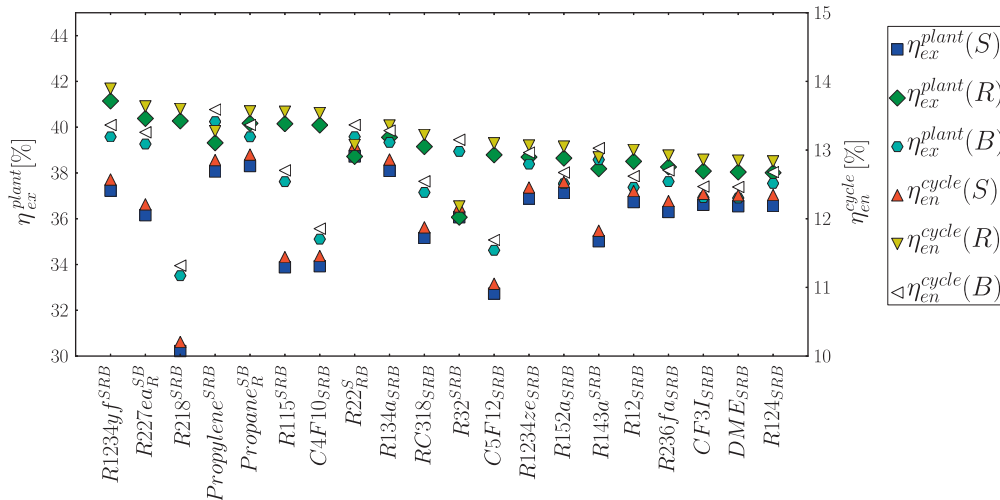


Fig. 11. Exergetic plant efficiency and energetic cycle efficiency of the 20 fluids with the highest mechanical power output in a single pressure level cycle with a limited brine outlet temperature of 75 °C. S stands for the simple cycle, R for the cycle with recuperation and B for the cycle with turbine bleeding. Subcritical cycles are denoted with a subscript, transcritical ones with a superscript.

vel, are not the best choice in the constrained case. Their critical temperature is very low (30–45 °C) and the maximum pressure in the cycle is much above the critical pressure to decrease the amount of irreversibilities created in the heat exchanger. Due to this high pressure, the temperature rise in the pump is high and recuperation becomes hard. Turbine bleeding is also not successful enough. The bleeding pressure is limited from above by the critical pressure. For the fluids mentioned above, the critical pressure is low and the temperature of state 3 will be low too. This leads to higher irreversibilities in the heat exchange and a low plant efficiency. It can be concluded that the optimum fluids in the constrained case have higher critical temperatures than in the unconstrained one.

For the optimization of the Kalina cycle with a constrained brine outlet temperature, a minimum temperature in the condenser of 21 °C is used. Because of the high cycle efficiency of the Kalina cycle, it performs relatively better than in the unconstrained case. An exergetic efficiency of almost 40% is obtained, which is almost the same as for the unconstrained case. The performance of the Kalina cycle is in this case similar to the best ORCs, mainly because of the temperature glide in the condenser which allows a lower minimum temperature in the condenser. When the minimum con-

denser temperature is limited to 25 °C, an exergetic efficiency of 37% is obtained.

The effect of an extra pressure level is shown in Fig. 12. It shows that the 20 best fluids are optimum when using recuperation and the difference in performance between the fluids is very small. Turbine bleeding is almost always better than the simple cycle, except for the very dry fluid perfluorobutane (C4F10). By adding a pressure level, fluids with high critical temperature become relatively better.

The transcritical cycles have not much benefit from the extra pressure level. Because of the small difference in performance between transcritical and subcritical cycles for one pressure level, the optimum cycles for two pressure levels are almost all subcritical. The addition of more pressure levels has the same effect as in the unconstrained case.

