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Comparison of Thermodynamic Cycles for Electricity Production from Low-Temperature Geothermal Heat Sources

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Abstract

The performance of different types of organic Rankine cycles (ORC) and of the Kalina cycle is investigated for low-temperature (100-150°C) geothermal heat sources. A variety of configurations is worth considering. The ORC's can be subcritical or transcritical and can have one or more pressure levels. Each cycle can be a standard one, have recuperation or turbine bleeding. Comparison of these cycles 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 limit 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 important.

Keywords: Geothermal, ORC, Kalina

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, two types of 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. ORC's are used for different low-temperature heat sources (biomass, geothermal, solar, ...) [3].

Different configurations have been proposed in the literature. Cayer et al. [4] have optimized a transcritical ORC for a heat-source temperature of 100°C. CO_2, ethane and R125 are used as possible working fluids. Only the simple ORC configuration with a pump, heat exchanger, turbine and condenser are considered. Kanoglu [5] has investigated multi-pressure cycles. An exergy analysis of an existing power plant, which uses the combination of 2 subcritical cycles with isopentane, has been performed. Gnutek and Bryszewska-Mazurek [6] have considered a multi-cycle ORC. The power plant consists of 4 subcritical power cycles, using the heat source in series. Heberle and Brüggemann [7] 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. [8] have compared standard ORC's with ORC's 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 standard cycle. Different configurations of subcritical ORC's have been compared by Yari [9]. Standard cycles, cycles with recuperation, cycles with turbine bleeding and cycles with recuperation and turbine bleeding are simulated. 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. [10] 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.

For low-temperature sources, the Kalina is often mentioned as an alternative for the ORC. The Kalina cycle is similar to an ORC, but uses a mixture of water and ammonia with a variable composition as working fluid [11–13]. Although the Kalina cycle is often called to be superior to the ORC, DiPippo [14] has shown that an existing Kalina cycle has about the same performance as existing ORC's.

In this paper different types of ORC's and the Kalina cycle are thermodynamically optimized and compared, but only pure fluids are used for the ORC's. In section 2, different types of efficiencies are defined and compared to each other. Energetic and exergetic efficiencies for the plant and the thermodynamic cycle are defined. In the third section, different cycle configurations are described for the ORC's and the Kalina cycle. In section 4, these configurations are optimized using different organic fluids for different cases. The influence of some parameters is 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* is defined as the fraction of the net mechanical power output to the heat input:

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

$\dot{W}_{net}$ 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

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

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. So, equation (1) can be written as:

$$\eta_{en}^{cycle} = \frac{\dot{W}_{net}}{\dot{m}_{brine}(h_{in} - h_{out})} \tag{3}$$

Analogous to the energetic cycle efficiency, also an *exergetic cycle efficiency* can be defined:

$$\eta_{ex}^{cycle} = \frac{\dot{W}_{net}}{\dot{m}_{brine}(e_{in} - e_{out})} \tag{4}$$

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 is defined as:

$$e = (h - h_0) - T_0(s - s_0), \tag{5}$$

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. The advantage of the exergetic cycle efficiency is that it is the fraction of real work output to the maximum obtainable output and therefore is an indication how far away the cycle is from the ideal one.

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 and exergetic *plant efficiencies* are defined, respectively:

$$\eta_{en}^{plant} = \frac{\dot{W}_{net}}{\dot{Q}_{available}} = \frac{\dot{W}_{net}}{\dot{m}_{brine}(h_{in} - h_0)} \tag{6}$$

$$\eta_{ex}^{plant} = \frac{\dot{W}_{net}}{\dot{E}_{available}} = \frac{\dot{W}_{net}}{\dot{m}_{brine} e_{in}} \tag{7}$$

$\dot{Q}_{available}$ and $\dot{E}_{available}$ are the heat and exergy in the heat source, respectively. The difference with Eqs. (3) and (4) is that h_{out} and e_{out} have been replaced by h_0 and $e_0 = 0$, respectively. These plant efficiencies can be rewritten in function of the respective cycle efficiencies:

$$\eta_{en}^{plant} = \frac{\dot{W}_{net}}{\dot{m}_{brine}(h_{in} - h_{out})} \; \frac{\dot{m}_{brine}(h_{in} - h_{out})}{\dot{m}_{brine}(h_{in} - h_0)} \tag{8}$$

$$= \eta_{en}^{cycle} \eta_{en}^{cooling} \tag{9}$$

$$\tag{10}$$

$$\eta_{ex}^{plant} = \frac{\dot{W}_{net}}{\dot{m}_{brine}(e_{in} - e_{out})} \; \frac{\dot{m}_{brine}(e_{in} - e_{out})}{\dot{m}_{brine} e_{in}} \tag{11}$$

$$= \eta_{ex}^{cycle} \eta_{ex}^{cooling} \tag{12}$$

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 $\frac{\dot{W}_{net}}{\dot{m}_{brine}}$ should be maximized. As seen from Eqs. (6) and (7) 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.

Equations (9) and (12) show that the heat source should be cooled down as far as possible (i.e., $\eta_i^{cooling} \rightarrow 1$) *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 (ORC's) and the Kalina cycle will be described and optimized for maximum plant efficiency.

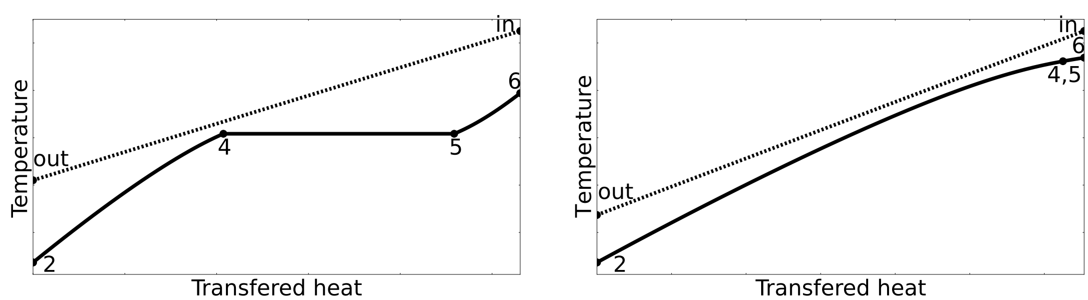
3 Organic Rankine Cycle

The Organic Rankine Cycle (ORC) is a Rankine cycle with an organic working fluid. Many organic fluids have lower critical temperatures than water and are better suited for power production from low-temperature heat sources. In this section, different types of ORC's are described.

3.1 Subcritical and Transcritical ORC's

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.

Figure 1 shows the temperature-heat diagram of a subcritical and a transcritical cycle. 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.



