CO_2 – based transcritical Rankine cycle coupled with a ground-cooled condenser

Montaser Mahmoud[1,2], Mohamad Ramadan[3,4*], Sumsun Naher[1], Keith Pullen[1] and Abdul-Ghani Olabi[5,6]

[1]Department of Engineering, School of Mathematics, Computer Science and Engineering, City, University of London, London, UK

[2]Lebanese International University, PO Box 146404 Beirut, Lebanon

[3]International University of Beirut, PO Box 146404 Beirut, Lebanon

[4]Associate member at FCLAB, CNRS, Univ. Bourgogne Franche-Comte, Belfort cedex, France

[5]Sustainable and Renewable Energy Engineering, University of Sharjah, Sharjah, UAE

[6]Mechanical Engineering and Design, Aston University, School of Engineering and Applied Science, Aston Triangle, Birmingham, B4 7ET, UK

Abstract

This paper presents a new system that incorporates the CO_2-based transcritical Rankine cycle with a geothermal condenser. The innovation studied in the current proposed system is the use of shallow ground heat exchanger to activate a ground-cooled condenser instead of air-cooled in a power cycle. This combination is very beneficial for power generation, especially in summer because ground is more effective than the ambient air in extracting heat from the working fluid passing through the condenser due to its lower temperature. Thus, for the same amount of heat added to the cycle, more power could be produced due to the increase in pressure difference between the gas heater and condenser. Consequently, the operating cost of the system will be reduced. This parametric study covers a wide range of heat source applications with gas temperature and mass flow rate being varied between 500-1500°C and 100-350 kg/hr respectively. The objective is to generate the maximum possible power while minimizing the sizes of the heat exchangers. The results show that

https://doi.org/10.1016/j.tsep.2021.100948

the best parameters to be controlled are the turbine inlet temperature and CO_2 pipe diameter. The incorporation of ground-cooled condenser significantly enhanced the cycle's performance in which the net output power could be increased by ~30% compared to the conventional Rankine cycle. This enhancement mainly depends on the ground's temperature since it directly changes the condensation temperature and hence affects the working fluid expansion.

Keywords: Rankine cycle, transcritical Rankine cycle, geothermal condenser, geothermal energy, carbon dioxide, power generation

Nomenclature	
$\dot{Q}$	heat transfer rate (W)
$\dot{W}$	power (W)
$\dot{m}$	mass flow rate (kg/s)
ΔT_{lm}	logarithmic mean temperature difference
A	area (m^2)
D	diameter (m)
h	heat transfer coefficient ($W/m^2.K$)
k	thermal conductivity (W/m.K)
L	length (m)
L_r	characteristic length (m)
n_c	number of coil turns
Nu	Nusselt number
η	efficiency
p	pressure (MPa)
p_c	coil pitch (m)
Pr	Prandtl number
Ra	Rayleigh number
Re	Reynolds number
T	temperature (°C)
U	overall heat transfer coefficient ($W/m^2.K$)
Abbreviations	
CSRC	CO_2-based supercritical Rankine cycle
CTRC	CO_2-based transcritical Rankine cycle
EES	Engineering Equation Solver
GCC	grand composite curve
GHE	ground heat exchanger
ORC	organic Rankine cycle

https://doi.org/10.1016/j.tsep.2021.100948

RES	renewable energy source
SP	size parameter
Subscripts	
c	coil
g	generator
p	pump
t	turbine

1. Introduction

Electricity demand is increasing rapidly nowadays due to the massive development of technology and dependency on electric devices. However, traditional fossil fuel-based power systems are damaging from an environmental point of view as well as being harmful to human health. GE is a crucial factor in adopting ecofriendly systems such that it is one of the most favorable renewable energy sources (RESs) concerning emissions and environmental effects [1, 2]. With this recent focus on RESs, it is necessary to mention some related issues such as intermittency and negative impact on electric network. Thus, it is highly recommended to integrate other flexible sources to provide smooth and continuous power to the grid. However, this may be accompanied by emitting large amounts of CO_2 which is the case of diesel generators. To decrease these emissions, methanation could be applied which is a method of decarbonization used to recycle the CO_2 exiting the diesel generator [3]. Another attractive solution is the incorporation of energy storage systems and especially in applications involving wind and solar energies [4, 5]. Obviously, it is favorable to seek out green storage systems such as fuel cell [6], borehole thermal energy storage [7] and mechanical energy storage systems [8]. The latter is mainly used for short-term use and could be found in three different forms: flywheel [9], pumped hydro [10] and compressed air [11]. These are highly recommended to be used in

https://doi.org/10.1016/j.tsep.2021.100948

small grid district energy systems [12, 13]. However, in thermal applications, latent heat storage materials [14] are the most frequently used energy storage systems due to their suitability with such applications and waste heat recovery technologies [15, 16].

In contrast with other RESs, GE is a stable energy source since it is not highly affected by ambient conditions. GE can be divided into two types that are shallow and deep systems as shown in Figure 1. Shallow GE has been frequently integrated into air conditioning systems to improve their performance by using a ground coupled heat exchanger that can be found in the form of ground source heat pump [17, 18] or earth-air heat exchanger [19, 20]. The major component in a shallow geothermal system is the ground heat exchanger (GHE) which is surrounded by grout material and installed in a borehole [21, 22]. It could be found in three different forms: vertical [23], horizontal [24] and slinky [25]. The slinky-shaped GHE is usually used to decrease the volume of ground installation. This type could also be installed in two different ways; either vertically or horizontally. Esen *et al.* [26] investigated the difference between the two slinky types in a solar assisted ground source heat pump application. The authors deduced that the coefficient of performance is always higher when using the horizontal slinky GHE.

https://doi.org/10.1016/j.tsep.2021.100948

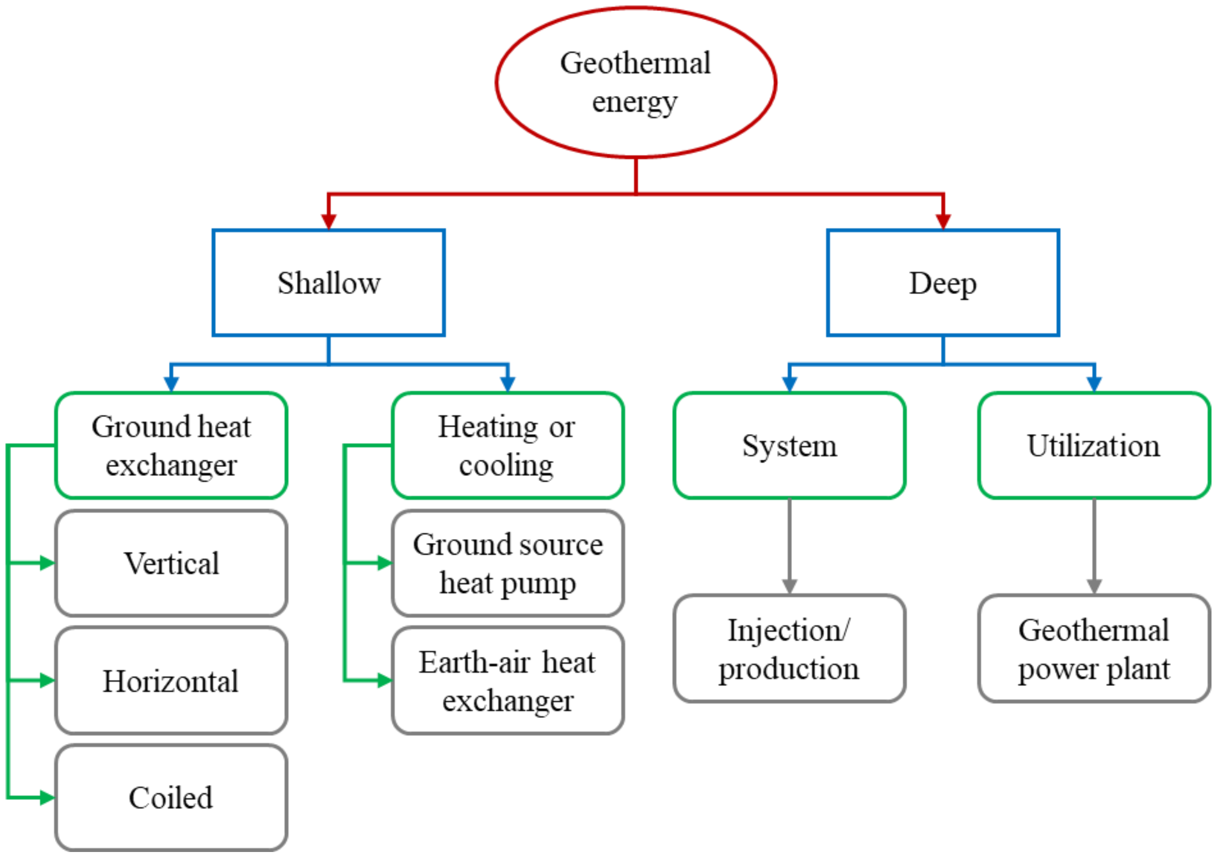


Figure 1: Geothermal energy; types, installations, and main utilizations

Even though GE is often considered as a low-grade source, but it can compete with traditional systems especially in those countries where the levelized cost of electricity is high. Deep GE can be used as a heat source in a standalone power plant, combined heat and power [27] or combined cooling, heat and power [28, 29]. Usually, all previous studies that investigated geothermal energy in power generation have used the ground as a heat source [30, 31]. Such systems are based on extracting hot fluid from deep geothermal layers to heat up the working fluid of a power cycle [32, 33]. Geothermal power plants can be found mainly in three different forms: dry steam [34], binary [35, 36] and flash cycle [37, 38] depending on the available geothermal conditions. The latter is usually used as a single [39] or double flash cycle [40] by using one or two flash separators, respectively. These

https://doi.org/10.1016/j.tsep.2021.100948

systems have also passed through several enhancements to improve the cycle's performance such as the combined flash-binary [41] and regenerative cycle [42]. One of the most recent attracting applications reported on the use of deep GE systems is hydrogen production in which it suits well the nature of such power plants. This is due to that the power cycle does not usually absorb all heat energy carried by the geothermal fluid before being reinjected to the ground. For this reason, the remaining energy can be further extracted by preheating water entering an electrolyzer which is responsible for producing hydrogen [43]. A key issue in the geothermal power plant is the choice of working fluid in the organic Rankine cycle (ORC) which is responsible for generating power in the case of low-grade source of energy. To select the most suitable fluid, it is necessary first to study the available geothermal conditions and mainly the temperature. In addition to the choice of working fluid, the design of the power cycle has also a huge effect on the overall performance. Lu *et al.* [44] compared the single flash with double flash cycles and concluded that the latter has the lowest power generation cost and shortest payback period. Currently, several studies are investigating hybrid GE systems [45, 46] such that the ground source is used to support the main unit. One of the most frequent explored applications on hybrid geothermal systems is to use the ground as a preheater followed by solar power unit [47].

