An investigation on organic Rankine cycle incorporating a ground-cooled condenser: working fluid selection and regeneration

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Abstract

This research presents an optimization of a power generation system incorporating a ground-cooled condenser. It focuses on the mitigation of the adverse effects of critical parameters such as low mass flow rates and large ground loops. This includes an investigation of different configurations (basic and regenerative cycles) and working fluids to ascertain their effects on the performance of the system. Three refrigerants, R123, R124 and R245fa, were compared in terms of working fluid's performance. The purpose of the system optimization is to increase the net output power whilst reducing the capital cost of installation. At an inlet expander pressure of 3 MPa, as the condensation temperature was decreased from 25°C to 15°C the enhancements in net output power were 7.35%, 12.13% and 8.77% for R123, R124 and R245fa, respectively in the case of basic organic Rankine cycle. However, the highest performance in terms of net output power was recorded for R123 in both configurations under the investigated conditions. Based on the calculations of net output power and heat rejection, the regenerative cycle is highly recommended since it provides significant increase in the output power without considerably changing the amount of heat rejected to the ground.

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Keywords: Shallow geothermal energy, ground-cooled condenser, power generation, organic Rankine cycle, ground heat exchanger.

Nomenclature	
$\dot{Q}$	heat transfer rate (kW)
$\dot{W}$	power (kW)
$\dot{m}$	mass flow rate (kg/s)
ΔT_{lm}	logarithmic mean temperature difference
cp	specific heat (kJ/kg.K)
h	enthalpy (kJ/kg)
η	efficiency
T	temperature (°C)
ε	effectiveness
Abbreviations	
CTRC	CO_2-based transcritical Rankine cycle
DCV	directional control valve
EAHE	earth air heat exchanger
EES	Engineering Equation Solver
GE	geothermal energy
GHE	ground heat exchanger
GPP	geothermal power plant
GSHP	ground source heat pump
ORC	organic Rankine cycle
RES	renewable energy source
Subscripts	
f	working fluid
g	gas
ge	generator
i	inlet

o	outlet
p	pump
r	regenerator
t	turbine
wp	water pump

1. Introduction

Geothermal energy (GE) has been reported as one of the most stable renewable energy sources (RES) since it is almost independent of ambient air temperature. It is neither stochastic nor intermittent compared to other RES such as solar and wind energies [1, 2]. The development of GE systems also encourages to decrease the dependency on fossil fuels and hence, mitigating the corresponding negative environmental effects [3]. Thus, investigating and improving GE systems support the global scope promoting sustainable energy and environmental protection [4, 5]. GE is mainly classified into two types: deep and shallow. The former is typically used for direct heating or activating geothermal power plants (GPPs) [6, 7]. Deep GE systems are very expensive, and they are based on extracting hot geothermal fluid from deep underground layers [8, 9]. The geothermal fluid will be reinjected back to the ground after being utilized. However, shallow GE can be used to provide heating or cooling via ground coupled heat exchangers [10]. Such systems could be found in the form of a ground source heat pump (GSHP) or earth air heat exchanger (EAHE). GSHP is based on the refrigeration cycle, while the ground is considered as the heating or cooling source instead of ambient air compared to the conventional air source heat pump [11, 12]. Thus, it is necessary to use a ground heat exchanger (GHE) to transfer heat from/to the ground [13, 14]. There are different types of GHE configurations such as vertical [15, 16] and horizontal [17, 18]. Coiled GHEs have also been considered as enhancements of the conventional types to decrease the volume of installations. These could be installed in the form of spiral [19] or slinky [20] shape. EAHE is simpler than that of GSHP since it only utilizes an underground duct to circulate air and supply it to the conditioned space via a blower [21, 22].

Many previous studies have investigated the incorporation of geothermal energy in power generation systems using the ground as a source of heat [23]. Such applications utilize deep geothermal energy to activate dry steam [24], binary [25] and flash [26] power cycles depending on the available geothermal conditions. Dry steam and flash GPPs directly utilize the hot geothermal fluid to activate the power cycle. The difference between these cycles is that the geothermal fluid is available in the form of steam and mixture, respectively. While, in the binary cycle GPP, a heat exchanger is used to transfer heat from the geothermal fluid to the working fluid of the power cycle [27]. This cycle is mainly used in case of low-grade GE source. GPPs have also passed through several enhancements to improve the cycle's performance such as the double flash [28, 29], combined flash-binary [30, 31] and regenerative cycle [32, 33]. However, it was also reported that shallow geothermal energy can be used as a cooling source for a power generation system [34, 35]. The ground can provide a lower cooling medium temperature compared to that of ambient air, so that the condenser's pressure could be decreased accordingly. Thus, the net output power can be maximized since it depends mainly on the pressure difference between the gas heater and condenser. In Ref. [34], a new system was proposed incorporating a CO_2-based transcritical Rankine cycle with a ground-cooled condenser. The study covered a wide range of heat source conditions whilst varying the gas temperature from 500ºC to 1500ºC and mass flow rate from 100 to 350 kg/hr. The aim of the study was to maximize the net output power of the cycle considering minimum heat exchangers lengths. Vidhi *et al.* [35] investigated the use of EAHE as a condenser in power plants at low to medium temperature heat sources. The power plant considered in the study was the supercritical Rankine cycle covering heat source temperatures ranging between 125ºC and 175ºC. The heat exchanger was buried under the ground at a depth of 2 m. The EAHE was able to increase the efficiency of the supercritical Rankine cycle by 1% and reduce the daily power fluctuations.

The aim of this paper is to compare the basic and regenerative organic Rankine cycle (ORC) incorporating a ground-cooled condenser. It includes investigating the effect of cycle's conditions and working fluid on the power cycle's performance. The most important parameter studied is the change in condensation temperature since it represents the effect of using the ground-cooled condenser on the cycle's performance. The amount of heat rejected to the ground is also a crucial parameter that needs to be taken into consideration to avoid large ground loop installations. Figure 1 shows the different applications of shallow and deep GE whilst highlighting the system studied in the current work.

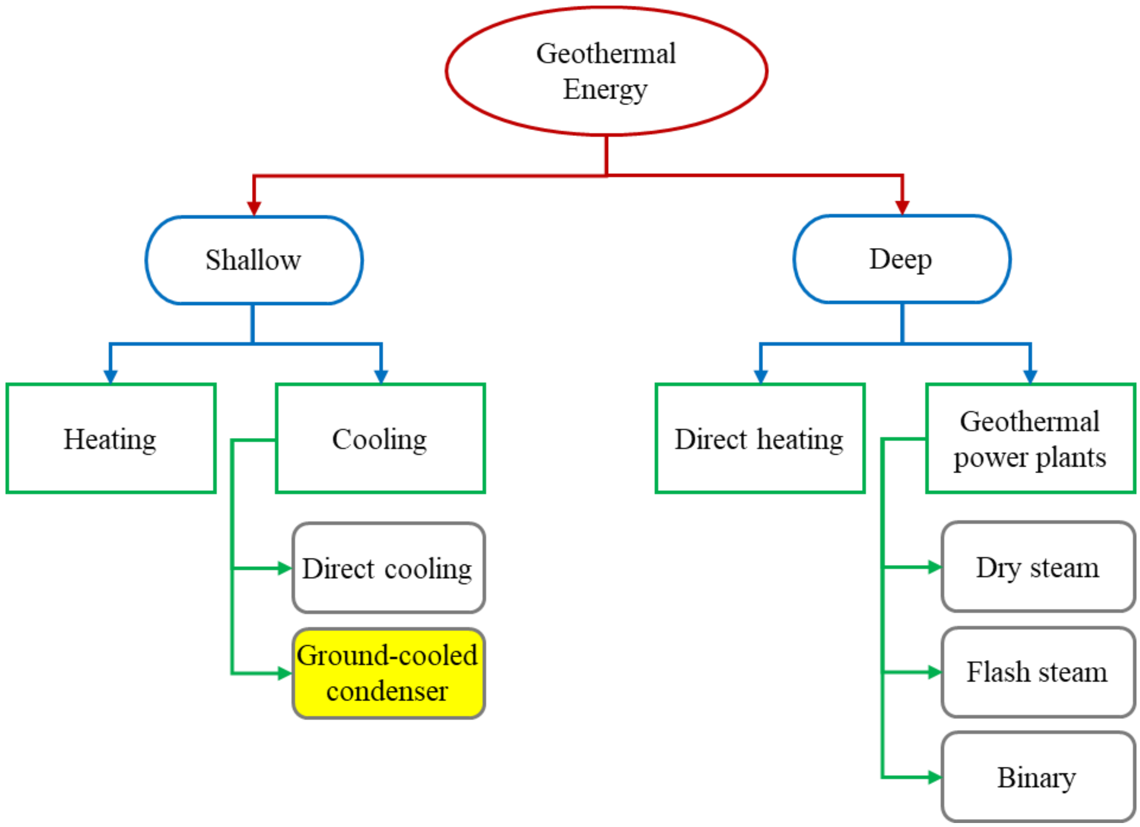
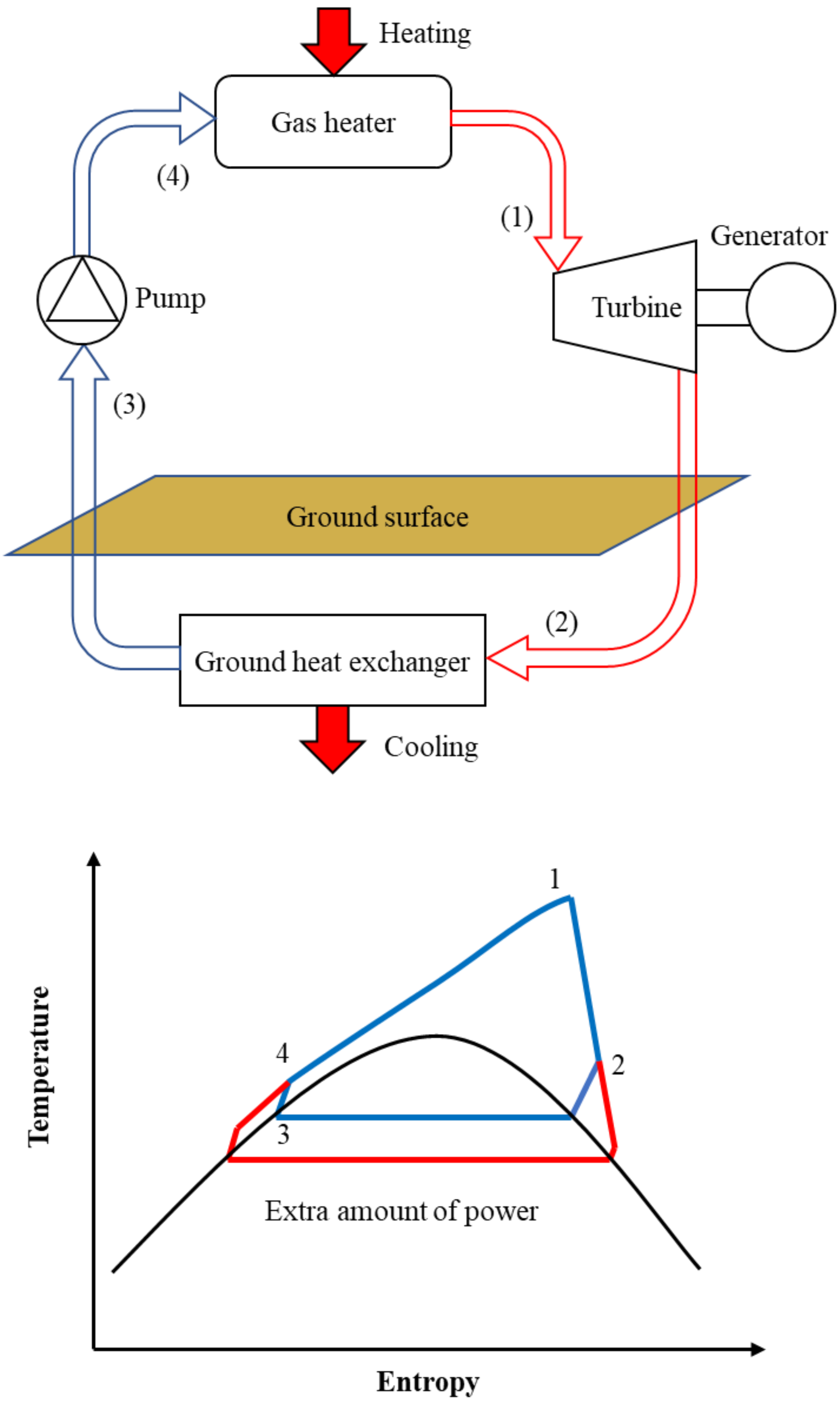