(a) Subcritical (b) Transcritical

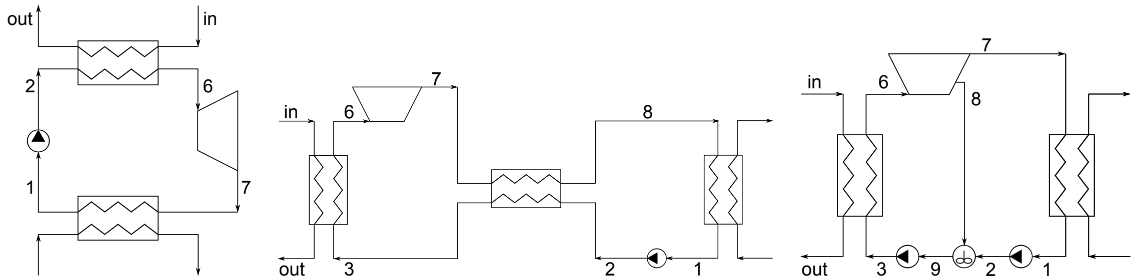
Figure 1: Temperature-heat diagrams of a subcritical (a) and transcritical (b) cycle. The working fluid (–) is heated by the cooling brine (···).

3.2 ORC Configurations

Three ORC configurations are investigated, as shown in figure 2. These three configurations can be subcritical or transcritical, depending on the maximum pressure in the cycle. In the standard ORC (figure 2a), the working fluid is pumped from the low to the high pressure (1→2), heated by the brine (2→6), expanded in the turbine (6→7) and cooled in the condenser (7→1). Often, state 7 is still at a high temperature and a recuperator can be used as shown in figure 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 figure 2c. Part of the working fluid (state 8) is extracted from the turbine at

[1] The so-called "supercritical steam cycle" is actually a transcritical cycle according to this definition.

an intermediate pressure and is mixed with state 2 to form state 9. Then the working fluid is pumped to the high pressure (9→3) and the rest of the cycle is analogous to the standard ORC.



(a) Standard ORC (b) ORC with recuperation (c) ORC with turbine bleeding

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

3.3 Number of Pressure Levels

The cycles shown in figure 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. Figure 3a shows a standard 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. Figure 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 $p_a < p_b < p_c$. For the example in the figure, the working fluid heating curve exists of the following parts:

I→II Heating of sub-cycles a, b, and c

II→III Evaporation of sub-cycle a

III→IV Heating of sub-cycles b and c

IV→V Evaporation of sub-cycle b

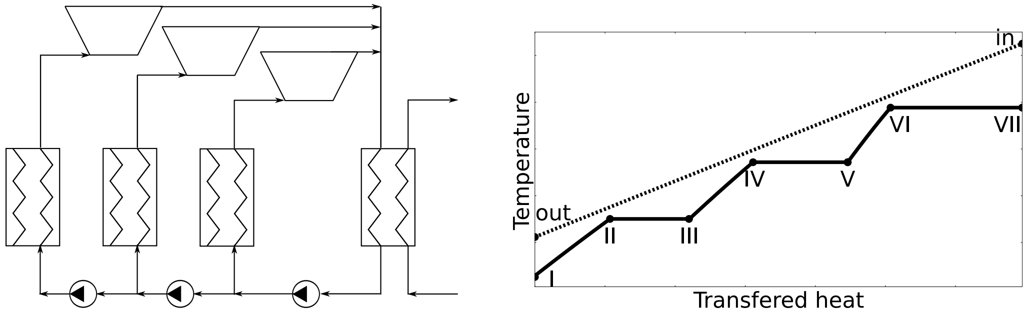
V→VI Heating of sub-cycle c

VI→VII Evaporation of sub-cycle 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 ORC’s

For the modeling of the ORC’s, 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



(a) Triple-pressure standard ORC (b) Temperature-heat diagram

Figure 3: Scheme and temperature-heat diagram of a triple-pressure cycle. The working fluid (–) is heated by the cooling brine (· · ·).

for all types of fluids (even for the so-called dry[2] 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-ε) 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:

$$h_2 = h_1 + \frac{h_{2s} - h_1}{\eta_P} \tag{13}$$

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:

$$\dot{W}_P = \dot{m}_{wf}(h_2 - h_1) \tag{14}$$

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

$$\begin{aligned} \dot{I}_P &= (\dot{m}_{wf}e_1 + \dot{W}_P) - (\dot{m}_{wf}e_2) \\ &= \dot{m}_{wf}T_0(s_2 - s_1) \end{aligned} \tag{15}$$

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

$$\dot{W}_P = (1-\varepsilon)\dot{m}_{wf}(h_2 - h_1) + \varepsilon\dot{m}_{wf}(h_3 - h_9) \tag{16}$$

$$\dot{I}_P = (1-\varepsilon)\dot{m}_{wf}T_0(s_2 - s_1) + \varepsilon\dot{m}_{wf}T_0(s_3 - s_9) \tag{17}$$

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

$$h_7 = h_6 - (h_6 - h_{7s})\eta_T \tag{18}$$

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

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:

$$\dot{W}_T = \dot{m}_{wf}(h_6 - h_7) \tag{19}$$

$$\dot{I}_T = \dot{m}_{wf}T_0(s_7 - s_6) \tag{20}$$

For the cycle with turbine bleeding, these equations are:

$$\dot{W}_T = \dot{m}_{wf}(h_6 - h_8) + (1 - \varepsilon)\dot{m}_{wf}(h_8 - h_7) \tag{21}$$

$$\dot{I}_T = \dot{m}_{wf}T_0(s_8 - s_6) + (1 - \varepsilon)\dot{m}_{wf}T_0(s_7 - s_8) \tag{22}$$

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

$$\dot{W}_{net} = \dot{W}_T - \dot{W}_P \tag{23}$$

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:

$$\dot{m}_{brine}(h_{in} - h_{out}) = \dot{m}_{wf}(h_6 - h_3) \tag{24}$$

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

$$(h_7 - h_8) = (h_3 - h_2) \tag{25}$$

and the minimum temperature difference is given by ΔT_{min}^{recup}. The pinch point temperature differences ΔT_{min}^{HX} and ΔT_{min}^{recup} are generally not the same. The energy equation for the condenser in the standard or recuperated cycle is given by

$$\dot{m}_{wf}(h_x - h_1) = \dot{m}_{cooling}(h_{out}^{cooling} - h_{in}^{cooling}) \tag{26}$$

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

The irreversibilities created in the HX and recuperator are:

$$\begin{aligned}\dot{I}_{HX} &= (\dot{m}_{brine}e_{in} + \dot{m}_{wf}e_3) - (\dot{m}_{brine}e_{out} + \dot{m}_{wf}e_6) \\ &= \dot{m}_{brine}T_0(s_{out} - s_{in}) + \dot{m}_{wf}T_0(s_6 - s_3) \qquad (27)\\ \dot{I}_{recup} &= \dot{m}_{wf}[(e_2 + e_7) - (e_3 + e_8)] \\ &= \dot{m}_{wf}T_0[(s_3 - s_2) + (s_8 - s_7)] \qquad (28)\\ & \qquad (29)\end{aligned}$$