Recently, CO_2-based transcritical Rankine cycles (CTRC) [48, 49] and supercritical Rankine cycles (CSRC) [50, 51] are being adopted instead of ORC due to the abundancy of CO_2 and superior heat transfer characteristics. Based on exergo-economic analysis, Wang *et al.* [52] confirmed that CO_2 could recover more energy than organic fluids in a Brayton cycle waste application with a low compressor pressure ratio. CO_2 could fit with

https://doi.org/10.1016/j.tsep.2021.100948

both waste heat recovery applications; low [53] and high grades [54] of energy. It has small size parameter (SP) compared to the ORC. As reported in Ref. [55], adding preheater and regenerator can increase the output power and thermal efficiency by 150% and 184% respectively but may be accompanied by increase in the heat exchangers' sizes [55].

In this paper, CTRC is used to generate electricity from different heat source conditions while using the underground water as a condenser. The innovative aspect in this study is to use shallow geothermal energy as a cooling source (condenser) in a Rankine power cycle to improve its performance. Geothermal condenser can provide lower cooling medium temperature, so that the cycle's low pressure could be decreased accordingly. Thus, the net output power can be maximized since it mainly depends on the pressure difference between the gas heater and condenser. The different conditions of the cycle are discussed in this paper such as pressures, temperatures, and flow rate to optimize the cycle's performance and heat exchangers' size. The software used for calculating the thermodynamic properties and developing the code is the Engineering Equation Solver (EES) which is appropriate for thermodynamic analyses.

2. Proposed System

The aim of this paper is to study a new system using CTRC with a ground-cooled condenser. It is proposed to generate electricity especially in summer because the ground temperature under a certain depth is approximately independent of the ambient conditions. Therefore, the ground heat exchanger is a better choice compared to the air heat exchanger when the ambient temperature is high during summer. To provide the suitable heat transfer conditions, the geothermal condenser is embedded in a well to facilitate the process of heat

https://doi.org/10.1016/j.tsep.2021.100948

exchange by the help of underground water. Figure 2 presents the proposed CTRC showing its all components.

In this research study, the underground water temperature is assumed to be constant which means that it is not affected by the amount of heat rejected from the cycle to the ground. This assumption can be considered only in two cases, either the well's volume is relatively very large, or the ground-cooled condenser is functioning as a secondary condenser. The latter is highly recommended to ensure stability since the primary condenser (air/water-cooled) can reduce the amount of heat rejected to the ground and compensate the coolth energy during off-periods. In this case, the ground/underground water can be used as a cooling storage medium and especially because the proposed system is mainly designed to be convenient for waste heat recovery applications [56, 57] knowing that such applications usually do not operate 24 hours/day.

https://doi.org/10.1016/j.tsep.2021.100948

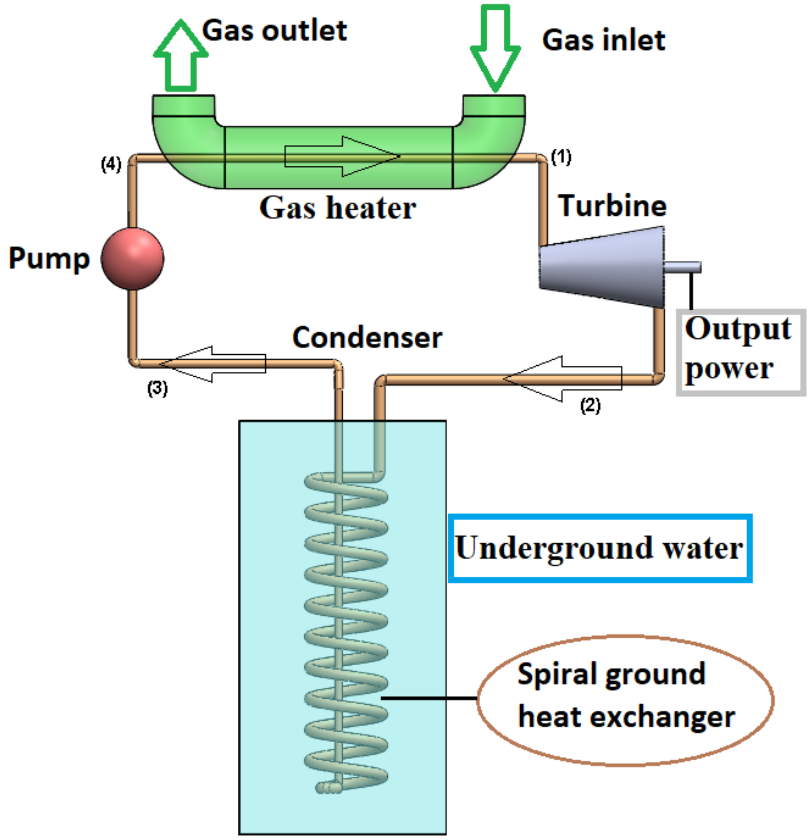


Figure 2: CO_2-based transcritical Rankine cycle (CTRC) combined with an underground water condenser

As shown in Figure 2, the condenser selected is a helical GHE to minimize the total volume of the GHE. The gas heater is chosen as a counter flow heat exchanger also to obtain the minimum possible size. This study covers several variable conditions of gas, cycle, and underground water to find out the optimal conditions for the cycle and minimum size of the heat exchangers.

3. Modelling and Calculation

The model presented in this section shows the equations used for calculating the cycle's performance and heat exchangers' lengths that were developed in EES. The thermodynamic performance of the power plant depends on the amount of heat extracted

https://doi.org/10.1016/j.tsep.2021.100948

from the gas, choice of pressures and temperatures of the cycle and the efficiencies of turbine, pump, and generator. The net output power can be calculated by:

$$\dot{W}_{net} = \dot{W}_t . \eta_g - \dot{W}_p \quad (1)$$

where $\dot{W}_t$ is the turbine power, η_g is the generator efficiency and $\dot{W}_p$ is the pump power. $\dot{W}_t$ and $\dot{W}_p$ mainly depend on the mass flow rate of CO_2 and the enthalpy variation. After calculating the net output power, the efficiency can be evaluated as follows:

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \quad (2)$$

where $\dot{Q}_{in}$ is the heat extracted from the gas and added to the cycle (gas heater). The value of heat added is based on the specific heat, flow rate and temperature of hot gas. It is necessary then to check the area of heat exchangers involved in the proposed system shown in Figure 2 (gas heater and condenser). The lengths of gas heater and condenser are mainly affected by the amount of heat added and extracted from the cycle, respectively. The area of each heat exchanger is based on the energy balance equation (between CO_2 and heat source/sink) which is represented by:

$$\dot{Q} = U. A. \Delta T_{lm} \quad (3)$$

where $\dot{Q}$ is the heat added or extracted to/from the system, U is the overall heat transfer coefficient that depends on the convection heat transfer coefficient of both fluids and the thermal conductivity of the pipe, A is the heat exchanger's area and ΔT_{lm} is the logarithmic mean temperature difference. The heat transfer coefficient of all fluids involved (hot gas, underground water, and CO_2) is given by:

https://doi.org/10.1016/j.tsep.2021.100948

$$h = \frac{Nu.k}{L_r} \tag{4}$$

where *Nu* is the Nusselt number, *k* is the thermal conductivity and L_r is the characteristic length. In the case of internal flow forced convection, the hydraulic diameter is considered as the characteristic length and the following equation could be used to calculate the Nusselt number of hot gas and CO_2 as a working fluid [58]:

$$Nu = 0.023.Re^{0.8}.Pr^{n} \tag{5}$$

where *Re* and *Pr* are the Reynolds and Prandtl numbers, respectively. *n* is equal to 0.3 in case of cooling and 0.4 for heating. When the velocity of the fluid is equal to zero, free convection will take place which is the case of underground water. According to Ref. [59], the following Nusselt number correlation could be used for coiled heat exchanger natural convection applications:

$$Nu = Ra^{0.3071}\left(\frac{H}{D}\right)^{-0.1097} \tag{6}$$

where *Ra* is the Rayleigh number, *H* is the height of the coil and *D* is the coil diameter. This correlation is only considered when the characteristic length is taken as the coil length (L_c) which is presented in equation 7 [59].

$$L_c = n_c({p_c}^2 + \pi^2 D^2)^{0.5} \tag{7}$$

where n_c is the number of turns and p_c is the pitch (distance between two consecutive turns).