Figure 1: Shallow and deep geothermal energy applications

2. Ground-cooled condenser

The aim of installing a ground-cooled condenser is to extract more amount of power from a low-grade heating source (see Figure 2). The ground provides a better cooling medium than ambient air especially in summer when the latter temperatures are relatively high. The enhancement in the cycle's net output power is displayed in Figure 2b showing the increase in the working fluid's expansion between states 1 and 2.



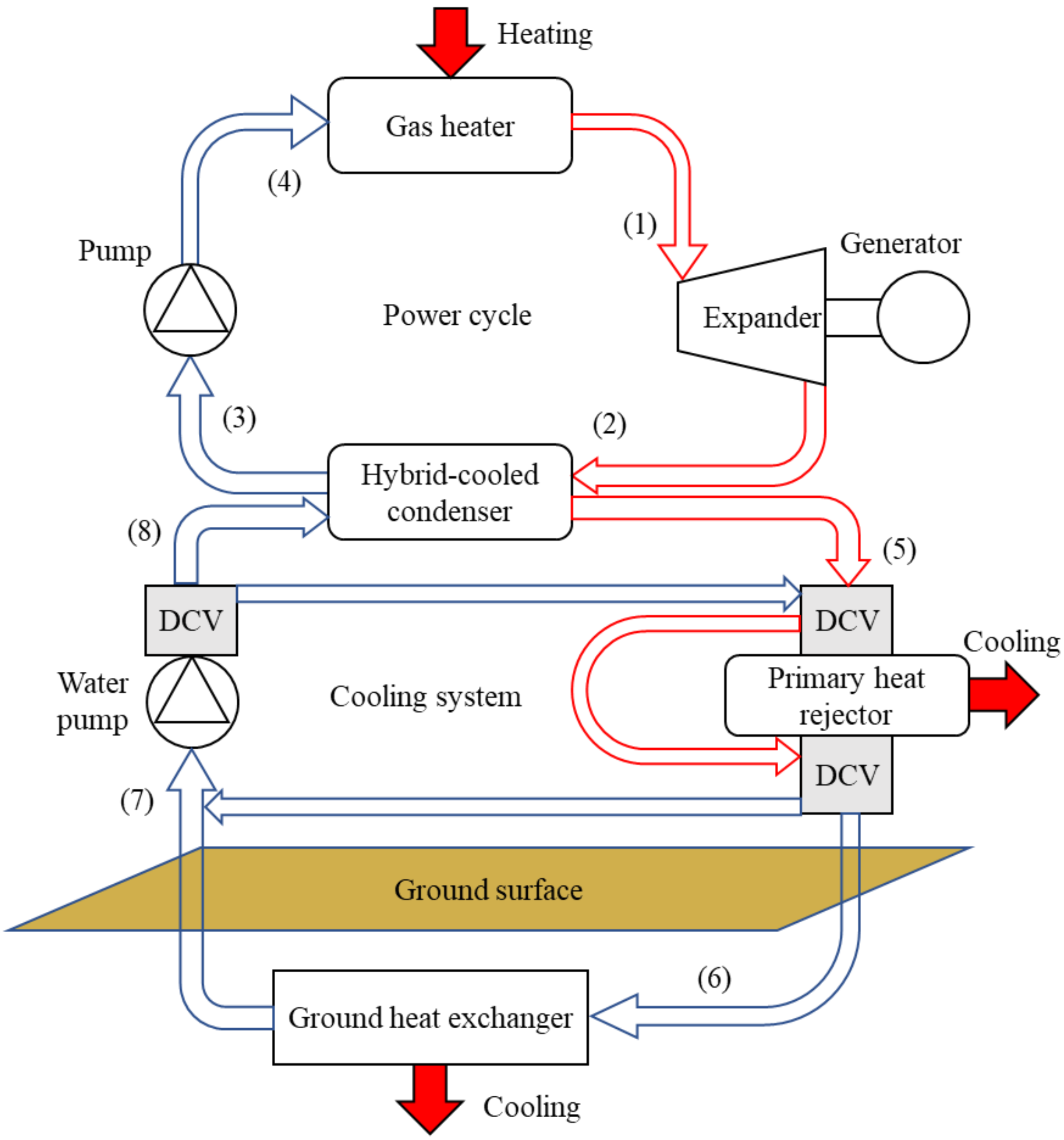
(a)

(b)

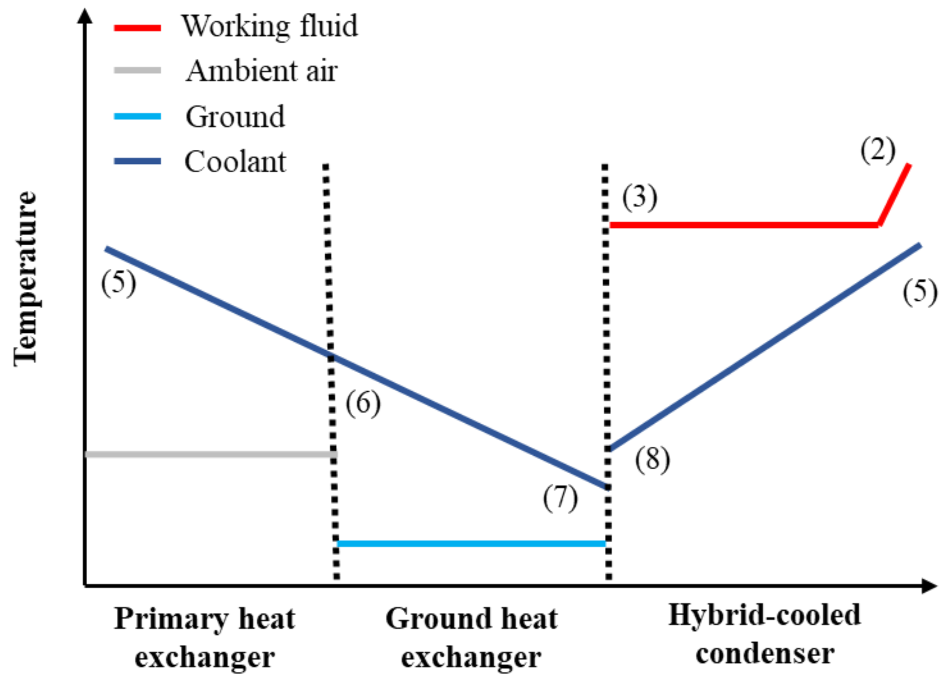
Figure 2: CO_2-based transcritical Rankine cycle combined with a ground-cooled condenser; (a) components and (b) temperature-entropy diagram

The heat rejected from a power cycle can be relatively massive with respect to the shallow ground heat exchanger cooling capacity, otherwise a large installation would be required. This causes additional expenses making the system inappropriate for such utilizations. For this reason, it is better to integrate another cooling source into the proposed system such as an air-cooled or water-cooled heat exchanger to operate as a primary heat rejector (see Figure 3). Another advantage of this integration is to provide coolth compensation to the ground heat exchanger in which this can immensely enhance its performance during operating hours. GE-based hybrid systems have been frequently investigated by researchers since they offer various advantages compared to GE individual systems [36]. They are mainly used to reduce the capital cost of installation and to avoid ground thermal imbalance (heat accumulation or thermal depletion). However, most of these hybrid systems are used for heating or power generation by combining GE and solar energy subsystems [37, 38].

The proposed cooling system has four operating modes that are controlled by the directional control valves (DCVs). Three of these modes are used when the power cycle is activated such that one or both heat exchangers can be operating depending on the ambient and ground temperatures. In most cases, the coolant (water) should pass through the primary heat exchanger before entering the GHE unless the ambient air temperature is higher than that of water at state 5. However, if the ground temperature is higher than that of ambient air, then the GHE must be overtaken. The fourth mode could be used only when the power cycle is turned off and the ambient air temperature is lower than that of the ground which mainly occurs during night hours. This is a coolth recovery mode used to extract heat from the ground such that water flows through both heat exchangers without entering the hybrid-cooled condenser.


Heating
Gas heater
(4)
(1)
Generator
Pump
Power cycle
Expander
(3)
(2)
Hybrid-cooled
condenser
(8)
(5)
DCV
DCV
Cooling
Water
pump
Cooling system
Primary heat
rejector
DCV
(7)
Ground surface
(6)
Ground heat exchanger
Cooling

(a)



(b)

Figure 3: (a) Schematic diagram of the optimized system and (b) temperature variation in the cooling system; directional control valve (DCV)

The model used in Ref. [34] for calculating the CO_2-based transcritical Rankine cycle (CTRC) performance is also considered in the current study that was developed in Engineering Equation Solver (EES). The thermodynamic performance of the cycle depends mainly on the gas and power cycle's conditions (mass flow rate, temperatures, and pressures). The net output power can be calculated by:

$$\dot{W}_{net} = \dot{W}_t . \eta_{ge} - \dot{W}_p - \dot{W}_{wp} \tag{1}$$

where $\dot{W}_t$ is the turbine power, η_{ge} is the generator efficiency (90%), $\dot{W}_p$ is the pump power, and $\dot{W}_{wp}$ is the power demand of the water pump. The current study focuses on small scale power cycles which will be noticed from the resulting low net output power. This makes it necessary to replace the turbo expander with a scroll expander which better fits such applications [39]. This is crucial to avoid inefficient expansions in small turbines that operate at very high rotational speeds. In such cases (below 10 kW), the scroll expander efficiency is almost always considered as ~70%.