The irreversibilities created in the cooling installation are given by

$$\dot{I}_{cool} = \dot{m}_{wf}(e_x - e_1) \tag{30}$$

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

$$\dot{I}_{cool} = (1 - \varepsilon)\dot{m}_{wf}(e_7 - e_1) \tag{31}$$

3.4.3 Mixing

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

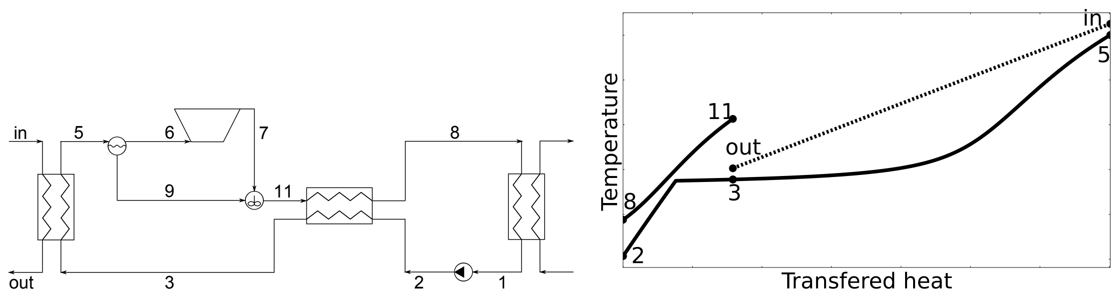
$$\varepsilon h_8 + (1-\varepsilon)h_2 = h_9 \tag{32}$$

and the irreversibility created is:

$$\begin{aligned} \dot{I}_{mix} &= \dot{m}_{wf}\left([(1-\varepsilon)e_2 + \varepsilon e_8] - e_9\right) \\ &= \dot{m}_{wf} T_0 (s_9 - [(1-\varepsilon)s_2 + \varepsilon s_8]) \end{aligned} \tag{33}$$

4 Kalina Cycle

The Kalina cycle is a type of Rankine cycle, where a mixture of ammonia and water is used and the concentration of ammonia changes in the cycle. The scheme of a typical Kalina cycle, which always contains a recuperator, is shown in figure 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.



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

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

Figure 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 ORC's 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:

$$\varepsilon = \frac{x_{int} - x_{low}}{x_{high} - x_{low}} \tag{34}$$

5 Results and Discussion

In this section, the results of the comparison between the ORC configurations of Figs. 2 & 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 [15].

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.

T_{in}	125°C
p_{in}	5 bar
T_{cond}	25°C
T_0	15°C
p_0	1 bar

η_P	80 %
η_T	85 %
ΔT_{min}^{HX}	5°C
ΔT_{min}^{recup}	5°C
ΔT_{min}^{cond}	5°C

Table 1: Input parameters.

For some fluids available in RefProp, only supercritical cycles are possible and they are therefore skipped. All other fluids, even those which are legally prohibited, are used in the calculations to obtain the thermodynamic optimum cycle configuration. 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 Net Power Output

In this section, two cases are investigated. In the first case, there is no limit 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).

5.2.1 Brine cooling temperature without limit

a Single pressure

Figure 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 standard configuration. The fluids are ordered from the best (R227ea) to the worst fluid (C5F12) of the 20 (in terms of exergetic plant efficiency, which is the characteristic with most "value"). Although the exergetic plant efficiency decreases monotonically, the energetic cycle efficiency does not. It is seen from Eqs. (9) and (12) 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.

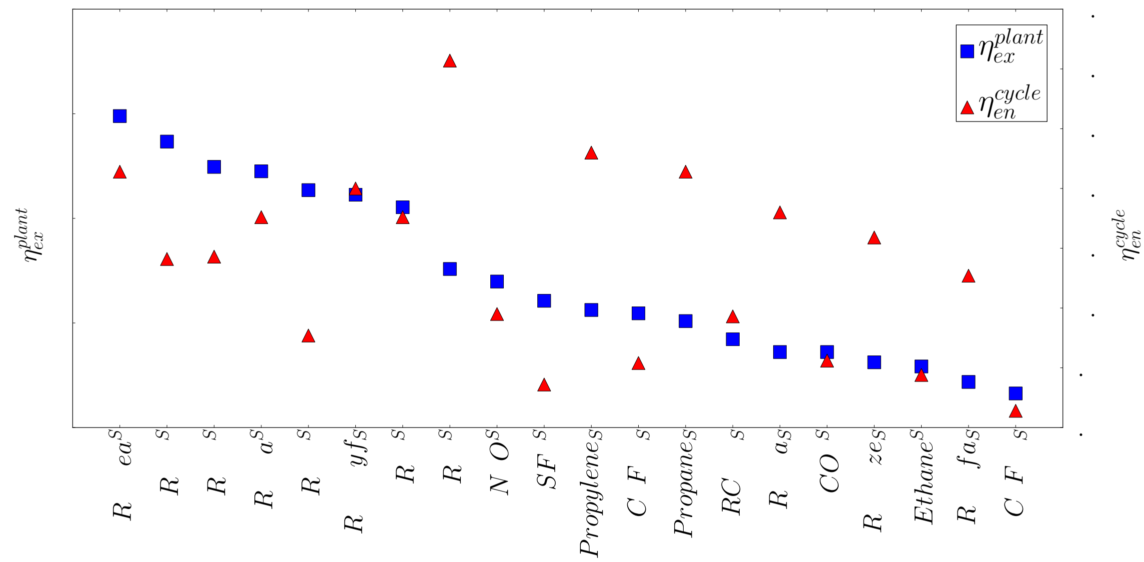


Figure 5: Exergetic plant efficiency and energetic cycle efficiency of the 20 fluids with the highest mechanical power output in a single pressure level cycle for the standard (S) configuration. Subcritical cycles are denoted with a *sub*script S and transcritical cycles with a *super*script S.

In this figure, only cycles with a standard 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 transfered 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.

All pure fluids condense at a constant temperature, so that the cooling water temperature increases 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 thermodynamicaly outperformed by standard subcritical ORC's. 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 figure 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.

Figure 6 shows the net mechanical power production and irreversibilities as a fraction of the incoming brine exergy for 3 optimized cycles. The first one is a transcritical cycle with R227ea, the second one is a subcritical cycle with Propylene and the third one is the Kalina cycle. The cycle with R227ea performs 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.