5.4. Influence of changing parameters

In this section, the influence of some parameters is investigated. These parameters are the brine inlet temperature, the constraint on the brine outlet temperature, the condenser temperature and

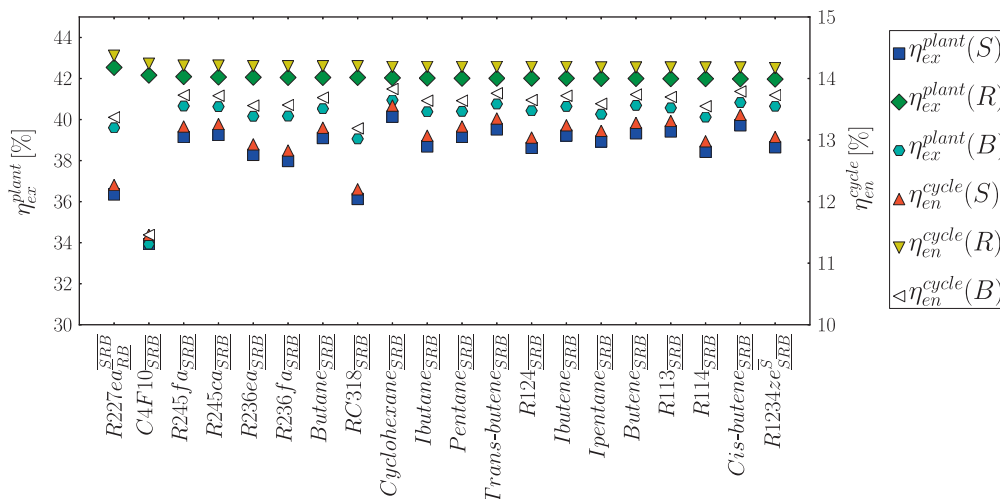


Fig. 12. Exergetic plant efficiency and energetic cycle efficiency of the 20 fluids with the highest power output in a dual pressure level cycle with a limited brine outlet temperature of 75 °C. S stands for the simple cycle, R for the cycle with recuperation and B for the cycle with turbine bleeding. Subcritical cycles are denoted with a subscript, transcritical ones with a superscript. The low pressure sub-cycle is underlined, the high pressure sub-cycle is overlined.

the minimum temperature difference between the brine and the working fluid. These parameters are varied, while all other parameters have the value given in Table 1. For every influencing parameter, the effect on one or a few of the best ORCs is shown. Comparison with the Kalina cycle is added.

5.4.1. Brine inlet temperature

The brine inlet temperature is varied from 100 to 150 °C. There is no constraint on the brine outlet temperature and for every brine inlet temperature, the optimum working fluid is selected. The exergetic efficiency with these fluids is given in Fig. 13. It is seen that the optimum fluid depends strongly on the brine inlet temperature and fluids with a higher critical temperature perform relatively better when the brine inlet temperature is increased. For the fluids R227ea and RC318, the exergetic plant efficiency increases almost linear for low brine inlet temperatures (subcritical cycles). The transition of a subcritical to a transcritical cycle is discontinuous, followed by a weak rise in the efficiency for higher temperatures. For even higher temperatures the exergetic plant efficiency is almost constant and starts to decrease. The specific power output on the contrary keeps increasing (not in figure). The plant efficiency decreases, but the increase of the incoming exergy of the brine is higher (see Eq. (6)). The investigated Kalina cycle is added as a comparison and is clearly outperformed by the ORCs. The figure also shows that the maximum exergetic efficiency depends on

the brine inlet temperature. This is because of the fixed temperature differences in the heat exchangers, which are relatively more important for the low brine inlet temperatures.

In a second case, the minimum brine outlet temperature is limited to 75 °C. These results are given in Fig. 14. The exergetic efficiency rises with brine inlet temperature, because the available brine temperature drop $T_{in} - T_{out}$ rises linearly for a fixed T_{out} . For the constrained case, there is no clear transition from a subcritical to a transcritical cycle. Again the investigated Kalina cycle is added as a comparison. For higher brine inlet temperatures, the Kalina cycle performs almost as good as the best ORCs.

5.4.2. Constraint on brine outlet temperature

The constraint on the brine outlet temperature is varied between 25 and 100 °C. Both the temperature at which scaling occurs and the temperature of a possible heat demand depend strongly on the location (brine composition, old or new houses, etc.). Therefore, a wide range in brine outlet temperatures is chosen and the influence is investigated. Fig. 15 shows the exergetic efficiency for four single-pressure, recuperated ORCs and the Kalina cycle considered. In the unconstrained case, the brine outlet temperature is about 50–70 °C, depending on the fluid. So, the exergetic plant efficiency is only influenced by the constraint when the constraint temperature is above the unconstrained brine outlet temperature.