4. Validation

The proposed model was validated by comparing the net output power with the CO_2-RC studied by Shu *et al.* [60] based on the conditions presented in Table 1. The cycle was used

https://doi.org/10.1016/j.tsep.2021.100948

as a heat recovery system at the exhaust of an engine. The exhaust gas is formed of 19.84% CO_2, 8.26% H_2O and 71.49% N_2.

Table 1: The parameters considered for the model validation [60]

Parameters (Units)	**Values**
Gas inlet temperature (°C)	777
Gas outlet temperature (°C)	120
Gas flow rate (kg/h)	202.6
CO_2 condensation temperature (°C)	25
Pump isentropic efficiency	0.8
Turbine isentropic efficiency	0.7
Generator efficiency	0.9

Figure 3 presents the validation of the proposed model by displaying the variation of net output power as a function of the temperature entering the turbine (100-700°C) such that (a) and (b) are the results obtained from Ref. [60] and (c) is the result of the CTRC used in the current research.

https://doi.org/10.1016/j.tsep.2021.100948

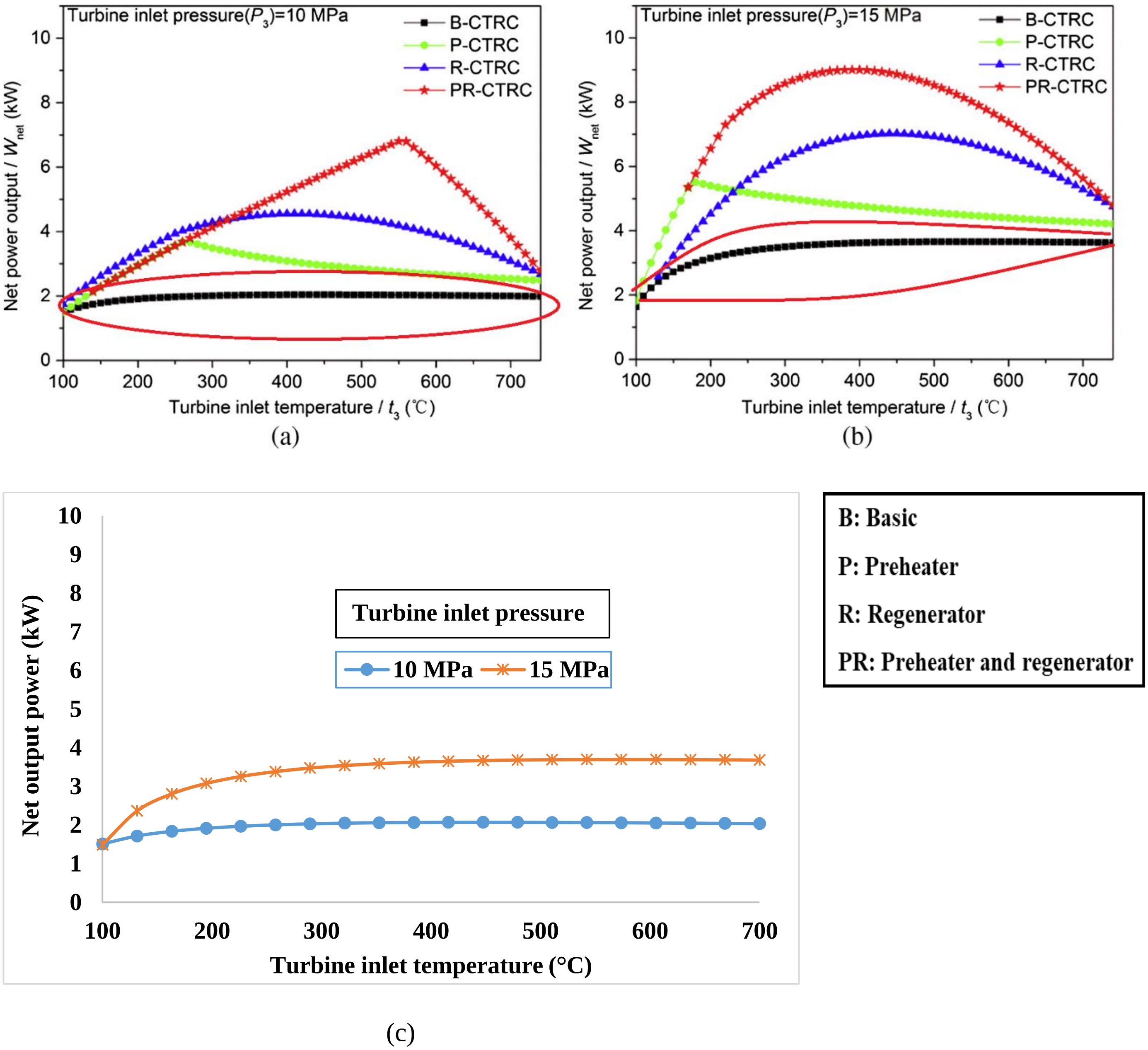


Figure 3: The variation of net output power as a function of turbine inlet temperature; (a) and (b) are obtained from [60] at 10 MPa and 15 MPa respectively while (c) refers to the proposed model

According to Figure 3, the proposed model behaves as expected due to the similarity in the net output power generated at two different operating pressures (10 MPa and 15 MPa) compared to the results obtained by Shu *et al.* [60]. As the turbine inlet temperature

https://doi.org/10.1016/j.tsep.2021.100948

increases from 100°C to 700°C, the power generated increases approximately from 1.8 kW to 2 kW and from 2 kW to 3.9 kW at p_1 = 10 MPa and p_1 = 15 MPa, respectively.

5. Results and Discussion

In this section, some parameters are always fixed while others could either be constants or variables. This will be clearly presented in each subsection such as when a parameter is not mentioned as a variable then it is assumed to be constant if not varying as an output. The different parameters to be studied are the turbine/gas inlet temperatures, turbine inlet pressure, underground water temperature, mass flow rate CO_2/gas, cycle's performance, and heat exchangers' sizes.

5.1 Design conditions

Table 1 and Table 2 present all cycle's conditions such that these values are considered as a reference for the upcoming simulations. However, some of them may vary aiming is to study a specific relation which will be clearly presented within the discussion and analysis of each result. The gas components are still the same as mentioned in section 4, even though they refer to an engine waste, but this study could fit any other heat source because a range of gas flow rate and temperature are examined. The main difference that may vary from one application to the other is the heat transfer coefficient of gas since it is directly affected by the gas components.

Table 2: Parameters considered for the proposed model

Parameters (Units)	**Values**
Gas pipe diameter (mm)	40
Pipe diameter of the cycle (mm)	20

https://doi.org/10.1016/j.tsep.2021.100948

Turbine inlet pressure (MPa)	15
Underground water temperature (°C)	15
Turbine inlet temperature (°C)	350
Coil diameter (cm)	50
Coil pitch (cm)	2

5.2 Thermodynamic performance

The performance of the cycle depends on three main parameters: working fluid's mass flow rate, net output power and thermal efficiency. It is very crucial to study the thermodynamic behavior of the cycle to find out the best conditions required to generate the maximum amount of power. The aim of this section is to study the effect of operating conditions (turbine inlet pressure and temperature, mass flow rate, and gas inlet temperature) on the mass flow rate of working fluid, output power and cycle's efficiency.

Figure 4 shows that the mass flow rate of the working fluid (CO_2) and turbine inlet temperature (T_1) are inversely proportional. At low T_1 (below 200°C), there is a slight difference between the mass flow rate of CO_2 at the different presented turbine inlet pressures, while it starts do decrease gradually to be nil after that. This is mainly due to the dependency of CO_2 enthalpy on the pressure at low temperatures. However, at high temperatures, the enthalpy of CO_2 entering the turbine is slightly affected by the change in pressure. On the other hand, the variation in turbine inlet pressure at high temperatures only changes the enthalpy of CO_2 exiting the turbine and hence affects the fluid's expansion in the turbine and output power. Therefore, the mass flow rate of the working is affected considerably by the turbine inlet temperature while insignificantly by the pressure.

https://doi.org/10.1016/j.tsep.2021.100948

In this simulation study, the amount of heat extracted form gas is assumed to be constant as shown in Table 1 since the gas flow rate and temperatures are fixed. This means that the enthalpy at state one and the working fluid's mass flow rate vary oppositely based on the energy balance between the gas and CO_2 in the gas heater. Besides, the temperature and enthalpy are directly proportional showing the reason for the decrease in mass flow rate (see Figure 4). According to equation (1), it is expected that the output power varies similarly as the mass flow rate of the working fluid. Conversely, Figure 3-c and Figure 4 have showed a different relation between the power generated and mass flow rate. This is due to that the net output power is highly affected by the enthalpy difference between inlet and outlet states of the turbine.