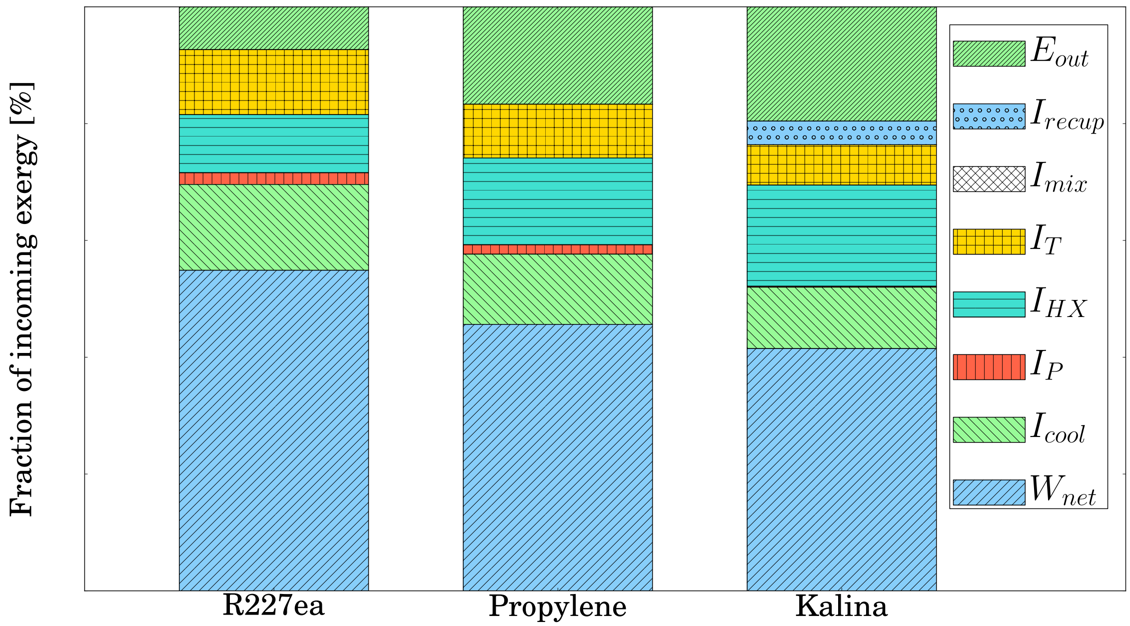


Figure 6: Net mechanical power production and irreversibilities created, as fraction of the incoming brine exergy for the optimum transcritical cycle with R227ea, the optimum subcritical cycle with Propylene and the optimum Kalina cycle.

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 ($\dot{I}_{HX}$ and $\dot{I}_{recup}$). In the Kalina cycle, the lower irreversibilities generated in the condenser (temperature glide), turbine and pump cannot compensate this.

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 standard subcritical ORC's even outperform the Kalina cycle considered in this context. This comparison is only thermodynamical, so no economic conclusions are made.

b Dual pressure

It is also possible to use more than 1 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 figure 7. Again, only the standard 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 2 pressure

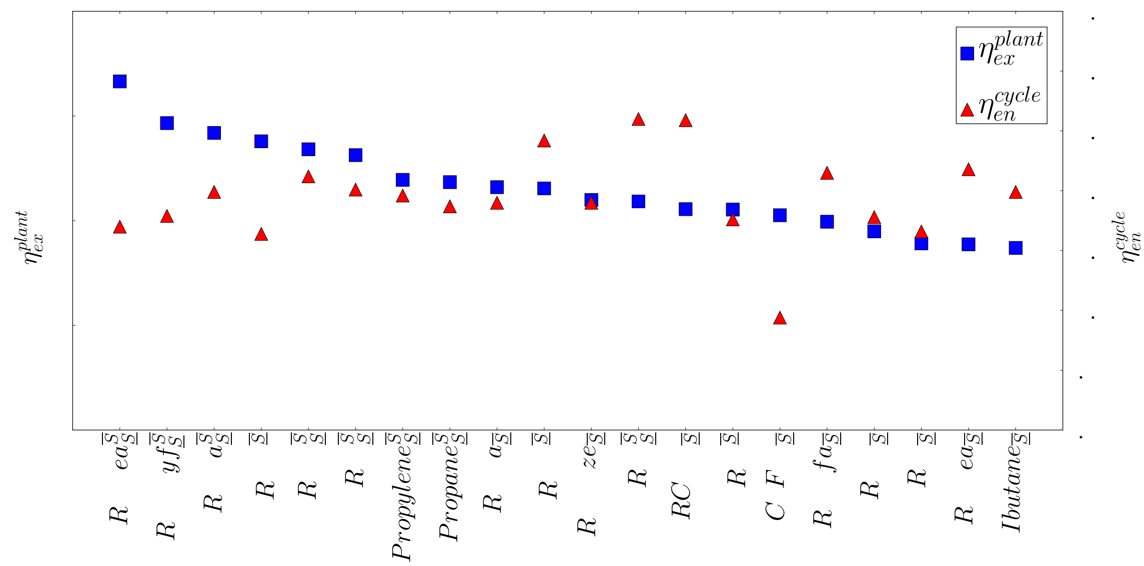


Figure 7: Exergetic plant efficiency and energetic cycle efficiency of the 20 fluids with the highest mechanical power output in a dual-pressure level cycle for the standard configuration. Subcritical cycles are denoted with a *sub*script S and transcritical cycles with a *super*script S. The low-pressure sub-cycle is underlined, the high-pressure sub-cycle has an overbar.

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 level cycles (figure 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.

c Triple and multi-pressure

Figure 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. Figure 9 shows the exergetic efficiency of a multi-pressure cycle with isobutane for different pressure levels. The improvement of the addition of an extra pressure level is shown on the right hand side. This improvement is defined as:

$$\epsilon_i = \frac{\dot{W}^i_{net} - \dot{W}^{i-1}_{net}}{\dot{W}^{i-1}_{net}} \tag{35}$$

with ϵ_i the improvement of addition of the ith pressure level and $\dot{W}^i_{net}$ 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

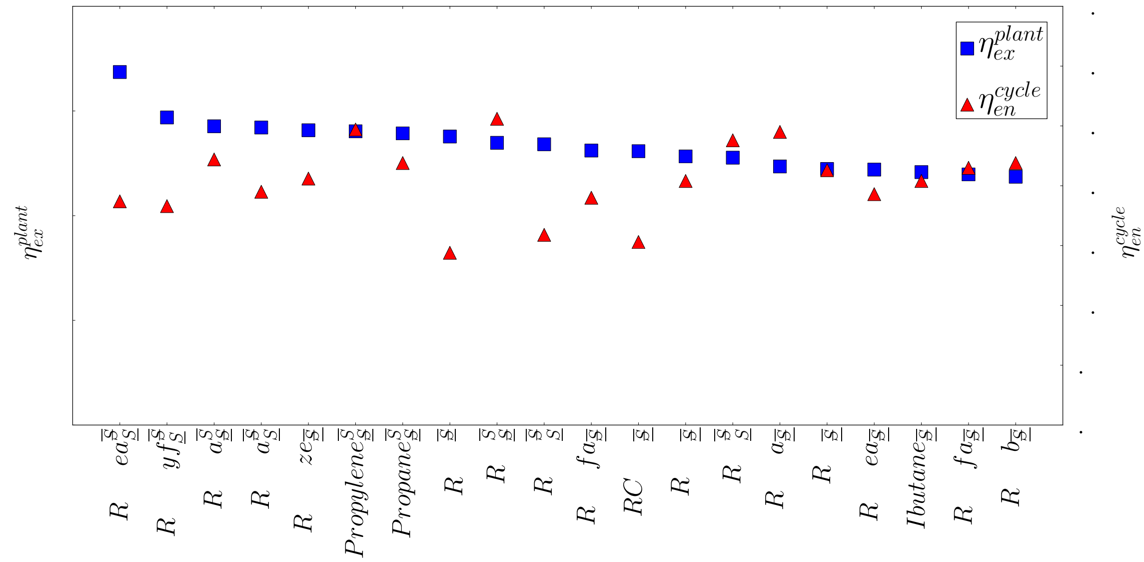


Figure 8: Exergetic plant efficiency and energetic cycle efficiency of the 20 fluids with the highest mechanical power output in a triple-pressure level cycle for the standard configuration. Subcritical cycles are denoted with a *sub*script S and transcritical cycles with a *super*script S. The low-pressure sub-cycle is underlined, the intermediate pressure sub-cycle is lined through and the high-pressure sub-cycle is overlined.