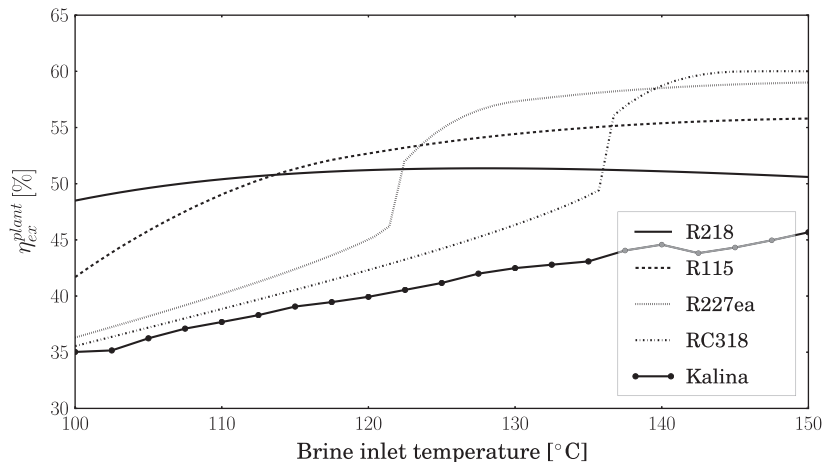


Fig. 13. Exergetic efficiency of the temperature-dependent optimum fluids without constraint on the brine outlet temperature. The Kalina cycle is added as a comparison.

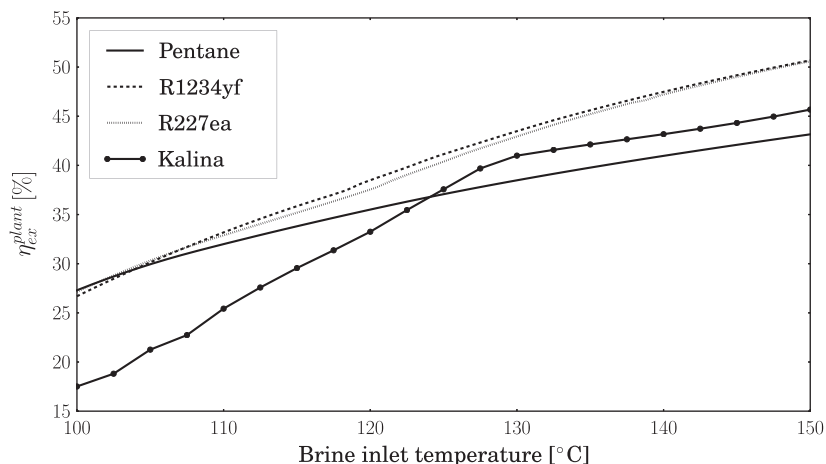


Fig. 14. Exergetic efficiency of the brine inlet temperature-dependent optimum fluids (all recuperative), for a brine outlet temperature limit of 75 °C. The Kalina cycle is added as a comparison.

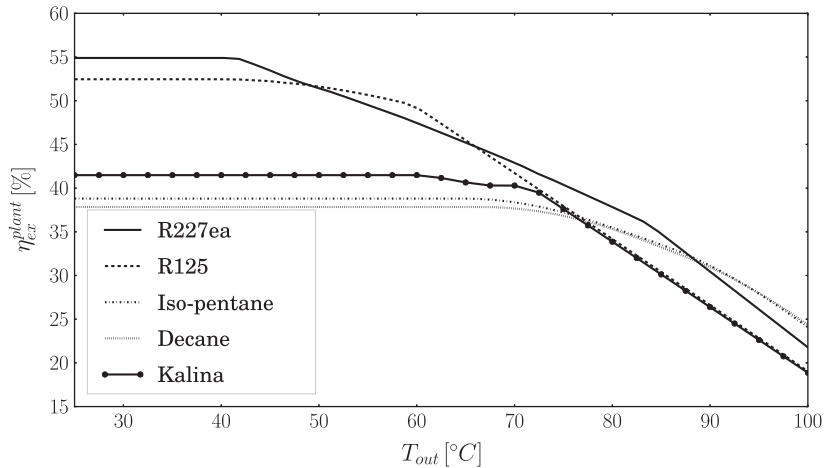


Fig. 15. Exergetic efficiency of R227ea, R125, isopentane, decane and Kalina in function of the limit on the brine outlet temperature. The brine inlet temperature is fixed at 125 °C.