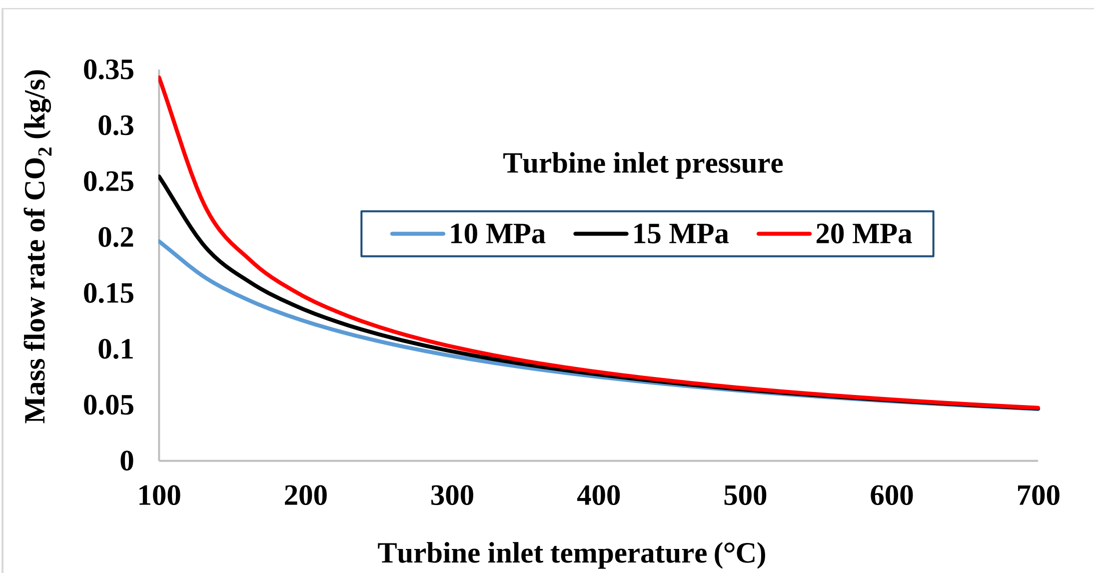


Figure 4: The effect of turbine inlet temperature (T_1) on the mass flow rate of CO_2 at different Pressures (p_1)

The efficiency of the cycle is highly affected by the change of turbine inlet pressure (p_1) as shown in Figure 5. The cycle's efficiency is independent from the mass flow rate of the working fluid in contrast with the output power (see equations 1 and 2). Almost when the turbine inlet temperature is higher than 150°C, the increase in p_1 could lead to an increase

https://doi.org/10.1016/j.tsep.2021.100948

in the efficiency such that the highest efficiency is achieved at turbine inlet pressure of 20 MPa which is around 11%. However, the efficiency usually increases with the turbine inlet temperature (T_1). This relation is also more effective at high pressures which is the case at 20 MPa since the efficiency rose remarkably from 4% to 11% while the T_1 varied from 150°C to 300°C. On the other hand, at 10 MPa, the efficiency did not show any remarkable jump under the same range of temperature. This does not mean that the efficiency is not affected by the turbine inlet temperature at low pressures, however, this depends on the range of temperature studied. The reason is that the efficiency is almost constant after exceeding a specific temperature in which the latter is proportional to the turbine inlet pressure. Thus, it is expected that the sudden rise in the cycle's efficiency at 10 MPa had already occurred below 100°C. At low temperatures (below 150°C), the efficiency referring to the highest turbine inlet pressure becomes the lowest. At these conditions, the cycle cannot be considered as transcritical cycle yet. Thus, it is very necessary to use high temperatures to ensure that the CO_2 is totally superheated otherwise a significant drop in the efficiency and power may occur. This may also be accompanied by a damage in the turbine since the nature of fluid entering it is not only vapor.

https://doi.org/10.1016/j.tsep.2021.100948

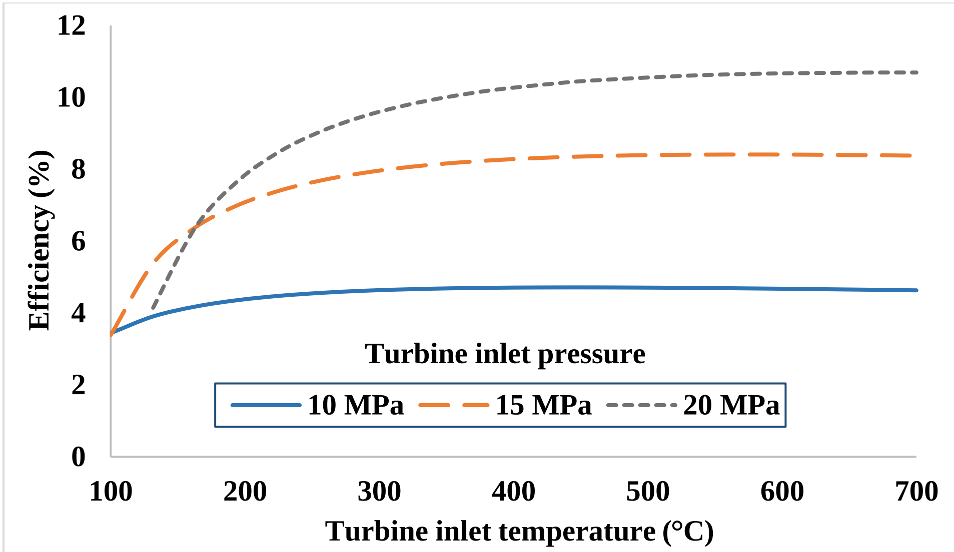


Figure 5: The effect of turbine inlet temperature (T_1) on the cycle's efficiency at different pressures (p_1)

The surface plot presented in Figure 6 shows the influence of gas temperature and its mass flow rate on the cycle's net output power. The proportional relation is very noticeable with an equality of influence, which means that both have approximately the same effect. This could also be verified since the vertex of the plot is at the highest gas temperature (1500°C) and mass flow rate (350 kg/hr), and that the surface plot has the same rate of increase with respect to the two axes. It is also essential to compare the opposite sides of the surface plot to study the effect of both variables on each other. In other words, the increasing rate of output power with respect to gas inlet temperature is higher at 350 kg/hr than that at 100 kg/hr such that it changes approximately from 3.8 kW to 14.5 kW and 1 kW to 6 kW, respectively. Similarly, the output power increases with a higher rate with respect to the mass flow rate at 1500˚C compared to that at 500˚C.

https://doi.org/10.1016/j.tsep.2021.100948

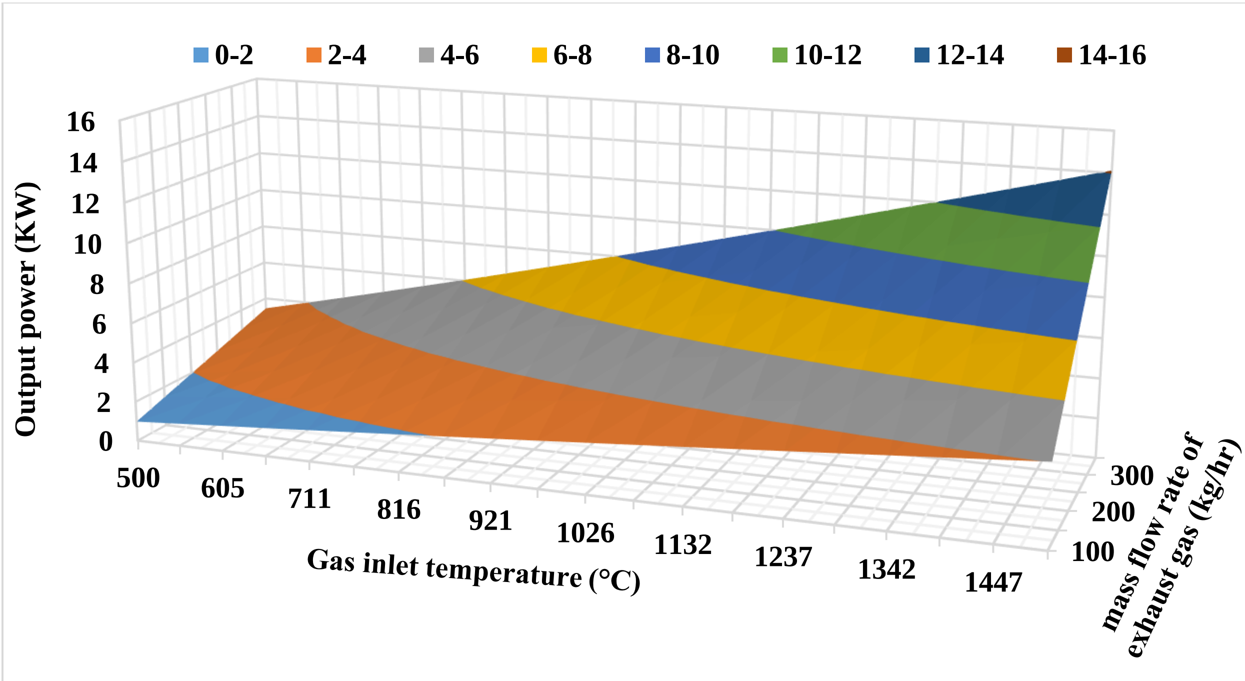


Figure 6: The variation of net output power as function of the gas temperature (T_{gi}) and its mass flow rate ($\dot{m}_g$)

The underground water temperature is one of the most important uncontrollable parameters, thus, it is necessary to study its effect before installation to determine the expected feasibility in terms of performance. Figure 7 shows the effect of underground water temperature on the cycle's efficiency and net output power. Both are inversely proportional to the water temperature. The same relation was obtained when varying the pinch temperature which is the minimum temperature difference between the working fluid and the underground water (cooling source). These results verify the direct effect of condenser's pressure on the cycle's performance. Therefore, as the pressure of the condenser increases, the output power decreases due to the reduction in enthalpy difference between the inlet and outlet turbine states. This shows the importance of using a ground-cooled condenser in summer which is the main reason of the current investigation. For example, if the underground water temperature is less than that of the ambient by 10˚C,

https://doi.org/10.1016/j.tsep.2021.100948

then the ground-cooled condenser will be able to increase the output power by 1 kW approximately compared to that of air-cooled condenser. This can be deduced from Figure 7 such that for each 1˚C decrease in the coolant temperature, a 0.1 kW additional power could be produced.