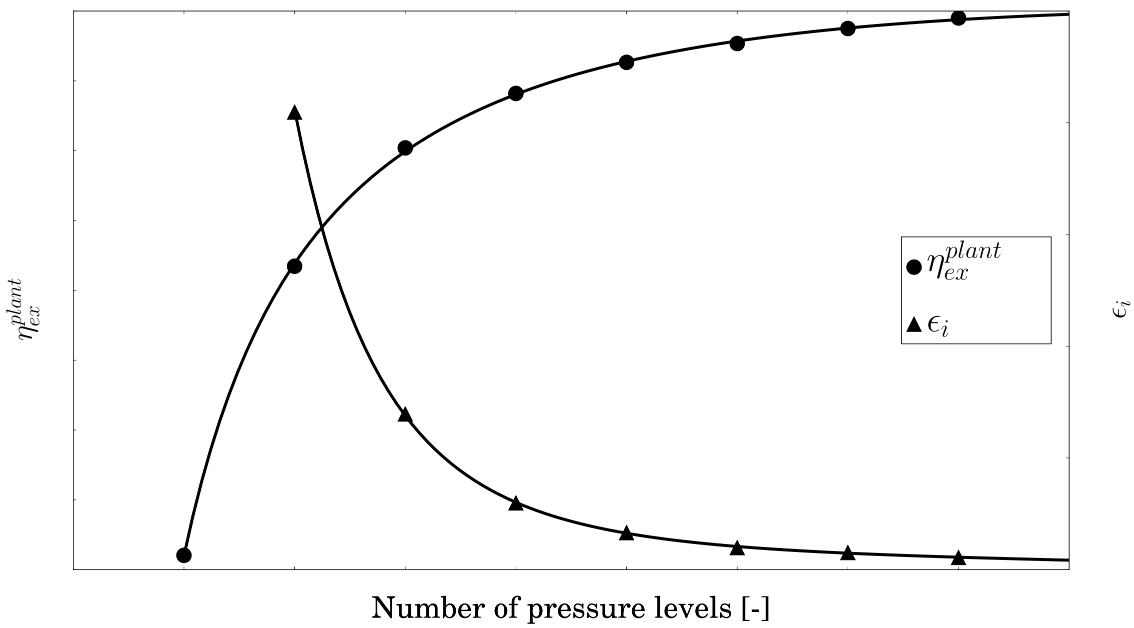


Figure 9: Exergetic plant efficiency (○) of a multi-pressure standard cycle with isobutane in function of the number of pressure levels. The improvement of an extra pressure level is shown (△).

is thermodynamically useful, but the effect decreases with increasing number of pressure levels.

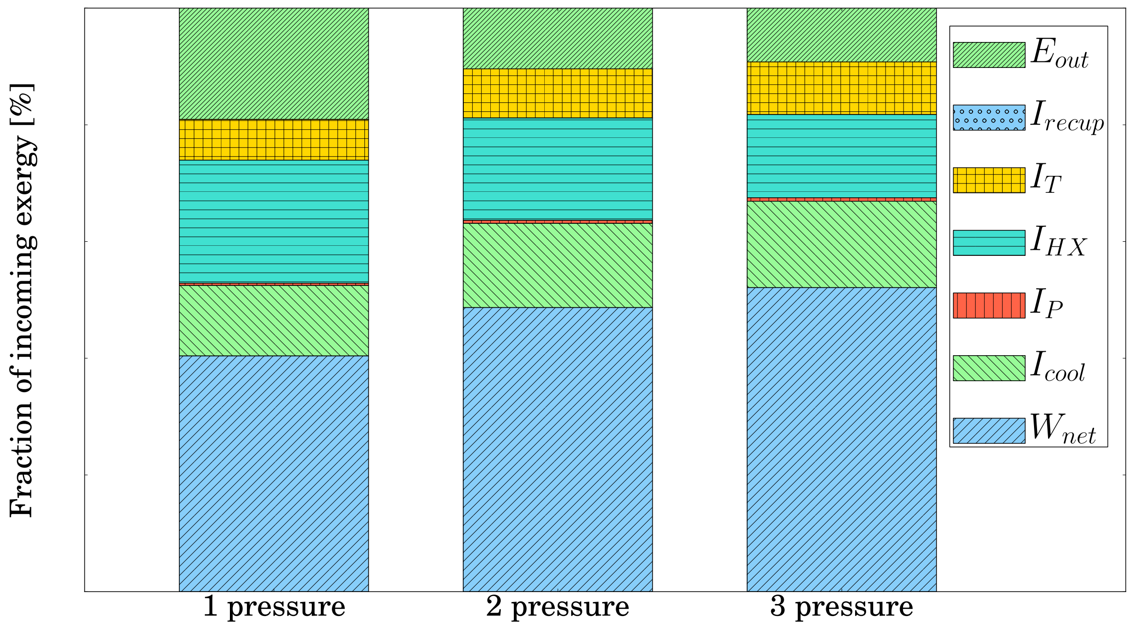


Figure 10: Net mechanical power production and irreversibilities created, as fraction of the incoming brine exergy for the optimum single, dual and triple-pressure cycle with isobutane.

Figure 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.

5.2.2 Limit on brine outlet temperature

Figure 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 (R) or turbine bleeding (B) can have higher power outputs than the standard one (S). This follows from equation (12). The brine cooling is limited to a temperature which is higher than the brine cooling temperature in the case without limit. 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 figure 11. As seen in figure 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 unlimited 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,