The fluids shown are the best ones for a certain range in brine outlet temperature limit and a comparison with the Kalina cycle is made. For relatively low brine outlet temperature limits, the best fluids for the unconstrained case (low critical temperature) perform the best. But when the constrained brine outlet temperatures become high, fluids which perform badly in the unconstrained case (high critical temperature) become the better ones. Up to an outlet temperature of 60 °C, the efficiency is decreased by only 10%, but for higher temperatures the efficiency decreases strongly, because the available temperature drop becomes small. For constrained brine outlet temperatures around 70 °C, the Kalina cycle performs equally well as the best ORCs, but for lower or higher temperatures the ORCs are the better choice.

5.4.3. Condenser temperature

For the ORCs the condenser temperature is varied from 20 to 35 °C. Fig. 16 shows the exergetic plant and energetic cycle efficiency for a simple cycle with R227ea in comparison to the standard case ($T_{cond} = 25$ °C). Also the outlet temperature of the brine is shown. The energetic cycle efficiency decreases with increasing condenser temperature, because the cycle low temperature heat reservoir increases in temperature (T_{cond}). The brine outlet temperature increases almost linearly with the condenser temperature; for R227ea the temperature difference between the condenser temperature and the brine outlet temperature is about 15 °C.

The exergetic plant efficiency decreases strongly with increasing condenser temperature, because the cycle efficiency decreases and the brine outlet temperature increases. This strong, linear decrease has earlier been described [24]. The figure shows that increasing the condenser temperature with 10 °C decreases the plant efficiency with about 20%.

5.4.4. Minimum temperature difference in HX

The minimum temperature difference in the heat exchanger between brine and working fluid HX is varied from 1 to 20 °C for the unconstrained case. The exergetic plant efficiency, energetic cycle efficiency and brine outlet temperature for cycles with R227ea are shown in Fig. 17. In the reference state for the efficiencies, the temperature difference is 5 °C.

For low temperature differences, the cycle is of the transcritical type. The cycle efficiency is constant, because the cycle itself (pressure, maximum temperature) does not change. So, the variation in plant efficiency is completely determined by the variation in brine outlet temperature for this transcritical cycle.

At a temperature difference of about 8.5 °C there is a sudden drop in the cycle efficiency and an increase in the outlet temperature due to the transition from a transcritical to a subcritical cycle for this fluid. For the subcritical cycle the cycle efficiency does depend on the temperature difference, because the pinch point is at the beginning of the evaporator. When the temperature difference

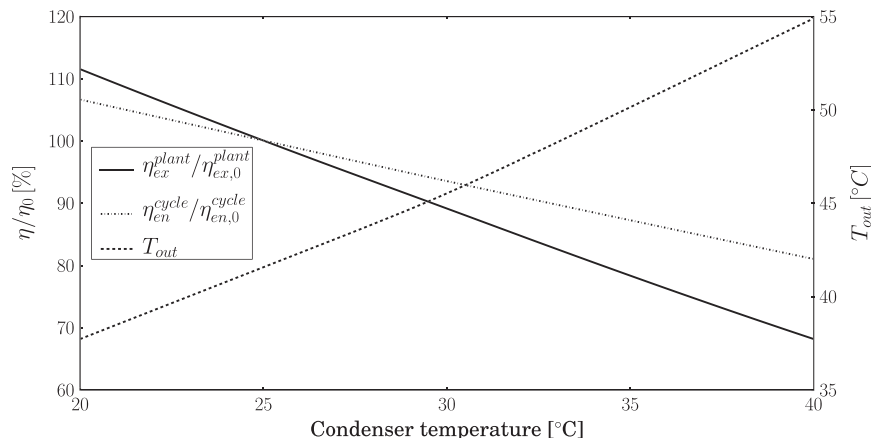


Fig. 16. Exergetic plant efficiency η_{ex}^{plant} , energetic cycle efficiency η_{en}^{cycle} and brine outlet temperature in function of the condenser temperature for a cycle with R227ea. $\eta_{ex,0}^{plant}$ and $\eta_{en,0}^{cycle}$ are the exergetic plant and energetic cycle efficiency when the condenser temperature is 25 °C, respectively.