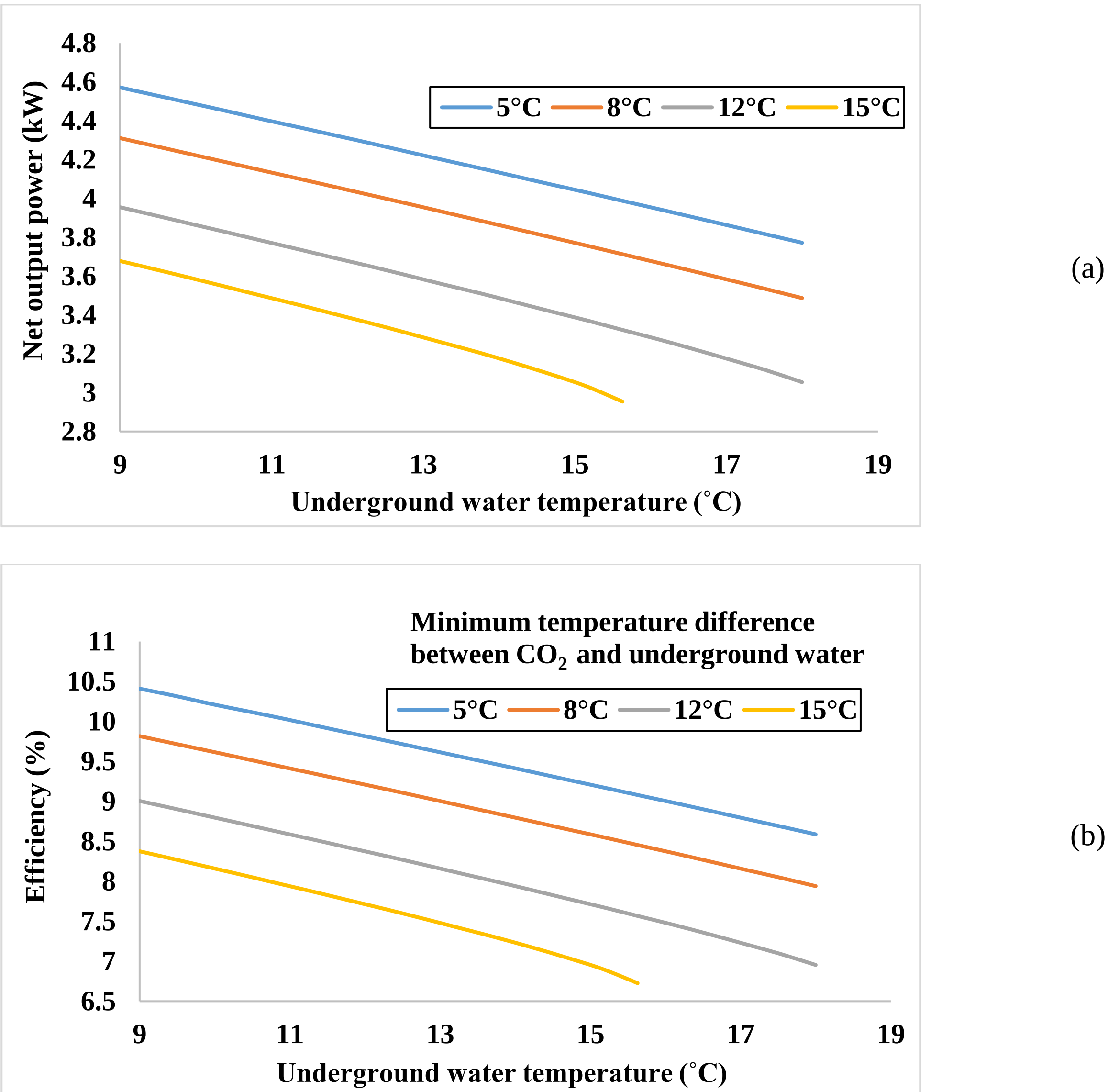


Figure 7: The effect of underground water temperature on the (a) net output power and (b) efficiency with respect to the minimum temperature difference in the condenser

https://doi.org/10.1016/j.tsep.2021.100948

5.3 Gas heater's size

The gas heater is usually considered as the most critical component because it is the heat exchanger responsible for extracting the maximum possible amount of heat. This objective must be achieved considering the smallest acceptable heat exchanger's size to save money and space.

According to Figure 8, as the pipe diameter of the CTRC increases, the length of the gas heater decreases in contrast to the effect of gas pipe diameter. In addition, the influence of gas pipe diameter on the heater's length appears more significantly at low CTRC pipe diameter. This impact is not only due to the variation in heat exchanger surface area and hydraulic diameter, but it also depends on the convective heat transfer coefficient which is directly affected by the flow velocity inside the pipe. Even though it is better to use small gas pipe diameters to decrease the total size of the gas heater, it is also necessary to study the effect of hydraulic diameter on the gas flow. This depends on other specifications that need to be taken into consideration in further studies to make sure that the gas is smoothly flowing without any negative effect on the systems' components and cycle's performance.

https://doi.org/10.1016/j.tsep.2021.100948

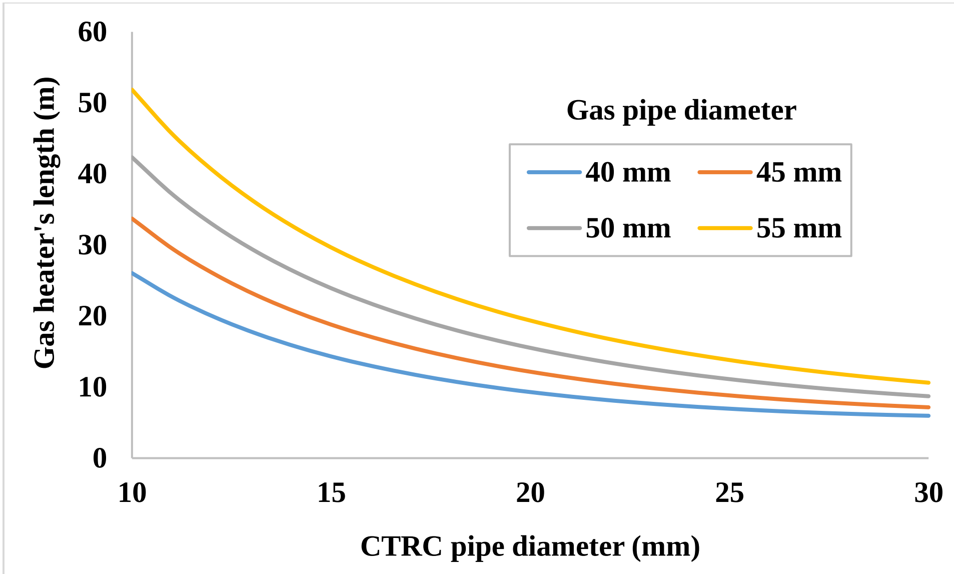


Figure 8: The relation between gas heater's length and the diameter of both pipes; gas and CO_2 which is the circulating through the CTRC

As the gas inlet temperature rises from 500°C to 1500°C, the length of the gas heater is reduced in all cases (see Figure 9), while this relation is inverted when varying the mass flow rate of gas. Although both parameters have the same influence on the heat added to the cycle, but they are also involved in the length's calculation from another side in which the overall heat transfer coefficient and logarithmic mean temperature difference are affected by the mass flow rate and temperature, respectively. The major parameter that affects the heat transfer coefficient in forced convection is the velocity which is a function of the mass flow rate. Moreover, the logarithmic mean temperature difference is one of most crucial factors that can change the size of the heat exchanger which represents the temperature difference between the working fluids. Thus, it can be deduced that the effect of gas temperature on the heat exchange area is higher than that on heat addition which is opposite to that of mass flow rate.

https://doi.org/10.1016/j.tsep.2021.100948

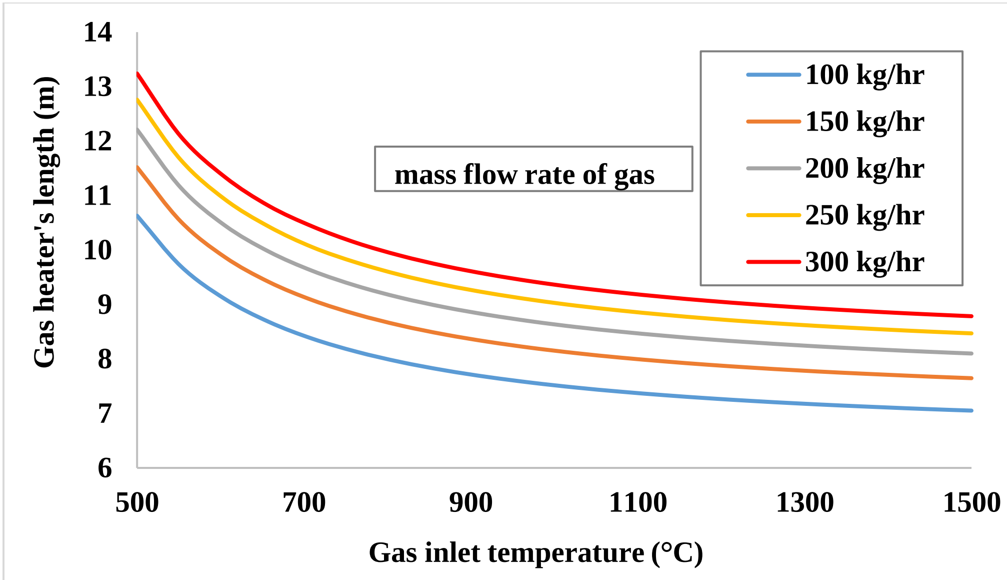


Figure 9: The effect of gas conditions (flow rate and temperature) on the gas heater's length

5.4 Condenser's size

The area of the condenser is a crucial part to be studied very precisely to make sure that the underground water can extract the required amount of heat from the cycle. This is very important due to the high complexity of maintenance since it is installed under the ground.