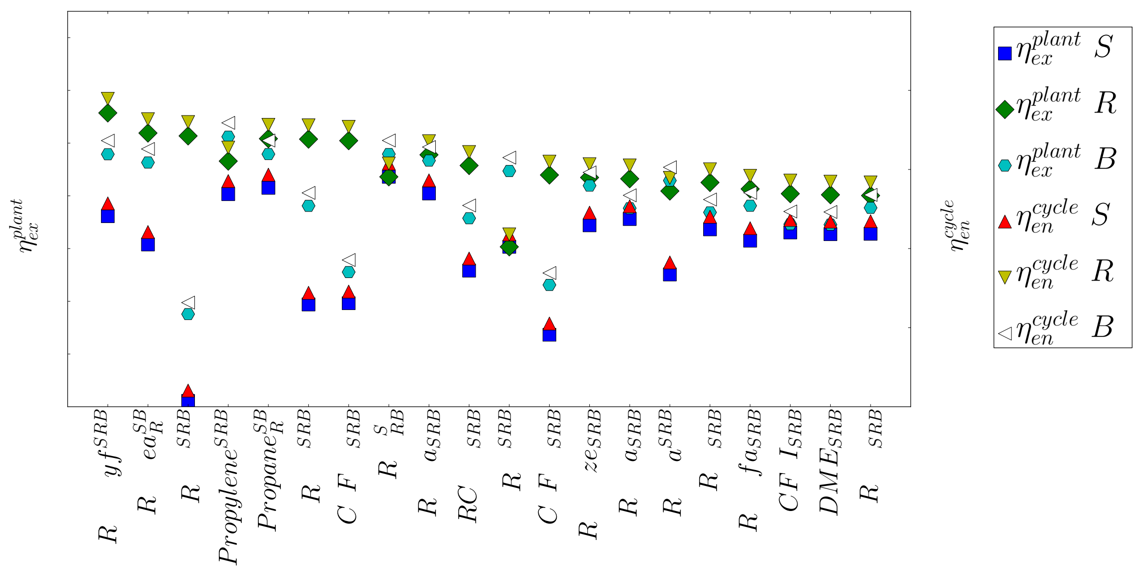


Figure 11: Exergetic plant efficiency and energetic cycle efficiency of the 20 fluids with the highest mechanical power output in a single pressure level cycle with a limited brine outlet temperature of 75°C. S stands for the standard cycle, R for the cycle with recuperation and B for the cycle with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

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.

For the optimization of the Kalina cycle with a limiting 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 unlimited case. The performance of the Kalina cycle is in this case similar to the best ORC's, mainly because of the temperature glide in the condenser which allows a lower minimum temperature in the condenser. When the minimum condenser temperature is limited to 25°C, an exergetic efficiency of 37% is obtained.

The effect of an extra pressure level is shown in figure 12. It shows that the 20 best fluids are optimum when using recuperation and the difference between the fluids is very small. Turbine bleeding is almost always better than the standard cycle, except for the very dry fluid perfluorobutane (C4F10).

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 2 pressure levels are almost all subcritical. The addition of more pressure levels has the same effect as in the unlimited case.

5.3 Influence of changing parameters

In this section, the influence of some parameters is investigated. For every influencing parameter, the effect on one or a few of the best ORC's is shown. Comparison with the Kalina cycle is

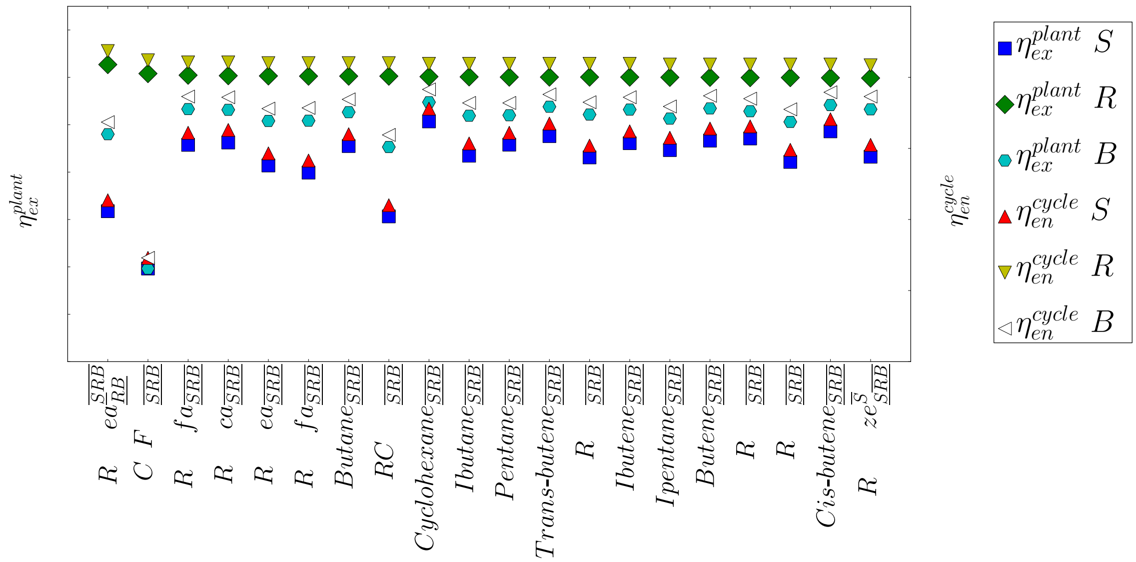


Figure 12: Exergetic plant efficiency and energetic cycle efficiency of the 20 fluids with the highest power output in a dual pressure level cycle with a limited brine outlet temperature of 75°C. S stands for the standard cycle, R for the cycle with recuperation and B for the cycle with turbine bleeding.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

added.

5.3.1 Brine inlet temperature

The brine inlet temperature is varied from 100 to 150°C. In the first case, the brine outlet temperature is unlimited and for every brine inlet temperature, the optimum working fluid is selected. The exergetic efficiency with these fluids is given in figure 13. It is seen that the optimum fluid depends strongly on the brine inlet temperature. For subcritical cycles (low brine inlet temperature), the exergetic plant efficiency of the optimum cycle increases almost linear with the brine inlet temperature. 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 investigated Kalina cycle is added as a comparison and is clearly outperformed by the ORC's. 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.

The influence of the brine inlet temperature on the plant efficiency is also investigated with a limit on the brine outlet temperature of 75°C. These results are given in figure 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 ORC's.