$\dot{W}_t$ and $\dot{W}_p$ mainly depend on the mass flow rate of the working fluid and the enthalpy variation as shown in equations (2) and (3). The isentropic efficiency of the pump is considered as 80% [34].

$$\dot{W}_t = \dot{m}_f(h_1 - h_2) \tag{2}$$

$$\dot{W}_p = \dot{m}_f(h_4 - h_3) \tag{3}$$

Equations (4) and (5) show the methods used for evaluating the cycle's efficiency and heat added to the cycle, respectively:

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \tag{4}$$

$$\dot{Q}_{in} = \dot{m}_g c_{pg}\left(T_{gi} - T_{go}\right) = \dot{m}_f(h_1 - h_4) \tag{5}$$

where $\dot{Q}_{in}$ is the heat extracted from the hot gas and added to the cycle's gas heater. The value of heat added is based on the specific heat (c_{pg}), flow rate ($\dot{m}_g$) and temperatures of hot gas (T_{gi} and T_{go}).

The methodology followed to evaluate the states of the proposed system is as follows:

- Considering a minimum temperature difference between the cooling source and the working fluid's condensation temperature.
- The states of coolant passing through the cooling system (5, 6, 7 and 8) can be calculated based on the ambient air and ground temperatures depending on the operating mode as mentioned previously in this section.
- Evaluating the low pressure of the power cycle and the enthalpy of the working fluid exiting the condenser (state 3).

- Considering the highest pressure of the cycle, state 4 can be calculated based on the isentropic efficiency of the pump (80%).
- At a given expander inlet pressure (state 1), the temperature will be chosen depending on the resulting net output power.
- Calculating the mass flow rate of the working fluid based on the energy balance in the gas heater as shown in equation (5).

3. Working fluids

The CO_2-based transcritical Rankine cycle presented in Figure 2a was used only as a reference system to investigate the potential of ground-cooled condenser. This section describes the methodology followed to select the most suitable working fluid for the optimized system displayed in Figure 3. Low to medium-grade sources of energy are considered in this study. Such sources require the adoption of ORC instead of the conventional steam Rankine cycle and this is mainly due to the low flow rate of water resulting in inefficient expansions and requirement for small turbines operating at high rotational speeds [40]. Even though, steam power cycle can theoretically generate more power in some cases, but it is not applicable to use it when the net output power is very low as will be presented in this research. Thus, it would not be economically feasible to adopt a Rankine cycle under the investigated conditions.

3.1 Characteristics of suitable working fluids

Firstly, to select the best working fluid, it is necessary to carry out parametric studies to compare the different usable working fluids. Choosing the fluid corresponding to the highest net output power and efficiency is not always the optimal method since it is essential to be aware of critical drawbacks and negative effects. These may include toxicity, flammability, and global warming potential. There are some other issues that are related to the fluid's nature like requiring high pressures to operate optimally or low flow rate. 1,1,2-Trichloro-1,2,2-trifluoroethane (R113) presents a good example of organic fluids that can produce high amount of power, while it is unfavorable due to its negative effect on public health and ozone layer. For these reasons and according to previous investigations, three compatible refrigerants will be compared in terms of performance that are 2,2-Dichloro-1,1,1-trifluoroethane (R123), 1-Chloro-1,2,2,2-tetrafluoroethane (R124) and 1,1,1,3,3-Pentafluoropropane (R245fa). However, CO_2 is also included in this comparison only to represent the results related to the reference system presented in Ref. [34]. After stating the negative effects of unsuitable working fluids, it is also important to mention the factors that encourage using the convenient fluids. The most important factors are mainly the availability, cost, and performance. These are the main reasons for choosing the mentioned refrigerants in this parametric study. The methodology used to choose the best working fluid can be represented by the following points:

- Proposing different working fluids based on their availability and cost.
- Studying the negative effects of the proposed working fluids.
- Eliminating the unacceptable fluids based on their environmental impacts.
- Investigating the best point temperature of each working fluid.
- Investigating the relation between best point temperature and cycle’s conditions.
- Comparing the working fluid’s performance regarding net output power.
- Examining the flow rate of the best working fluid to check if it is feasible for an expander. At this stage, using water as a working fluid (Rankine cycle) would be eliminated.

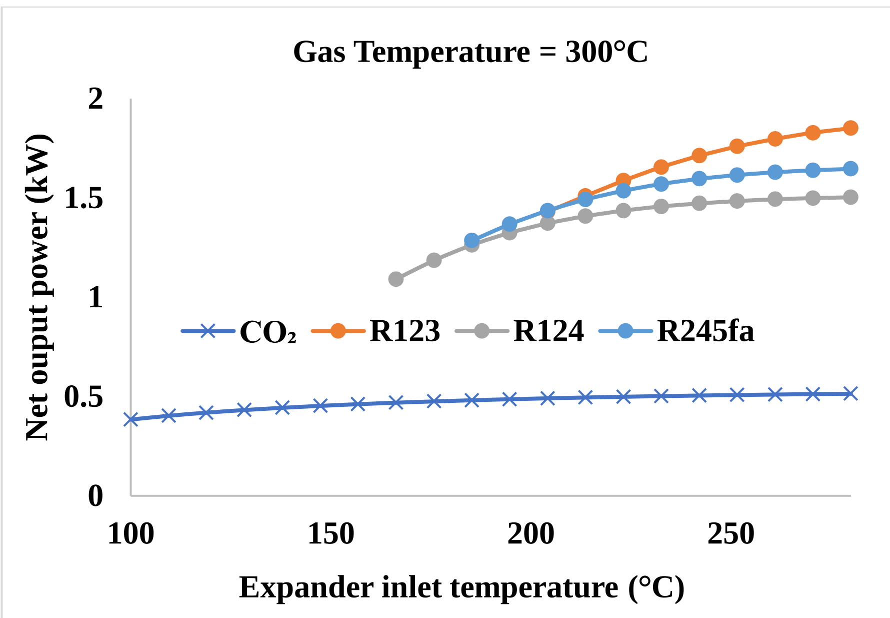
Table 1 presents the IUPAC names of the investigated working fluids in addition to their critical temperatures and pressures.

Table 1: The critical temperatures and pressures of the working fluids investigated

Working fluid	**IUPAC name**	**Critical temperature (°C)**	**Critical pressure (kPa)**
CO_2	Carbon dioxide	30.98	7377
R123	2,2-Dichloro-1,1,1-trifluoroethane	183.7	3668
R124	1-Chloro-1,2,2,2-tetrafluoroethane	122.3	3624
R245fa	1,1,1,3,3-Pentafluoropropane	154	3651

3.2 Effect of energy grade source

Each working fluid has optimal operating conditions in which these depend mainly on the available grade source of energy (flow rate and temperature). In this section, the mass flow rate of gas, expander inlet pressure and condensation temperature are kept constant at 200 kg/hr, 10 MPa and 25°C, respectively. Figure 4 shows the variation of net output power resulting from the compared working fluids as function of expander inlet temperature at gas temperatures of 300°C and 600°C. The gap presented in Figure 4a corresponds to the unsuitability of refrigerants at these conditions such that the lowest temperature of operation depends directly on the expander inlet pressure. Thus, it would be better to decrease the pressure to ensure the superheating of working fluid at low temperatures. It is essential to ensure that the fluid exiting the expander is in vapor state otherwise it will not be possible to operate properly, and the expander blades will be damaged with time. This is also better considering lower pressures inside the pipes and hence minimizing the required thickness and preventing leakage. This is the first reason that makes the refrigerants preferable with respect to CO_2. Another advantage is the high extracted power compared to CO_2 in which all investigated refrigerants show better performances with noticeable differences (see Figure 4). Among the studied working fluids, R123 presented the highest net output power at the different gas and expander inlet temperatures. In the next sections, CO_2 will be removed from the comparative study due the observed significant difference compared to other working fluids since this research focuses more on the thermodynamic and economical aspects. However, CO_2 would be an alternative solution if there are high restrictions since it is more environmentally benign. Thus, it is also important to develop the CO_2-based transcritical Rankine cycle incorporating a ground-cooled condenser investigated in Ref. [34].



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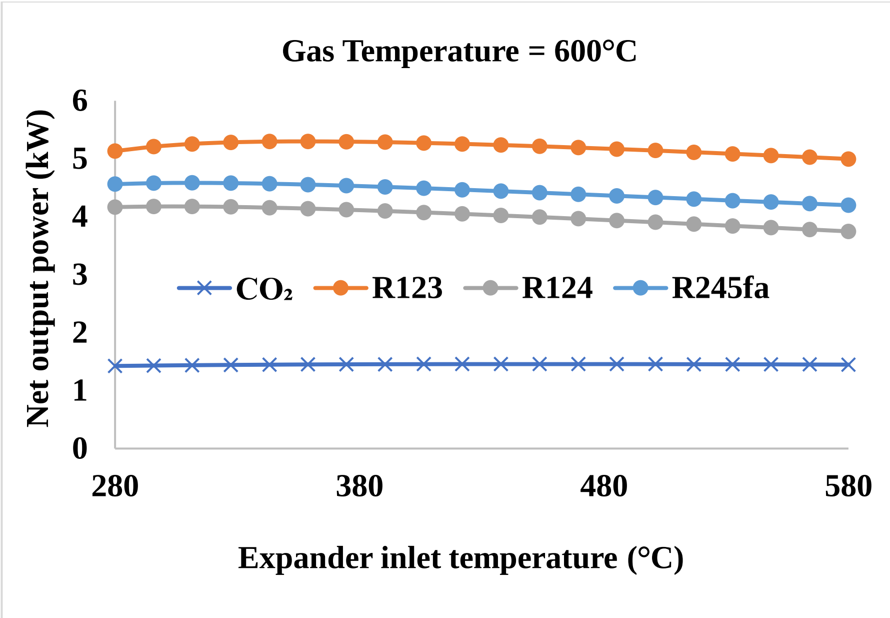
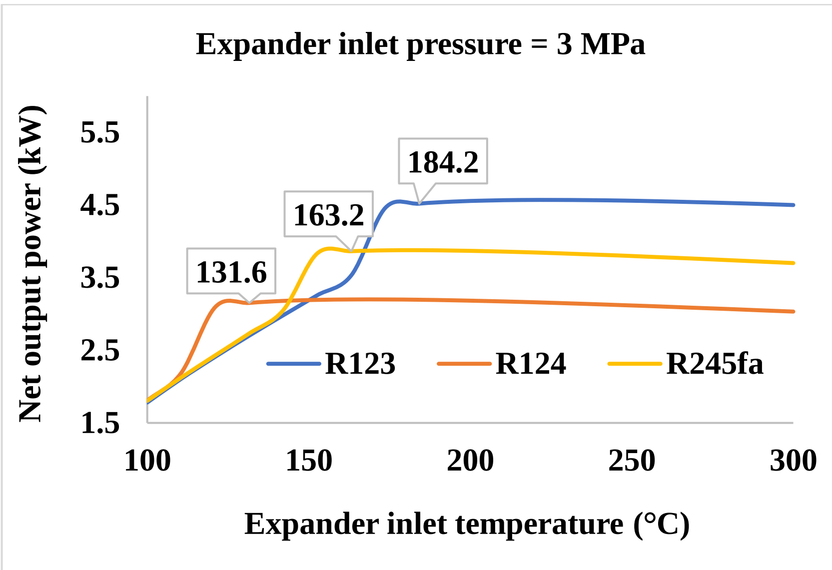