Figure 10 presents a linear relation between the gas inlet temperature and condenser's length, however, the ratio of proportionality is controlled by the mass flow rate of gas (diverging). The slope of condenser's length function is proportional to the mass flow rate with respect to the gas inlet temperature. For example, the condenser's length increases from 18 m to 32 m and 27 m to 65 m at 100 kg/hr and 300 kg/hr respectively. These relations verify the change in the mass flow rate of CO_2 which directly modifies the amount of heat rejected and thus, the condenser's length. In addition to that, it is necessary to keep in mind that they also have an influence on the convection heat transfer coefficient of the CO_2 within the condenser due to the change in the CO_2 mass flow rate.

https://doi.org/10.1016/j.tsep.2021.100948

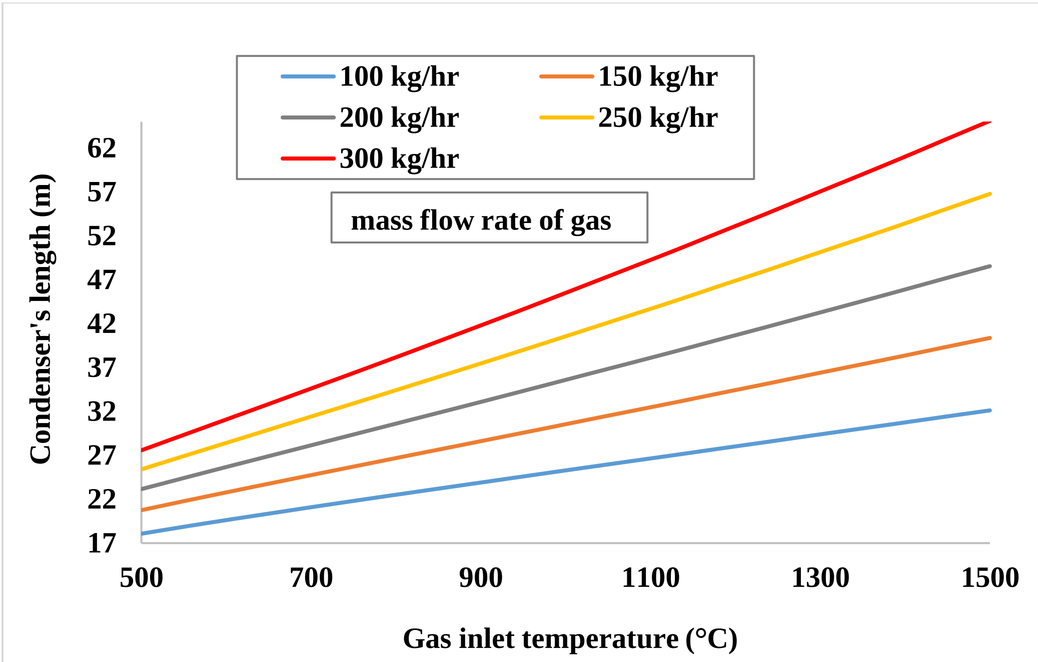


Figure 10: The effect of gas conditions (flow rate and temperature) on the gas heater's length

Figure 11 shows that the underground water has a great influence on the length of the condenser which can be obviously recognized from the significant empty space between the two curves corresponding to 15°C and 18°C water temperatures. However, all curves are taking the shape of wide parabola which means that for every specific underground water temperature there is a specific CTRC pipe diameter that corresponds to the minimum condenser length. The value of this diameter is approximately around 15 mm. However, this must not be the only criterion for selecting the optimal diameter. This is due to the diameter's effect on pressure drop which may affect other parameters in the cycle such as the net output power. This means that in some cases, choosing the smallest diameter would not be the best solution even though it clearly reduces the length of the condenser. Thus, it is very necessary to balance between the heat exchangers' sizes and overall cycle's performance.

https://doi.org/10.1016/j.tsep.2021.100948

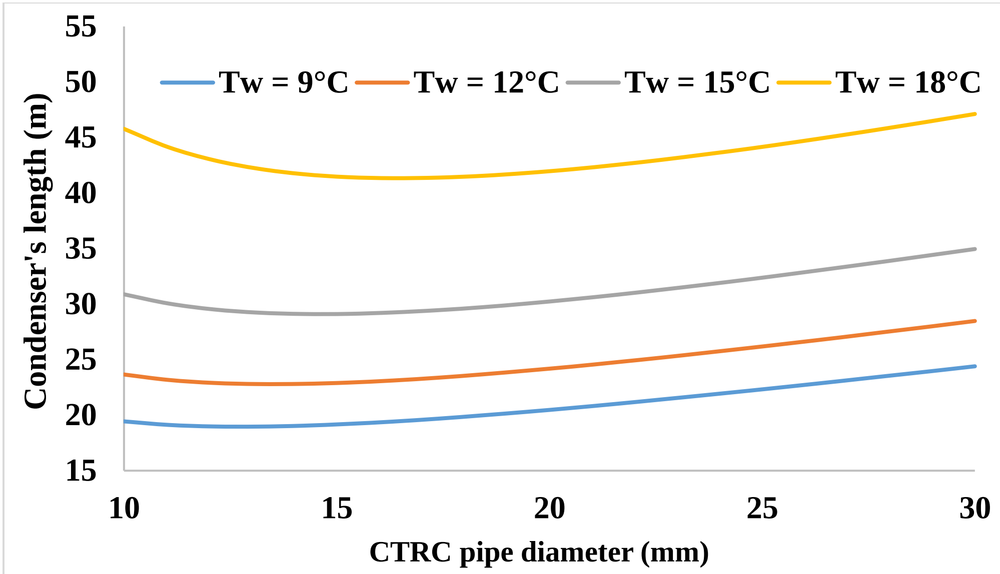


Figure 11: The variation of condenser's length as a function of CTRC pipe diameter at several underground water temperatures (T_w)

The result shown in Figure 12 is based on varying the condensation temperature similarly as the underground water temperature such as the difference between them is the minimum temperature difference between CO_2 and water. This pinch temperature has a greater effect on the condenser's length than that of the underground water temperature. This is due to the inversely proportional relation between the logarithmic mean temperature difference and the heat exchanger's area as shown in equation (3). In addition, the rate of decrease in the length is more sensitive at high pinch temperature difference. Figure 11 and Figure 12 seem to be presenting opposing results, however, the difference is that the condensation temperature in Figure 11 is kept constant at 25˚C which means that the net output power is also fixed. However, for a specific curve in Figure 12, as the underground water temperature increase the condensation temperature of CO_2 will increase similarly. This will consequently lead to a drop in the net output power due to the change in condenser's pressure. Figure 12 shows that it is a good option to fix the minimum temperature difference between the two fluids to minimize the condenser's length and hence to reduce

https://doi.org/10.1016/j.tsep.2021.100948

the capital cost of installation. On the other hand, this reduction must be balanced with the decrease in output power as shown in Figure 7 which is related directly to the operating cost.

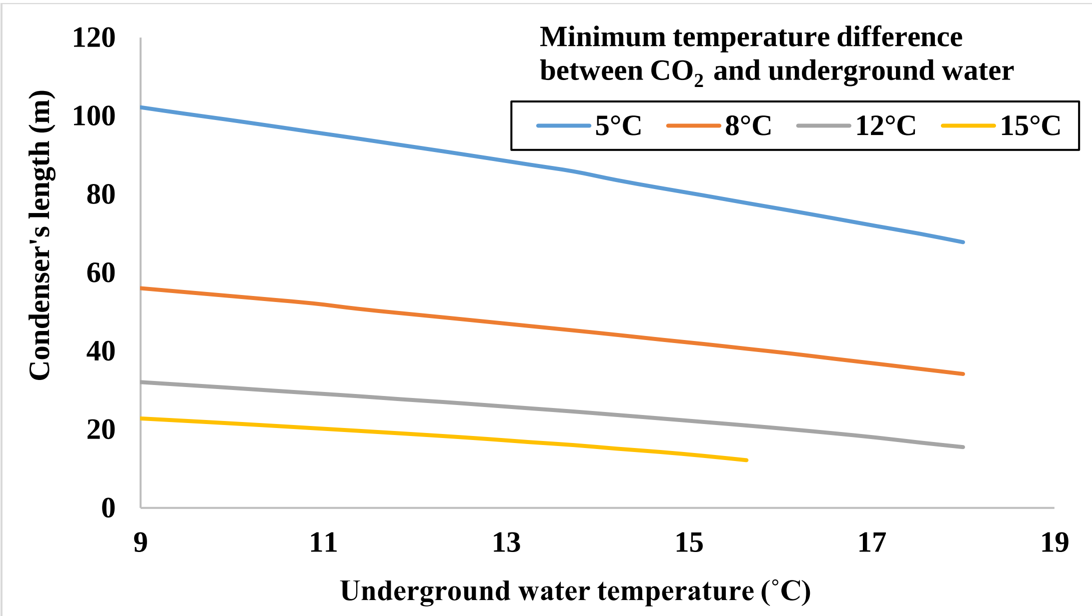


Figure 12: The effect of underground water temperature under different minimum temperature differences on the condenser's length

Figure 13 presents the major dimensions' parameters of the condenser. The relation between coil height, coil diameter and pipe diameter must be studied to choose the suitable volume. The curves are diverging when the pipe diameter is increasing. This shows that the variation in coil diameter is more effective on the condenser's length at large pipe diameters. The relation between pipe diameter and coil height is similar to that with total condenser's length presented in Figure 11 because at a specific coil diameter, the variation of coil height represents the same variation of length. Indeed, it is favorable to choose the lowest pipe diameter to minimize the coil height and consequently achieving the smallest volume of GHE. However, there are some other parameters that need to be taken into

https://doi.org/10.1016/j.tsep.2021.100948

consideration and investigated in future studies such as thermal pollution/imbalance and pressure drop.