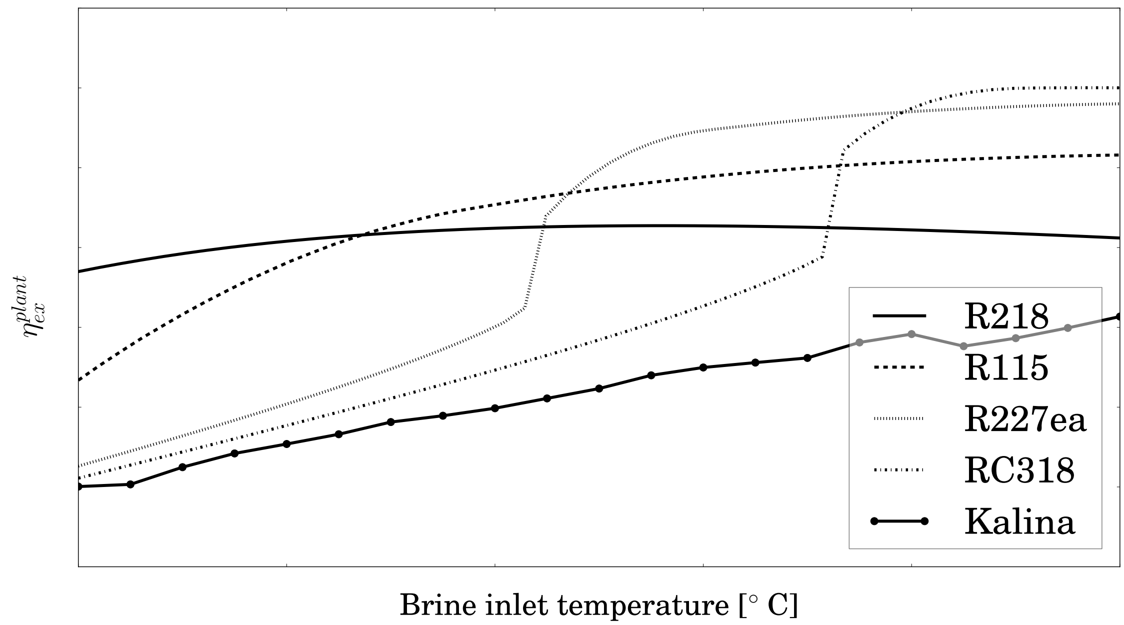


Figure 13: Exergetic efficiency of the temperature-dependent optimum fluids without constraint on the brine outlet temperature. The Kalina cycle is added as a comparison.

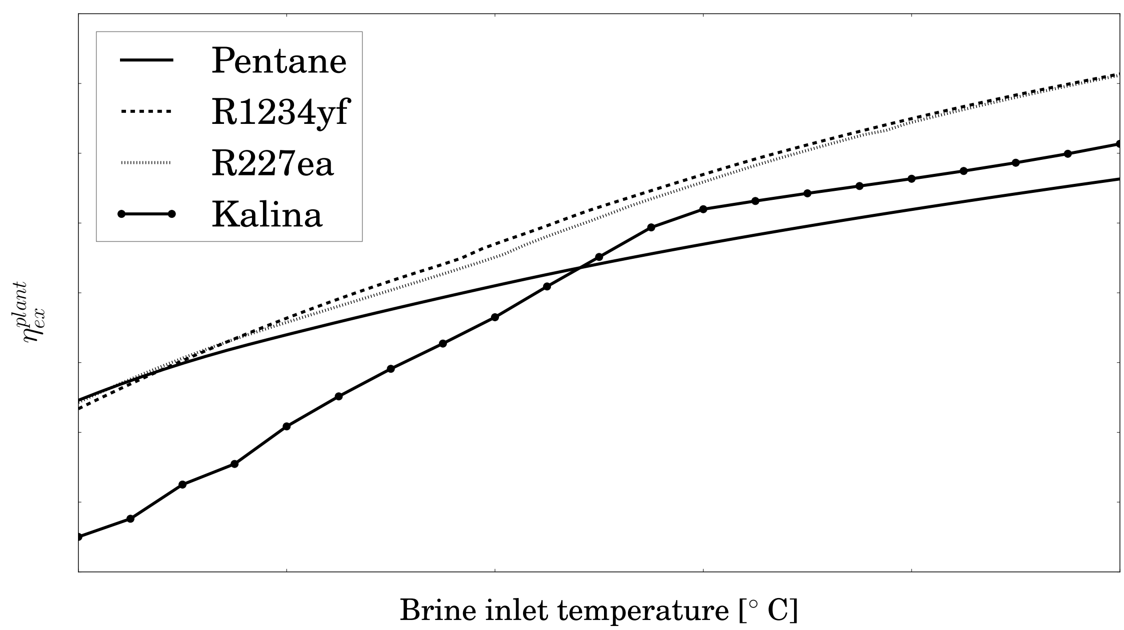


Figure 14: Exergetic efficiency of the brine inlet temperature-dependent optimum fluids (all recuperative), for a brine outlet temperature limit of 75°C. The Kalina cycle is added as a comparison.

5.3.2 Limit on brine temperature

The brine inlet temperature is fixed to 125°C and the brine outlet temperature limit is varied between 25 and 100°C. Figure 15 shows the exergetic efficiency for 4 single-pressure, recuperated ORC's and the Kalina cycle considered. For low temperatures, the brine outlet temperature is higher than the limit, so the exergetic efficiency is the same as in the unconstrained case. Once the limit is above the brine outlet temperature in the unconstrained case, the exergetic efficiency starts to decrease. The rate of this decrease and the temperature at which it starts depends on the fluid used.

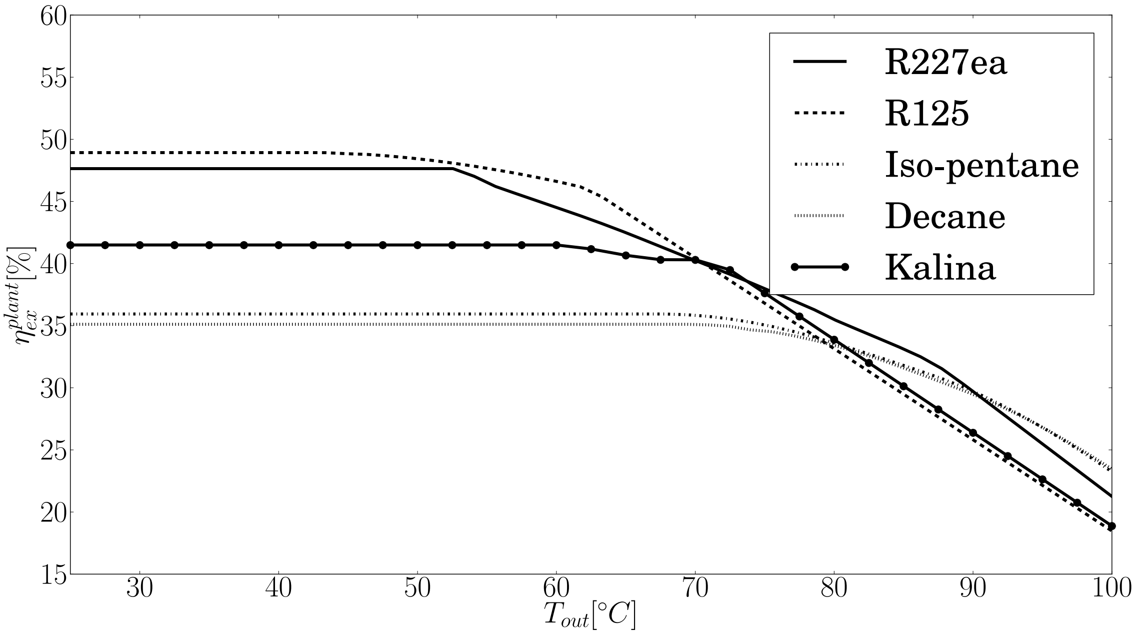


Figure 15: Exergetic efficiency of R227ea, R125, isopentane, decane and Kalina in function of the limit on the brine outlet temperature. The brine inlet temperature is fixed at 125°C.