(b)

Figure 4: Effect of expander inlet temperature on the performance of working fluids at gas temperature of (a) 300°C and (b) 600°C

3.3 Optimal operating temperature

After eliminating CO_2 from the possible working fluid choices, it is better to decrease the cycle's highest pressure. This is mainly because the chosen refrigerants can operate properly at low pressures and especially at low grade sources of energy. The gas flow rate, gas temperature and condensation temperature are maintained at 200 kg/hr, 600°C and 25°C respectively in this section. The optimal operating temperatures of the refrigerants are presented in Figure 5 in which it can be noticed that the best point temperature can vary with the change in the expander inlet pressure. This temperature is more noticeable at 3 MPa such that it is recorded as 131.6°C, 163.2°C and 184.2°C for R124, R245fa and R123, respectively. This shows that the refrigerant that has higher critical temperature requires higher operating temperature to achieve its optimal performance (see Table 1). Figure 5 shows that when the expander inlet pressure increases from 3 to 6 MPa, the curves become smoother which means that the variation in expander inlet temperature is more effective at low pressures. This makes the choice of operating temperature more critical at low grade sources of energy because it will not be possible to increase the expander inlet pressure. According to this comparison, R123 still corresponds to the highest performance at low expander inlet pressures achieving net output power of 4.57 kW at 3 MPa and 5.046 kW at 6 MPa. This is another reason for making R123 suitable for low grade applications since there is no significant difference between the net output power at the different studied pressures.



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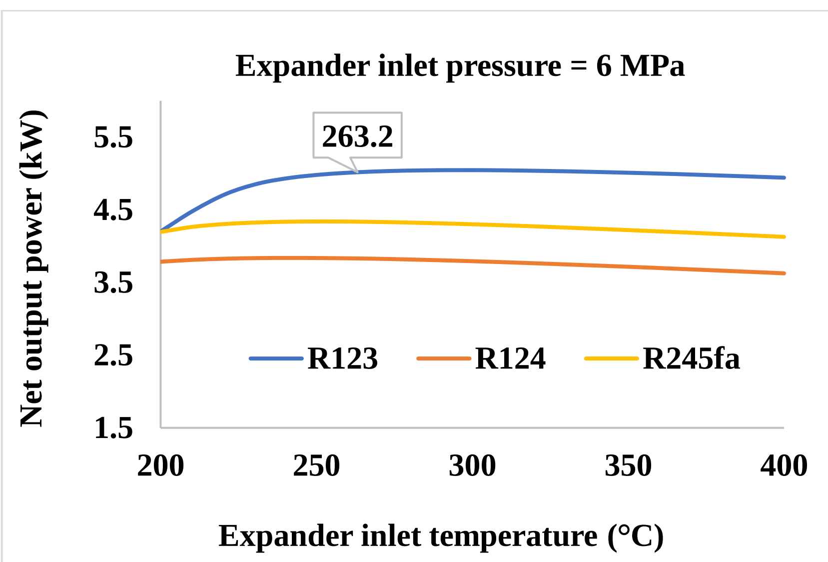
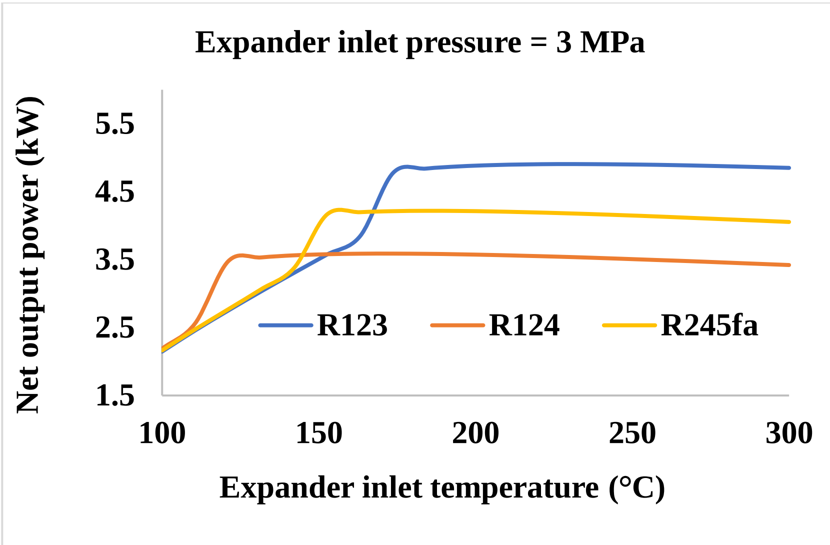


(b)

Figure 5: The best point temperature of investigated working fluids at a condensation temperature of 25°C and expander inlet pressure of (a) 3 MPa and (b) 6 MPa

3.4 Effect of condensation temperature

The aim of investigating the ground-cooled condenser presented in Ref. [34] was to decrease the condensation temperature to increase the difference between the low and high cycle's pressures. Consequently, this has helped improving the system's efficiency and net output power due to the increase in working fluid's expansion in the expander. To study the effect of using ground-cooled condenser (decreasing the condensation temperature), the values considered in the previous section are kept the same whilst only changing the condensation temperature from 25°C to 15°C. The results are presented in Figure 6 in which R123 generates the highest net output power with highest values of 5.367 kW at 3 MPa and 5.615 kW at 6 MPa. In comparison with the results presented in Figure 5, the corresponding enhancements are 7.35% and 6.36%. This shows that the importance of using ground-cooled condenser is more noticeable at low grade sources of energy.



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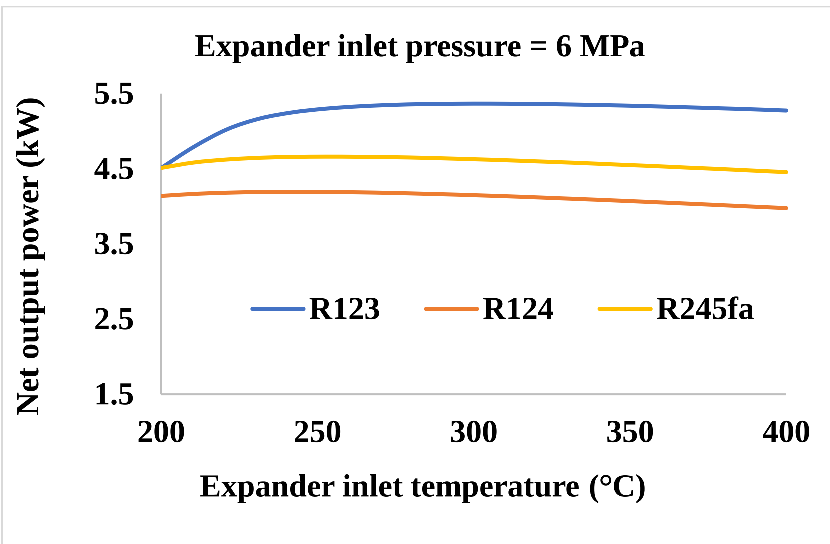


(b)

Figure 6: The variation of net output power as a function of expander inlet temperature at a condensation temperature of 15°C and expander inlet pressure of (a) 3 MPa and (b) 6 MPa

Table 2 shows the performance enhancements of all working fluids in terms of net output power. The highest enhancements are recorded by the refrigerant that has the lowest critical temperature which R124. However, the refrigerants with high critical temperatures that are R123 and R245fa still generate higher amounts of power at the investigated conditions. The highest enhancement recorded by R124 is also obtained at the low pressure (3 MPa) with a value of 12.13% representing the increase in net output power from 3.2 kW to 3.588 kW. These enhancements are still less than that reported in Ref. [34] for the CTRC which was approximately 30%. However, all these values present the enhancement in performance of each working fluid alone compared to its operation without using a ground-cooled condenser. Thus, it is still favorable to use the mentioned refrigerants since they can significantly produce more power than that of CTRC under the studied conditions as shown in Figure 4.

Table 2: The enhancement of net output power when using ground-cooled condenser

Working fluid	**Net output power enhancement (%)**	
	Expander inlet pressure: 3 MPa	Expander inlet pressure: 6 MPa
R123	7.35	6.36
R124	12.13	9.28
R245fa	8.77	7.37

4. Regenerative cycle

Adding a regenerator to the optimized system presented in Figure 3 would be an attractive method to enhance the thermodynamic performance of the cycle and specially when there is a huge difference between the temperatures of states (2) and (4). The regenerative cycle is depicted in Figure 7 which aims to take advantage of the energy remaining in the working fluid exiting the expander. On the other hand, it is necessary to check out if there are any negative consequences associated with this integration such as the amount of heat rejected to the ground. In this section, the inlet and outlet heat source temperatures are maintained at 500°C and 120°C, respectively.