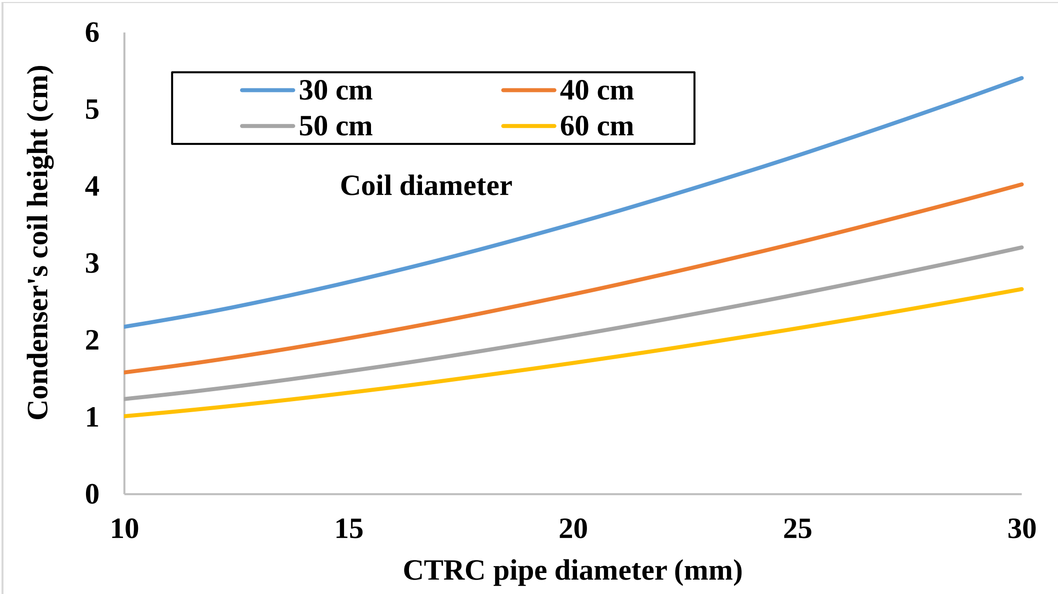


Figure 13: The variation of condenser's coil height with respect to the pipe and coil diameters

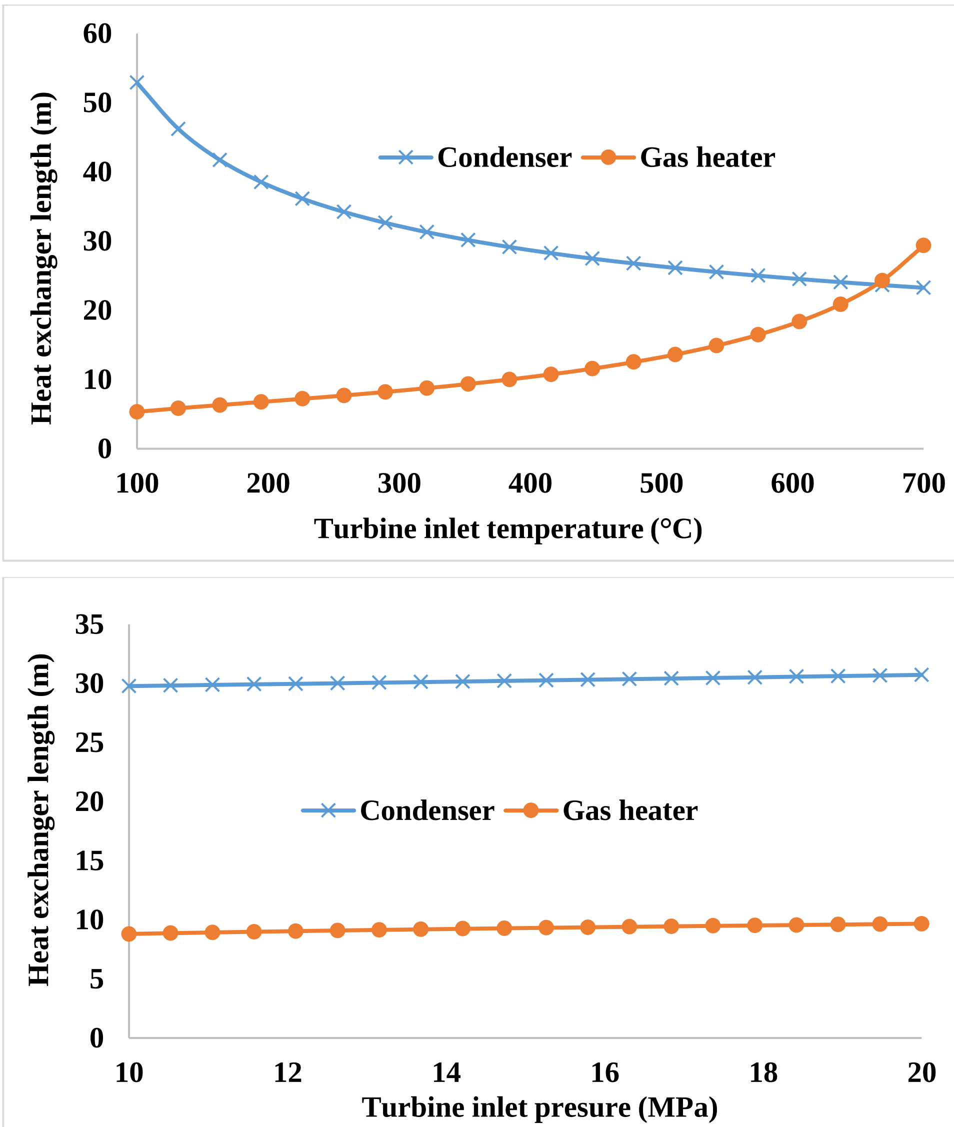
5.5 Gas heater vs condenser

This section will present a comparison between the heat exchangers in the proposed CTRC (gas heater and condenser) to balance between the two sizes. This will contribute to choosing the best optimum conditions to select the heat exchangers' sizes accordingly.

The turbine inlet temperature has more influence on the condenser's length compared to that of the gas heater (see Figure 14). From this point of view, it could be very beneficial to choose a high turbine inlet temperature so that both heat exchangers are of moderate lengths which is around 650°C (both lengths were approximately 23 m). The amount of heat added to the cycle is constant since it is only a function of gas flow rate and temperatures. Thus, the turbine inlet temperature will vary oppositely to the mass flow rate of CO_2 based on the energy balance between the two fluids. As the mass flow rate of

https://doi.org/10.1016/j.tsep.2021.100948

working fluid decreases the heat rejected from CTRC will decrease too leading to a drop in the condenser’s length. On the other hand, the turbine inlet pressure has no significant influence on the heat exchangers’ lengths such that condenser’s and heater’s length remain around 30 m and 9 m, respectively.



(a)

(b)

Figure 14: The difference between heat exchangers' lengths (gas heater and condenser) with respect to the (a) turbine inlet temperature (T_1) and (b) pressure (p_1)

After studying the influence of the highest temperature in the cycle (T_1), it is also important to study the lowest one which corresponds to the condensation (T_3). As shown in Figure

https://doi.org/10.1016/j.tsep.2021.100948

15, it will not affect the heater's length, while it remarkably changes that of the condenser and the net output power since they vary from 80 m to 13 m and 4.2 kW to 3.2 kW respectively for condensation temperature between 20°C and 30°C. The drop in power is due to the decrease in expansion while that of condenser's size because of variation in the amount of heat extracted and the logarithmic mean temperature difference. As the condensation temperature rises, the difference between fluid and underground temperature will increase too which improves the conditions of heat transfer and hence, a smaller heat exchanger will be required. Thus, it is very necessary to balance between the heat exchangers' lengths and the net output power to make sure that both meet the desired requirements. In addition, if the cycle is going to be designed considering a high condensation temperature, then it will not be feasible to use a ground-cooled condenser. The most important specification in this system is the ability to operate at low condensation temperatures in summer compared to the ambient temperature to increase the net output power of the cycle.