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 relative low brine outlet temperature limits, the best fluids for the unconstrained case (R227ea and R125) perform the best. But when the limiting temperatures become high, fluids which perform badly in the unconstrained case become the better ones. The exergetic plant efficiency decreases from 55% in the unconstrained case to 25% when a limit of 100°C is used. Up to an outlet temperature of 60°C, the efficiency is decreased by only 10%, but for higher temperatures the efficiency decreases strongly. The available temperature drop of the brine becomes very small and therefore the plant efficiency is small too.

For very low brine outlet temperatures, the Kalina cycle is outperformed by the best ORC's. For temperatures around 70°C, they obtain similar plant efficiencies. For low brine outlet temperatures, the pinch point lies in the evaporator as shown in figure 4b. For higher brine outlet temperatures, the pinch point lies at the end of the economizer. The strange evolution of the exergetic efficiency of the Kalina cycle at a temperature slightly below 70°C is because of the change of the pinch point place.

5.3.3 Condenser temperature

For the ORC's the condenser temperature is varied from 20 to 35°C. Figure 16 shows the exergetic plant and energetic cycle efficiency for a standard 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. With R227ea, the difference between the condenser temperature and the brine outlet temperature is about 15°.

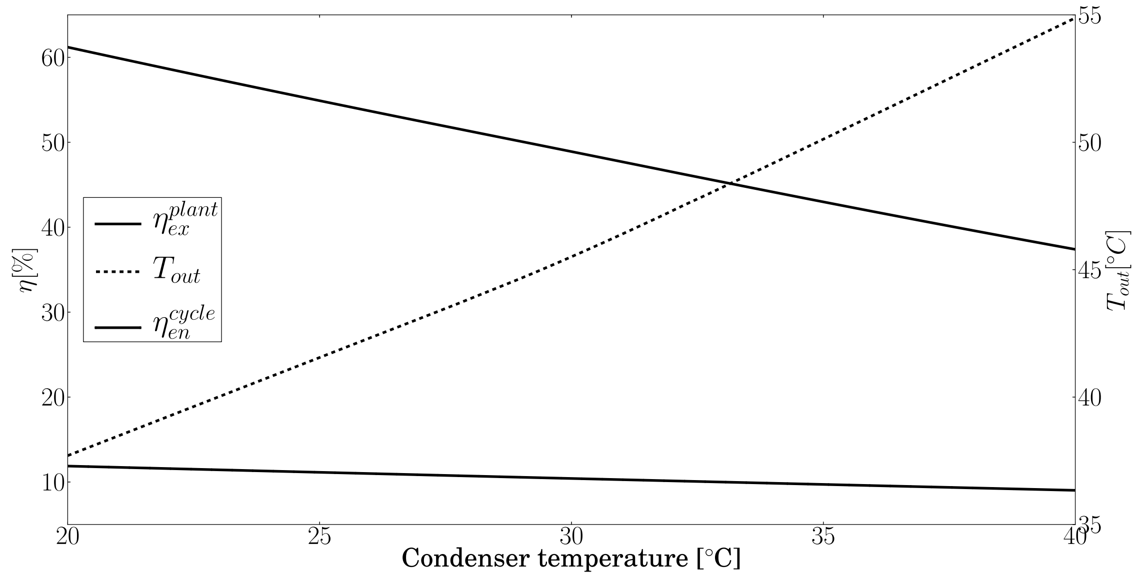


Figure 16: Exergetic plant efficiency, energetic cycle efficiency and brine outlet temperature in function of the condenser temperature for a cycle with R227ea

The exergetic plant efficiency decreases strongly with increasing condenser temperature, because the cycle efficiency decreases and the brine outlet temperature increases. The figure shows that increasing the condenser temperature with 10°C decreases the plant efficiency with about 20%. When the brine outlet temperature is limited, the effect is less as long as the brine outlet temperature limit is relatively high and the condenser temperature is relatively low. This means that the brine outlet temperature can stay constant at the limit and the exergetic plant efficiency is only influenced by the energetic cycle efficiency. In this case, the exergetic plant efficiency decreases with about 20% when the condenser temperature increases with 15°C, which is the same as the effect of the condenser temperature on the cycle efficiency in the unlimited case.

5.3.4 Minimum temperature difference in HX

The minimum temperature difference in the heat exchanger between brine and working fluid is varied from 0 to 20°C for the unconstrained case. The other parameters are the standard ones (table 1). The exergetic plant efficiency, energetic cycle efficiency and brine outlet temperature for cycles with R227ea are shown in figure 17. In the reference state for the efficiencies, the temperature difference is 5°C.

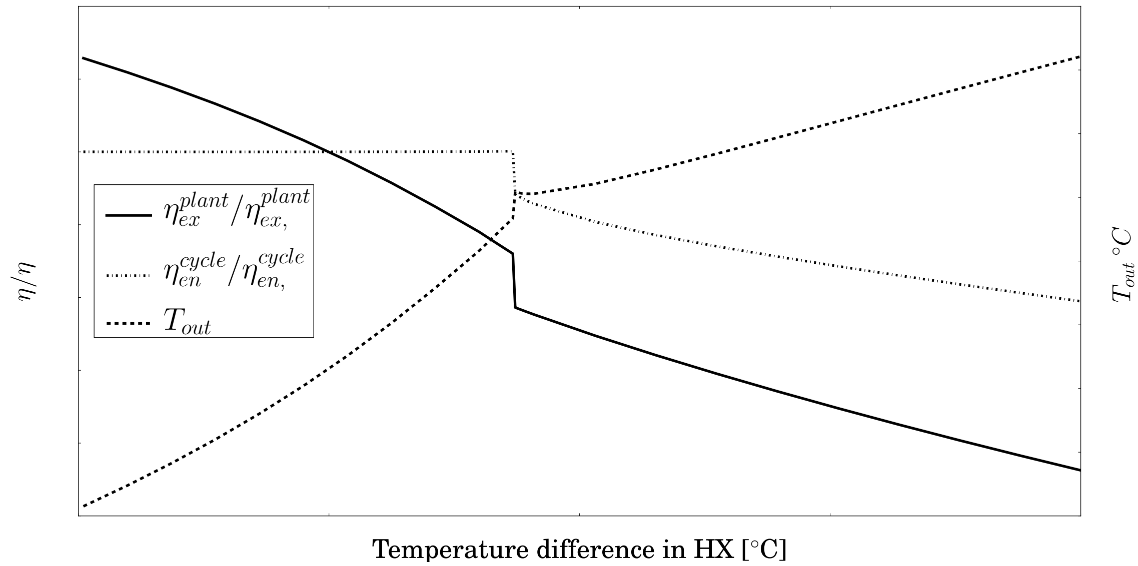


Figure 17: Exergetic plant efficiency, energetic cycle efficiency and brine outlet temperature for cycles with R227ea as a function of the temperature difference in the heat exchanger HX.

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. The temperature difference between brine and working fluid determines the slope of the brine cooling curve in the temperature heat (per unit working fluid) diagram. When the temperature difference becomes lower, the slope increases and the brine can cool down more. 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 increases, the pressure 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 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 ORC's 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 ORC's and the Kalina perform similarly.

When there is no limit 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 limit 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 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.

Acknowledgments

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