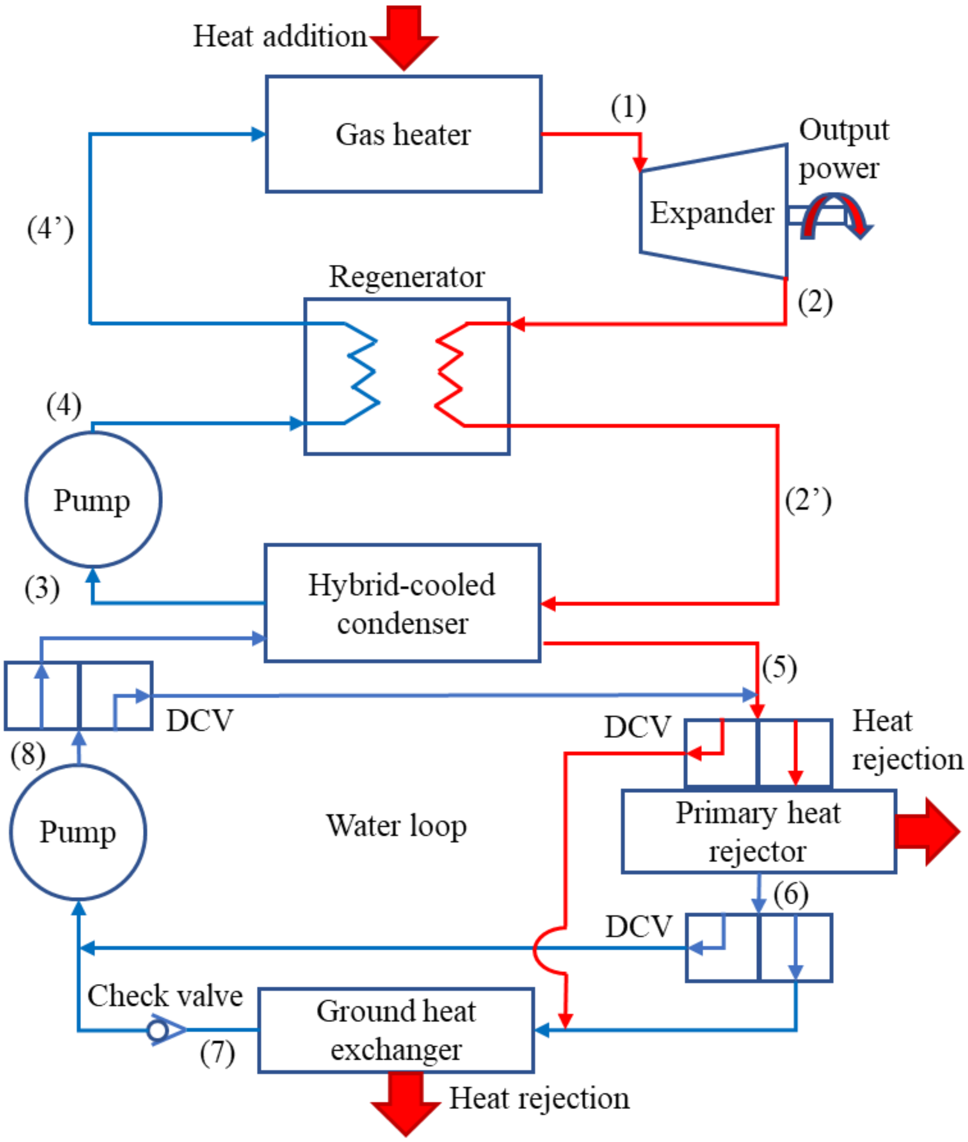


Figure 7: Schematic diagram of the regenerative cycle

Figure 8 presents the temperature-entropy diagram of the regenerative cycle showing the states of the organic Rankine cycle. It is essential to compare the temperatures entering and exiting the regenerator in all cases to ensure that the regeneration is applicable. This means that $T_{4'}$ must be always greater than T_4 as well as $T_{2'}$ must be greater than T_4 and T_3. For this reason, in the upcoming sections, some points have been eliminated from the figures representing the results. This mainly depends on the thermodynamic properties of the investigated working fluids. The temperature difference between the working fluid entering the gas heater and that exiting the expander is assumed to be 30°C to avoid the installation of large heat exchanger (regenerator).

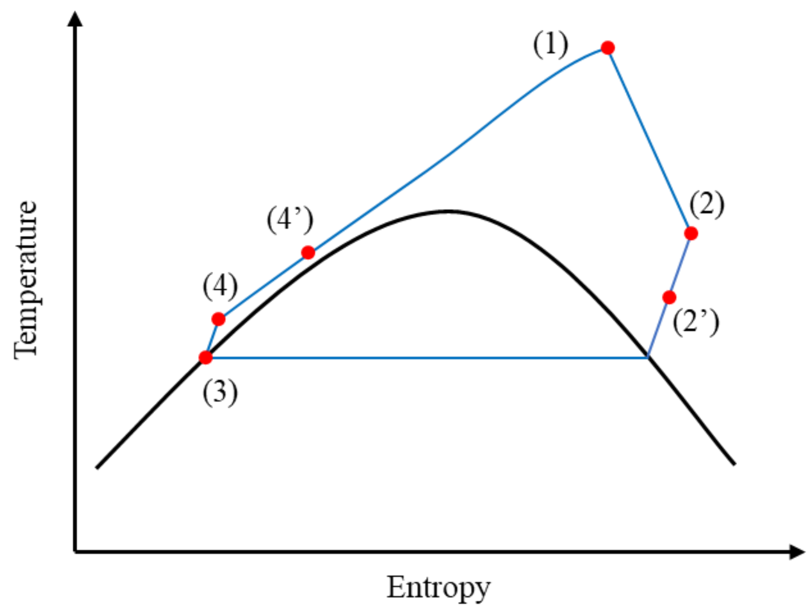


Figure 8: Temperature-entropy diagram of the regenerative transcritical Rankine cycle

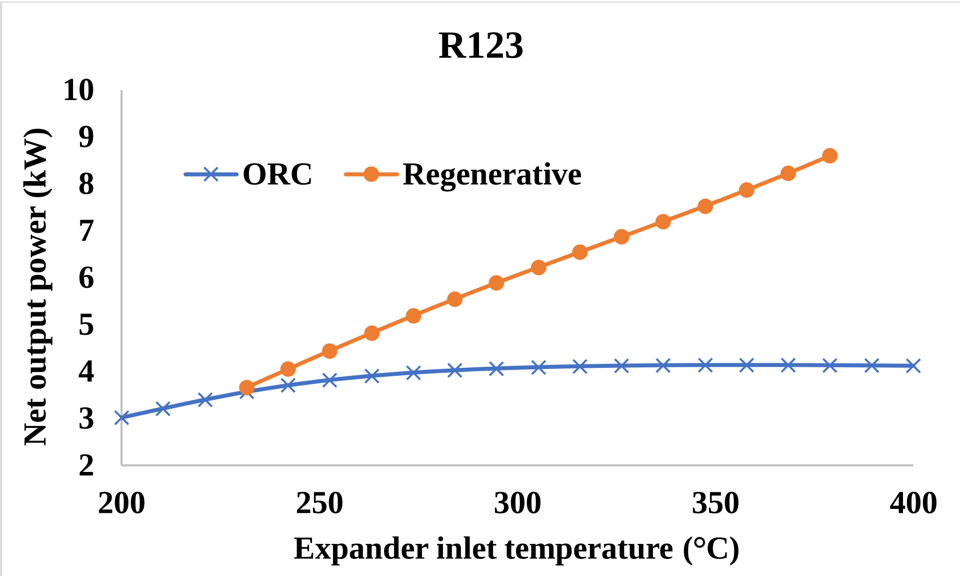
Equation (6) presents the relation between the enthalpies of working fluid entering and exiting the regenerator. The effectiveness of regenerator (ε_r) is assumed to be 80%.

$$\varepsilon_r = \frac{h_2 - h_{2'}}{h_{4'} - h_4} \tag{6}$$

where h_2, $h_{2'}$, h_4 and $h_{4'}$ are the working fluid's enthalpies exiting the expander, entering the condenser, exiting the pump, and entering the gas heater, respectively. This equation aims to calculate the enthalpy at state (2') and hence to evaluate $T_{2'}$ considering a negligible pressure drop inside the regenerator.

4.1 Expander inlet temperature

The temperature of working fluid entering the expander (T_1) has a significant effect on the net output power of the regenerative cycle as shown in Figure 9. The expander inlet pressure, condensation temperature and gas flow rate are maintained at 10 MPa, 25°C and 200 kg/hr, respectively. The net output power of the basic ORC is almost around 3.5 kW for all working fluids, however, that of the regenerative cycle has a sharp increase while increasing the expander inlet temperature from 200°C to 400°C. The effect of adding a regenerator is more considerable for R124 than R245fa and R123 at each specific temperature which can be noticed from the difference between the slopes. On the other hand, the unsuitability of regeneration occurs earlier for R124 and R245fa allowing R123 to reach the highest net output power of 8.86 kW at 386°C. The corresponding values for R124 and R245fa are 7.27 kW and 8.07 kW at 312°C and 344°C, respectively.



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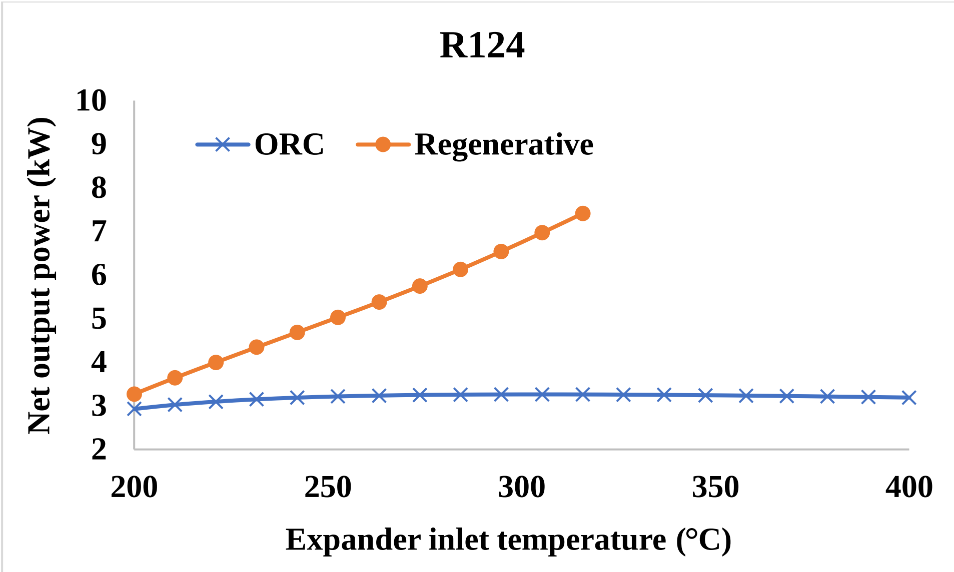