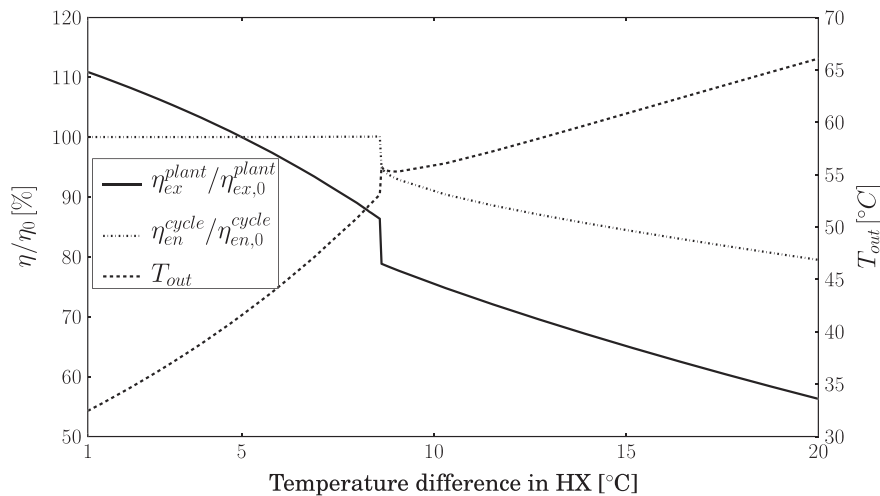


Fig. 17. Exergetic plant efficiency, energetic cycle efficiency and brine outlet temperature for cycles with R227ea as a function of the temperature difference in the heat exchanger *HX*.

increases, the maximum pressure of the cycle has to decrease to allow a reasonable brine cooling efficiency.

6. Conclusions

For a geothermal power plant, it is important to maximize the plant efficiency, which is the product of the cycle efficiency and the brine cooling efficiency. So, the brine should be cooled down as much as possible and the heat added to the cycle should be converted to mechanical power as efficient as possible. The exergetic plant efficiency η_{ex}^{plant} describes how good a certain cycle approaches the ideal one.

Transcritical and multi-pressure subcritical ORCs are in most cases the best performing cycles and can achieve exergetic plant efficiencies of more than 50%. They outperform the investigated Kalina cycle for low minimum brine outlet temperatures. For brine outlet temperatures around 70 °C, the best ORCs and the Kalina perform similarly.

When there is no constraint on the brine outlet temperature, recuperation or turbine bleeding are not useful. Both techniques improve the cycle efficiency, but the brine cooling efficiency decreases and the combination of these two effects has zero influence on the plant efficiency. A constraint on the brine outlet temperature causes a strong decrease in the power plant output, but in this case the decrease can partly be compensated by the use of recuperation or turbine bleeding. The Kalina cycle performs relatively better when the brine outlet temperature is limited.

The choice of the optimum working fluid depends strongly on the conditions. In the single-pressure, unconstrained case, fluids with a low critical temperature perform the best and a transcritical cycle is optimum. When an extra pressure level is allowed or a constraint is added to the brine outlet temperature, fluids with higher critical temperatures become relatively better and (multi-pressure) subcritical cycles are equally good or better than transcritical cycles.

The maximum theoretical temperature drop of the brine is low, because of the low brine inlet temperature. A small increase in the condenser temperature or the pinch point temperature difference, have a relatively strong influence on the available temperature drop of the brine and therefore also on the plant efficiency. So, for low temperature heat sources, it is very important to have a low condenser temperature and pinch point temperature difference.

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