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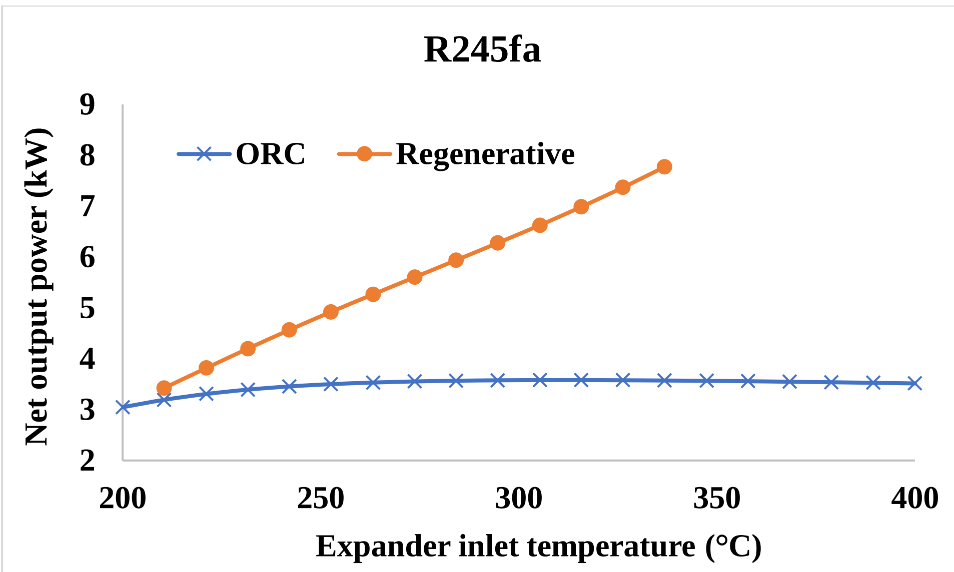


(c)

Figure 9: Effect of expander inlet temperature on the net output power of basic and regenerative organic Rankine cycles for (a) R123, (b) R124 and (c) R245fa

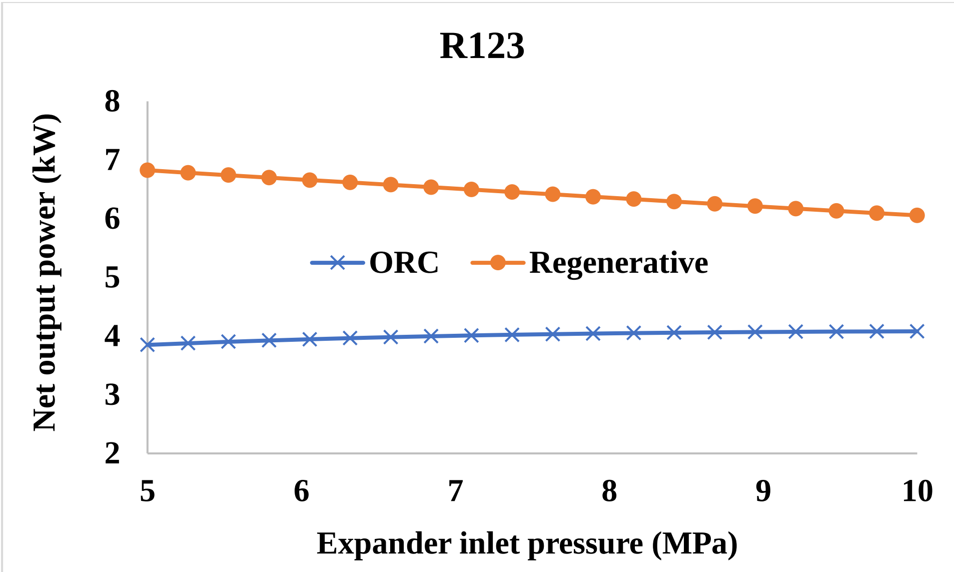
The highest net output power of the basic ORC and regenerative cycles for the three working fluids is recorded and tabulated (see Table 3). The corresponding expander inlet temperatures vary between ORC and regenerative cycle as well as between the working fluids. For the regenerative cycle, the highest power is always achieved at the maximum allowable temperature.

Table 3: The enhancement in net output power using a regenerative organic Rankine cycle

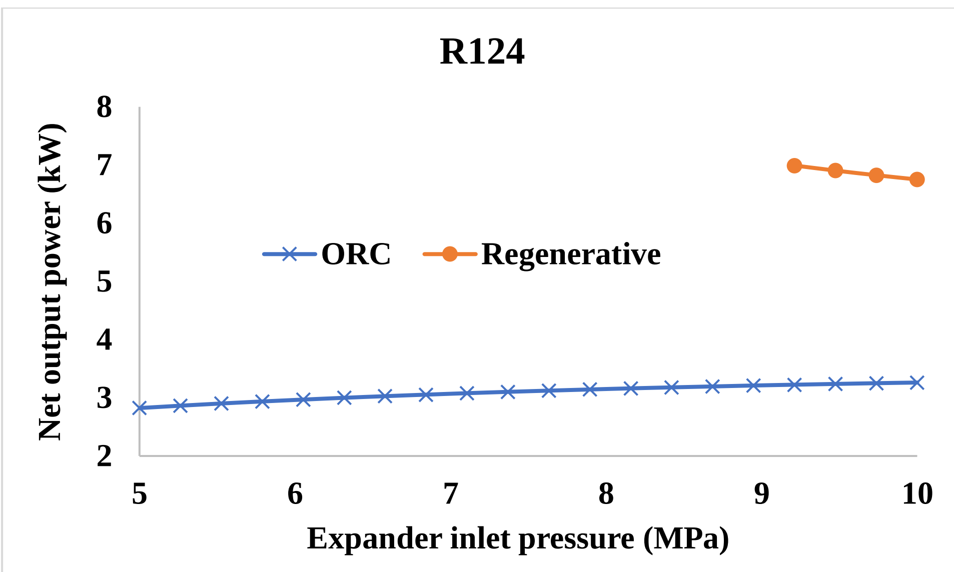
Working fluid	Net output power (kW)	
	Organic Rankine cycle	**Regenerative cycle**
R123	4.14 @ 358°C	8.86 @ 386°C
R124	3.26 @ 305°C	7.27 @ 312°C
R245fa	3.58 @ 310°C	8.07 @ 344°C

4.2 Expander inlet pressure

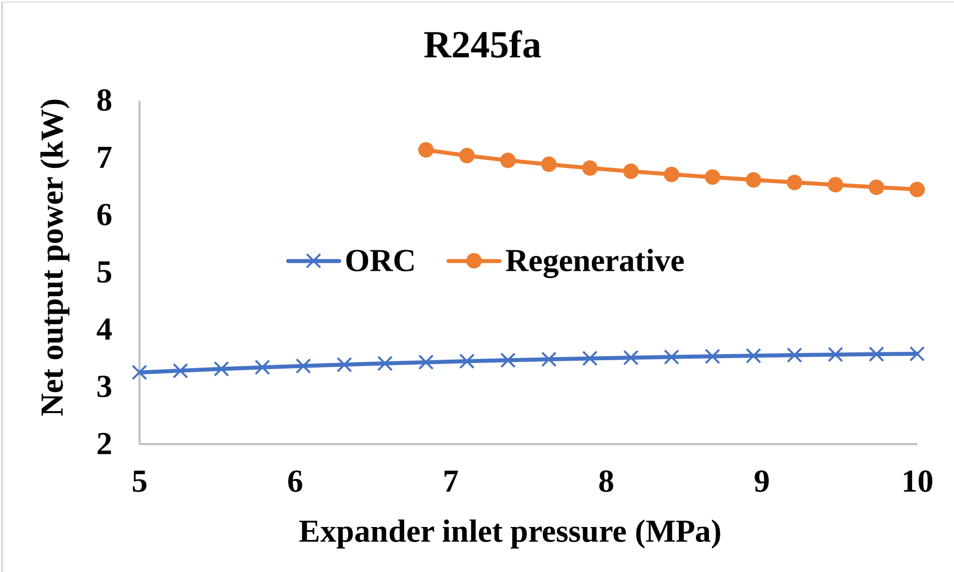
To investigate the impact of expander inlet pressure on the cycle's performance, the expander inlet temperature, condensation temperature and gas flow rate are kept constant at 300°C, 25°C and 200 kg/hr, respectively. It can be noticed from Figure 10 that as the expander inlet pressure increases the net output power of basic ORC increases while that of regenerative cycle decreases for all working fluids. The highest performances are recoded as 6.83 kW for R123, 7.2 kW for R124 and 7.26 kW for R245fa. Even though, R245fa corresponds to the highest net output power, but R123 is still the preferable working fluid. This is due to that R123 can operate at wide range of temperatures and pressures as shown in Figure 9a and Figure 10a. Another reason is that the results presented in Figure 9 is obtained at an expander inlet temperature of 300°C which is near to the best point temperature of R124 and R245fa (see Table 3). Thus, it would be possible to enhance the net output power of R123 by increasing the temperature (T_1). Additionally, R123 can operate at lower pressures than that of R124 and R245fa in case of regenerative cycle at the same expander inlet temperature. This makes R123 the favorable working fluid and specially because the net output power is still highly acceptable compared to R124 and R245fa when operating far from its optimal temperature at lower pressures. For R123, it can operate normally between 5 MPa and 10 MPa at an expander inlet temperature of 300°C. However, for R124 and R245fa, the regenerative cycle is unsuitable for pressures lower than 9.1 MPa and 6.8 MPa, respectively.



(a)



(b)

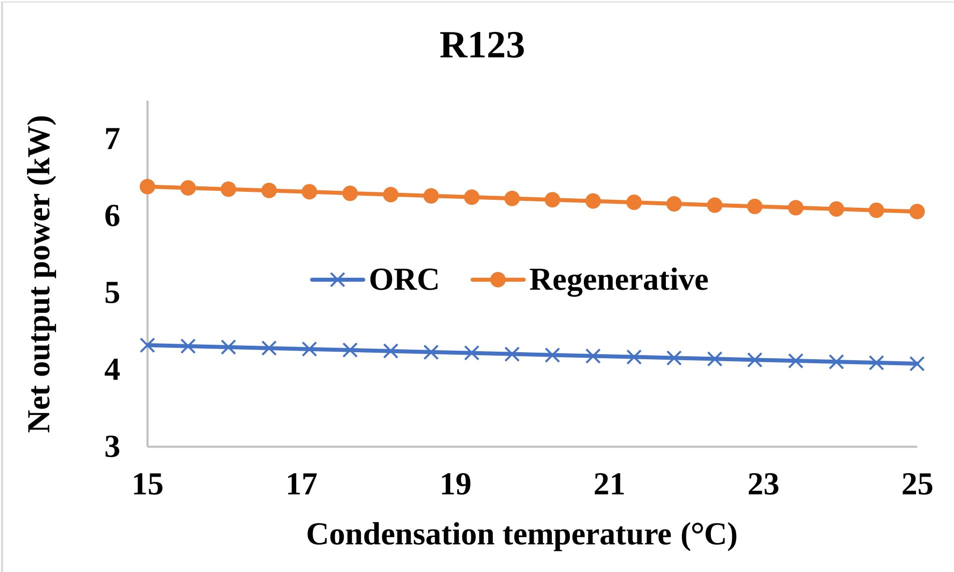


(c)

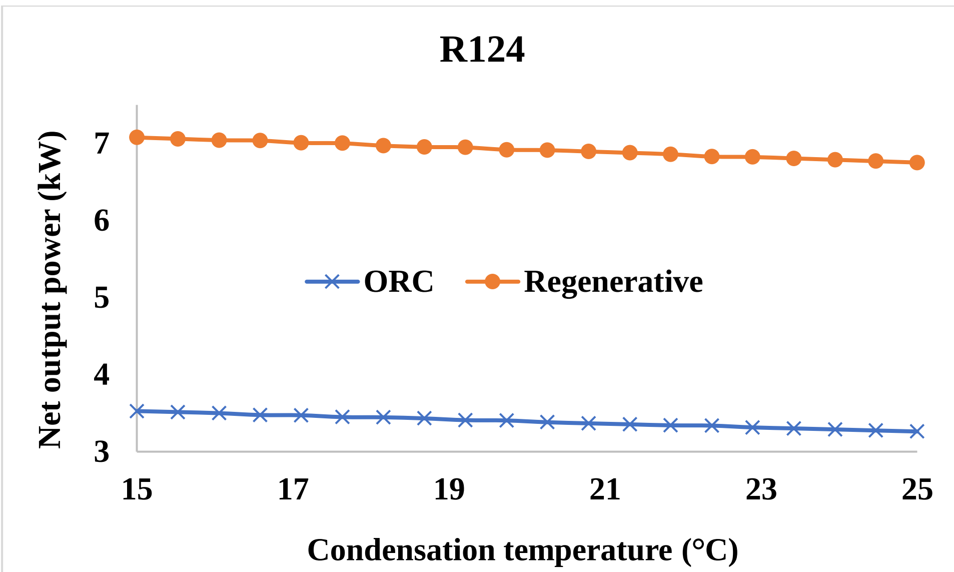
Figure 10: Effect of expander inlet pressure on the net output power of basic and regenerative organic Rankine cycles for (a) R123, (b) R124 and (c) R245fa

4.3 Condensation temperature

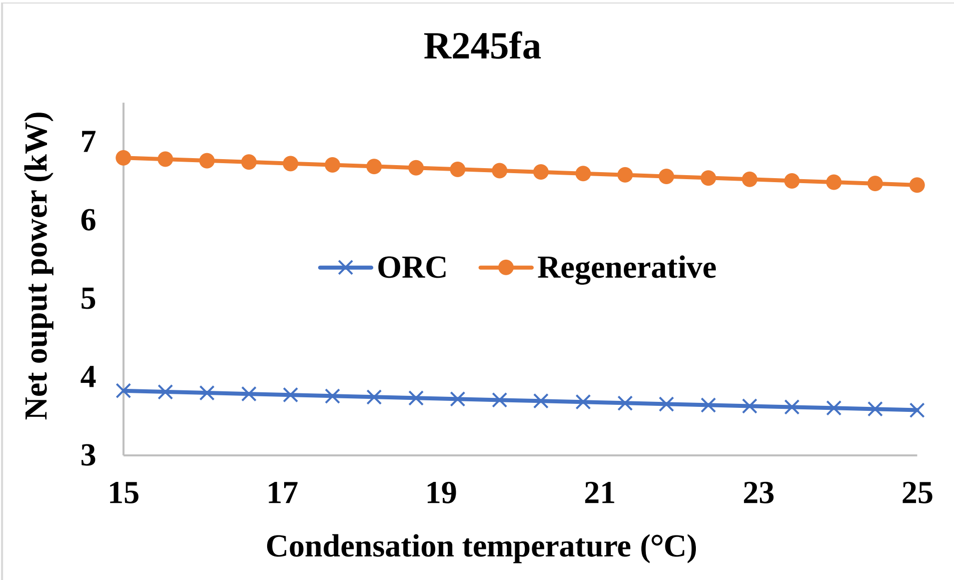
The variation of condensation temperature (T_3) affects the basic ORC and regenerative cycle almost in the same manner for all working fluids. Figure 11 shows that as the condensation temperature increases from 15°C to 25°C the net output power of the cycles decreases by an amount of ~0.3 kW. The decrease in condensation temperature corresponds to the use of ground cooling system since the ground is responsible for heat removal. Thus, if this system is activated, and considering a condensation temperature drop from 25°C to 15°C, the enhancement of net output power for regenerative R123, R124 and R245fa can be estimated as 5.4%, 4.8% and 5.4%, respectively. This shows that the ground cooling system is more effective for R123 and R245fa than that of R124 regenerative ORCs.


R123
Net output power (kW)
7
6
5
4
3
ORC
Regenerative
15
17
19
21
23
25
Condensation temperature (°C)

(a)


R124
Net output power (kW)
7
6
5
4
3
ORC
Regenerative
15
17
19
21
23
25
Condensation temperature (°C)

(b)



(c)

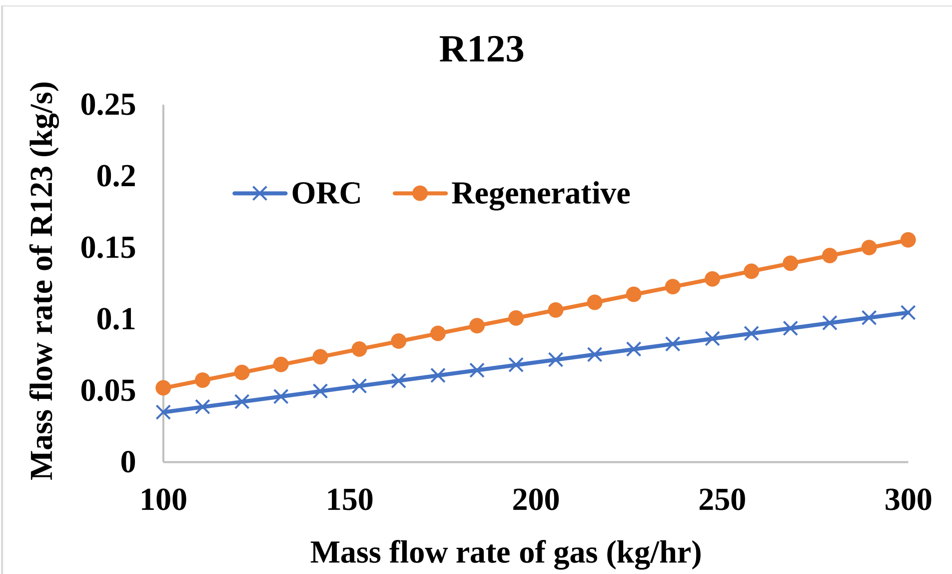
Figure 11: Effect of condensation temperature on the net output power of basic and regenerative organic Rankine cycles for (a) R123, (b) R124 and (c) R245fa

4.4 Flow rate of heat source

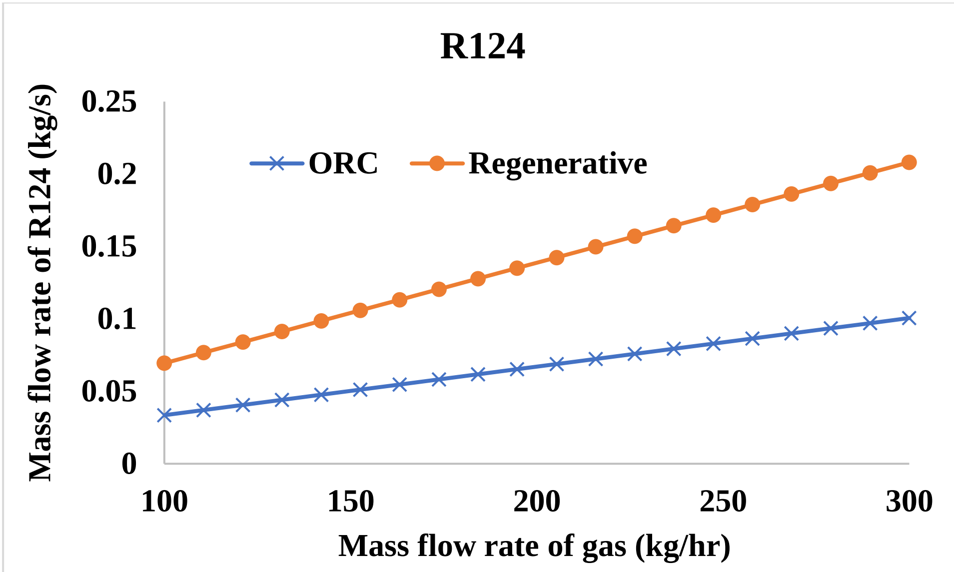
According to the energy balance inside the gas heater, the following equation can be used to represent the relation between heat source gas and ORC's working fluid in case of a regenerative cycle:

$$\dot{m}_g . c_{pg}\left(T_{gi} - T_{go}\right) = \dot{m}_f\left(h_1 - h_{4'}\right) \tag{7}$$

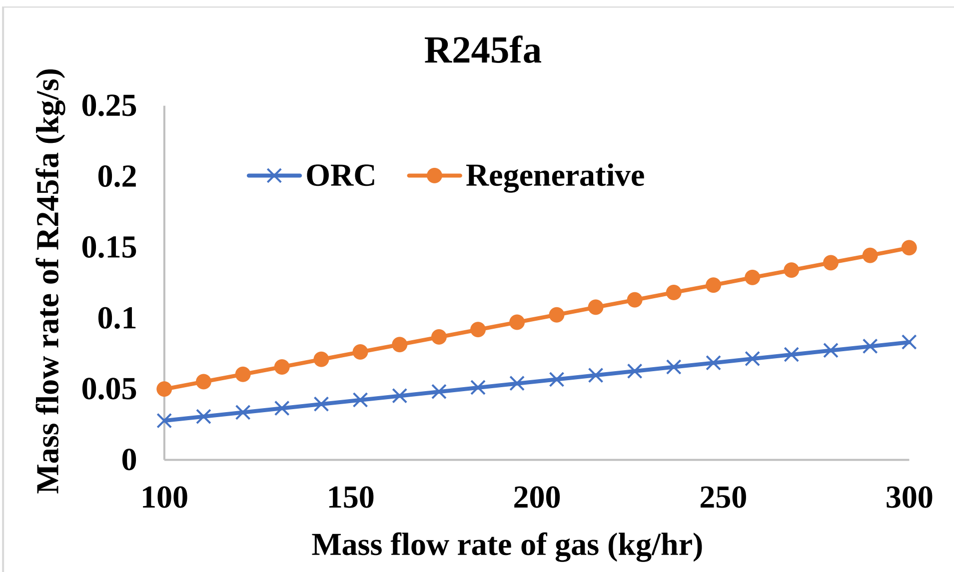
where $\dot{m}_g$, c_{pg}, T_{gi}, T_{go}, $\dot{m}_f$, h_1 and $h_{4'}$ are the gas flow rate, gas specific heat, gas inlet temperature, gas outlet temperature, working fluid's mass flow rate, enthalpy at state (1) and enthalpy at state (4'), respectively. It is obvious that the flow rates of heat source and working fluid are directly proportional. This relation can be observed in Figure 12 such that both flow rates vary similarly in ORC and regenerative cycle for all working fluids. However, the slope of curve corresponding to the regenerative cycle is greater than that of basic ORC showing that the regeneration approach is more attractive at high heat source flow rates. The mass flow rate of working fluid in the regenerative cycle is always higher than that of ORC revealing the results presented in Figure 9, Figure 10 and Figure 11. Therefore, the increase in net output power is mainly due to the increase in the working fluid's mass flow rate that is affected by the enthalpy change at the gas heater's inlet. Based on equation (7), if the enthalpy of the working fluid entering the gas heater increases, the mass flow rate will also increase. So, by comparing the basic ORC and regenerative cycle, the enthalpy has increased from state (4) to (4') resulting in an increase in the working fluid's mass flow rate.



(a)



(b)



(c)

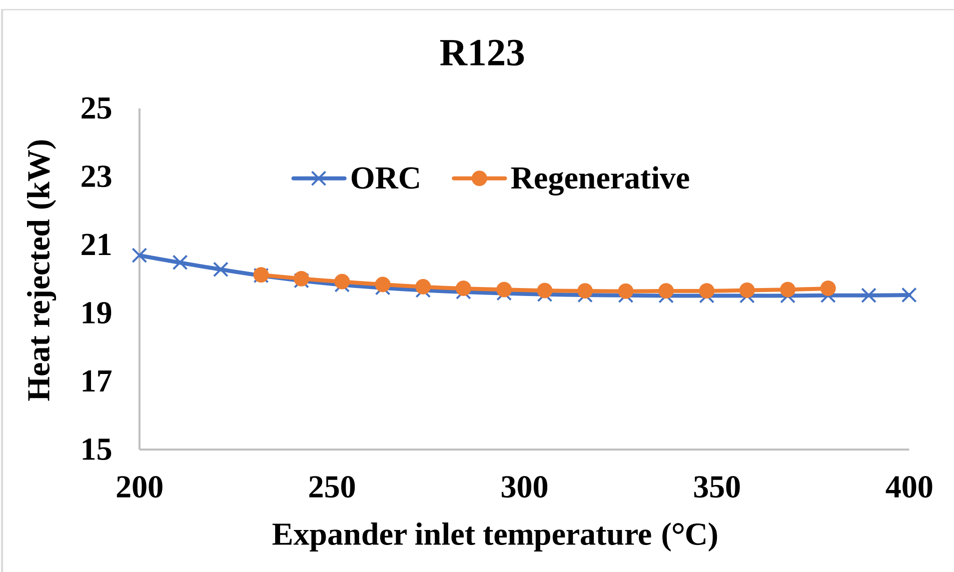
Figure 12: The relation between mass flow rate of heat source and working fluids in basic and regenerative organic Rankine cycles for (a) R123, (b) R124 and (c) R245fa

4.5 Heat rejection

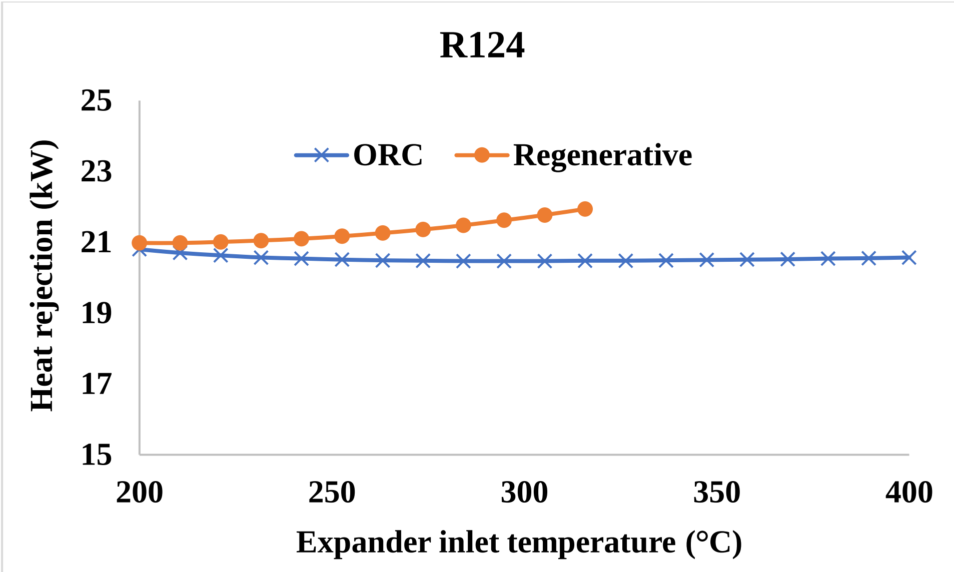
From the previous sections, the increase in working fluid's mass flow rate has been considered as an advantage to enhance the overall cycle's thermodynamic performance. On the other hand, this may cause negative impact on the ground loop resulting in a larger ground heat exchanger which mainly depends on the amount of heat rejected from the cycle. Equation (8) is used to estimate the heat transferred from the regenerative ORC to the ground.

$$\dot{Q}_{rej} = \dot{m}_f (h_{2'} - h_3) \tag{8}$$

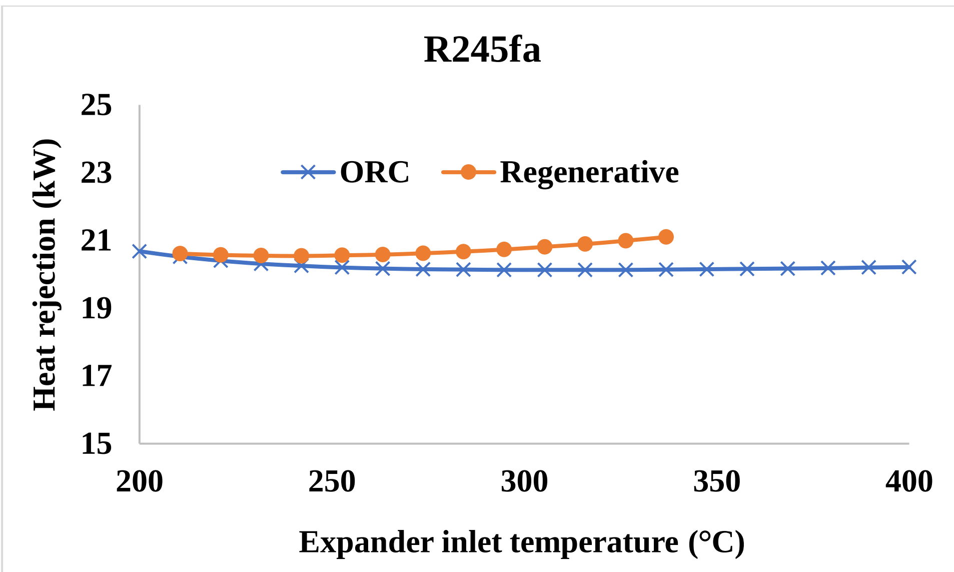
The amount of heat rejected from the cycle is presented in Figure 13 for the three investigated working fluids as a function of the expander inlet temperature. It can be noticed that the effect of adding a regenerator on the amount of heat rejected from the cycle is more considerable for R124 and R245fa than that for R123. This is clearly presented from the gap between the curves corresponding to ORC and regenerative cycle. The heat rejected in case of R124 and R245fa increases slightly from 20.98 to 21.94 kW and 20.66 to 21.18 kW, respectively while that of R123 decreases from 20.16 to 19.72 kW. This makes R123 also more attractive and especially because the highest net output power is achieved at high expander inlet temperatures (see Figure 9) while the heat rejected is less than that of other fluids by approximately 2 kW.



(a)



(b)



(c)

Figure 13: The influence of regenerator on the heat rejected to the ground for (a) R123, (b) R124 and (c) R245fa

4.6 Summary

The performance of working fluids varies significantly under the investigated conditions as presented in the previous sections. The effect of using the ground-based cooling system (decrease in condensation temperature) can be considered as an attractive approach to enhance the efficiency and extract more power from a Rankine cycle and especially in case of low-grade energy sources.

Table 4 summarizes the optimum performance of the three investigated working fluids in terms of net output power, efficiency, and heat rejection as well as the corresponding expander inlet temperature. These results have been recorded at an expander inlet pressure and condensation temperature of 10 MPa and 15°C, respectively.

Table 4: The optimum performance of investigated working fluids

Working fluid	**Net output power (kW)**	**Efficiency (%)**	**Expander inlet temperature (°C)**	**Heat rejection (kW)**
R123	9.1	37.7	385	19.2
R124	7.5	31.0	310	21.1
R245fa	8.3	34.3	342	20.5

5. Conclusions

A comparison between three working fluids was carried out that are R123, R124 and R245fa. Each working fluid has an optimal operating temperature at a given expander inlet pressure. It was shown that as the critical temperature increased the best point temperature also increased. As the pressure changed from 3 MPa to 6 MPa, the optimal expander inlet temperature for R123 was shifted from 184.2°C to 263.2°C. After performing the parametric and comparative studies, it is seen that R123 is the best suitable working fluid under the investigated conditions. This has been recorded for the basic ORC and regenerative cycle. In case of the basic ORC, the effect of decreasing the condensation temperature was more considerable for low expander inlet pressures and working fluids with low critical temperatures. At 3 MPa, as the condensation temperature was decreased from 25°C to 15°C the enhancements in net output power were 7.35%, 12.13% and 8.77% for R123, R124 and R245fa, respectively. In contrast, the effect of using the ground-cooled condenser was more significant at high expander inlet pressures in the regenerative cycle. R123 is not only preferable due to the corresponding thermodynamic performance, but it also resulted in the lowest amount of heat rejected to the ground as well as it can operate at wide range of pressures and temperatures. The latter reason makes the cycle more flexible while using R123 as a working fluid allowing it to operate normally at different heat source conditions. Thus, R123 will be considered as working fluid in future work to investigate a real application to check the feasibility of the system in terms of performance and cost. It is recommended to adopt the regenerative cycle in the case studied for the ground cooling system since it increased the net output power immensely without significantly affecting the amount of heat rejected to the ground. This means that adopting the regenerative cycle will not considerably increase the required ground heat exchanger size.

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