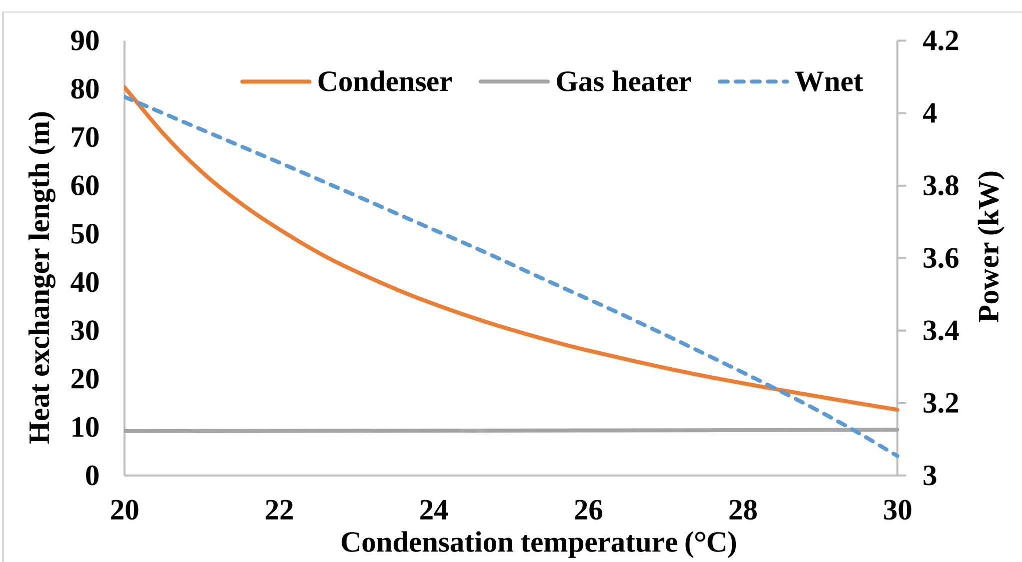


Figure 15: The variation of heat exchangers' lengths (gas heater and condenser) and net output power as a function of condensation temperature (T_3)

https://doi.org/10.1016/j.tsep.2021.100948

6. Conclusion

The proposed CTRC coupled with ground-cooled condenser can generate electricity from several heat sources such that it could produce a perfect amount of power that depends on the cycle's conditions and gas specifications. The parametric analysis contributes to extract the maximum amount of energy while preserving small heat exchangers. A positive linear relation was noticed between the net output power and the available heat (gas inlet temperature and flow rate) such that the power was 1 kW at T_{gi} = 500°C and $\dot{m}_g$ = 100 kg/hr while it jumps to 14.2 kW at T_{gi} = 1500°C and $\dot{m}_g$ = 350 kg/hr. The use of ground-cooled condenser generated approximately 30% more power compared to the conventional Rankine cycle. This enhancement is mainly affected by the ground's temperature since it controls indirectly the working fluid's expansion. The turbine inlet temperature was found to be an effective variable such that it does not affect the net output power significantly while providing acceptable heat exchanger's lengths. The second crucial parameter to be controlled is the pipe diameter of CO_2 since it could be used also to balance between both heat exchangers. The increase in the pipe diameter almost leads to a decrease in the gas heater's length while retaining an optimal condenser's length; a specific pipe diameter must be selected. Thus, a 15 mm pipe diameter seem to be favorable for the given cycle conditions.

Further studies should investigate CTRC enhancements on a specific waste heat recovery application and underground conditions while comparing this system with the conventional Rankine cycle and ORC. It is also necessary to compare the capital and operating costs between ground-cooled and air-cooled condensers to examine the economic feasibility of

https://doi.org/10.1016/j.tsep.2021.100948

the proposed system. Furthermore, to make the results of simulation more reliable, it would be helpful to consider the variation in turbine efficiency.

References

[1] G.S. Jia, Z.Y. Tao, X.Z. Meng, C.F. Ma, J.C. Chai, L.W. Jin, Review of effective thermal conductivity models of rock-soil for geothermal energy applications, Geothermics, Volume 77, 2019, Pages 1-11, ISSN 0375-6505, https://doi.org/10.1016/j.geothermics.2018.08.001.

[2] Montaser Mahmoud, Mohamad Ramadan, Sumsun Naher, Keith Pullen, Abdul-Ghani Olabi, The impacts of different heating systems on the environment: A review, Science of The Total Environment, Volume 766, 2021, 142625, ISSN 0048-9697 https://doi.org/10.1016/j.scitotenv.2020.142625.

[3] Ionel Balcu; Ionel Ciucanu, Corina Macarie, Bogdan Taranu, Dana-Alexandra Ciupageanu, Gheorghe Lazaroiu, Virgil Dumbrava, Decarbonization of Low Power Applications through Methanation Facilities Integration, 2019, IEEE PES Innovative Smart Grid Technologies Europe (ISGT-Europe), https://doi.org/10.1109/ISGTEurope.2019.8905506.

[4] Sinem Işık, Cengiz Yıldız, Improving thermal energy storage efficiency of solar collector tanks by placing phase change materials in novel finned-type cells, Thermal Science and Engineering Progress, Volume 19, 2020, 100618, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2020.100618.

[5] Luanna Maria Silva de Siqueira, Wei Peng, Control strategy to smooth wind power output using battery energy storage system: A review, Journal of Energy Storage, Volume 35, 2021, 102252, ISSN 2352-152X, https://doi.org/10.1016/j.est.2021.102252.

[6] Emmanuel Ogungbemi, Oluwatosin Ijaodola, F.N. Khatib, Tabbi Wilberforce, Zaki El Hassan, James Thompson, Mohamad Ramadan, A.G. Olabi, Fuel cell membranes – Pros and cons, Energy, Volume 172, 2019, Pages 155-172, ISSN 0360-5442, https://doi.org/10.1016/j.energy.2019.01.034.

[7] Jerzy Wołoszyn, Global sensitivity analysis of borehole thermal energy storage efficiency on the heat exchanger arrangement, Energy Conversion and Management, Volume 166, 2018, Pages 106-119, ISSN 0196-8904, https://doi.org/10.1016/j.enconman.2018.04.009.

[8] Montaser Mahmoud, Mohamad Ramadan, Abdul-Ghani Olabi, Keith Pullen, Sumsun Naher, A review of mechanical energy storage systems combined with wind

and solar applications, Energy Conversion and Management, Volume 210, 2020, 112670, ISSN 0196-8904, https://doi.org/10.1016/j.enconman.2020.112670.

[9] Paulo Gambôa, J. Fernando Silva, S. Ferreira Pinto, Elmano Margato, Input–Output Linearization and PI controllers for AC–AC matrix converter based Dynamic Voltage Restorers with Flywheel Energy Storage: a comparison, Electric Power Systems Research, Volume 169, 2019, Pages 214-228, ISSN 0378-7796, https://doi.org/10.1016/j.epsr.2018.12.023.

[10] A. Setas Lopes, Rui Castro, Carlos S. Silva, Design of water pumped storage systems: A sensitivity and scenario analysis for island microgrids, Sustainable Energy Technologies and Assessments, Volume 42, 2020, 100847, ISSN 2213-1388, https://doi.org/10.1016/j.seta.2020.100847.

[11] Jie Song, Xiaodong Peng, Xiangjun Fang, Ying Han, Zhanfeng Deng, Guizhi Xu, Lixiao Liang, Jibiao Hou, Hongwei Wu, Thermodynamic analysis and algorithm optimisation of a multi-stage compression adiabatic compressed air energy storage system, Thermal Science and Engineering Progress, Volume 19, 2020, 100598, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2020.100598.

[12] Montaser Mahmoud, Mohamad Ramadan, Sumsun Naher, Keith Pullen, Ahmad Baroutaji, Abdul-Ghani Olabi, Recent advances in district energy systems: A review, Thermal Science and Engineering Progress, Volume 20, 2020, 100678, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2020.100678.

[13] Sepehr Foroushani, Jason Owen, Majid Bahrami, Data-driven modelling of operational district energy networks, Thermal Science and Engineering Progress, Volume 22, 2021, 100802, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2020.100802.

[14] Hussam Jouhara, Alina Żabnieńska-Góra, Navid Khordehgah, Darem Ahmad, Tom Lipinski, Latent thermal energy storage technologies and applications: A review, International Journal of Thermofluids, Volumes 5–6, 2020, 100039, ISSN 2666-2027, https://doi.org/10.1016/j.ijft.2020.100039.

[15] Hussam Jouhara, Delpech Bertrand, Brian Axcell, Luca Montorsi, Matteo Venturelli, Sulaiman Almahmoud, Massimo Milani, Lujean Ahmad, Amisha Chauhan, Investigation on a full-scale Heat Pipe Heat Exchanger in the ceramics industry for waste heat recovery, Energy, 2021, 120037, ISSN 0360-5442, https://doi.org/10.1016/j.energy.2021.120037.

[16] José J. Fierro, César Nieto-Londoño, Ana Escudero-Atehortua, Mauricio Giraldo, Hussam Jouhara, Luiz C. Wrobel, Techno-economic assessment of a rotary kiln shell

radiation waste heat recovery system, Thermal Science and Engineering Progress, 2021, 100858, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2021.100858.

[17] Roozbeh Sangi, Pooyan Jahangiri, Dirk Müller, A combined moving boundary and discretized approach for dynamic modeling and simulation of geothermal heat pump systems, Thermal Science and Engineering Progress, Volume 9, 2019, Pages 215-234, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2018.11.015.

[18] Zhenjun Ma, Lei Xia, Xuemei Gong, Georgios Kokogiannakis, Shugang Wang, Xinlei Zhou, Recent advances and development in optimal design and control of ground source heat pump systems, Renewable and Sustainable Energy Reviews, Volume 131, 2020, 110001, ISSN 1364-0321, https://doi.org/10.1016/j.rser.2020.110001.

[19] Samia Hamdane, Chawki Mahboub, Abdelhafid Moummi, Numerical approach to predict the outlet temperature of earth-to-air-heat-exchanger, Thermal Science and Engineering Progress, Volume 21, 2021, 100806, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2020.100806.

[20] Łukasz Amanowicz, Janusz Wojtkowiak, Validation of CFD model for simulation of multi-pipe earth-to-air heat exchangers (EAHEs) flow performance, Thermal Science and Engineering Progress, Volume 5, 2018, Pages 44-49, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2017.10.018.

[21] Montaser Mahmoud, Mohamad Ramadan, Sumsun Naher, Keith Pullen, Abdul-Ghani Olabi, Advances in Grout Materials in Borehole Heat Exchangers, Reference Module in Materials Science and Materials Engineering, Elsevier, 2021, ISBN 9780128035818, https://doi.org/10.1016/B978-0-12-815732-9.00053-X.

[22] Montaser Mahmoud, Mohamad Ramadan, Keith Pullen, Mohammad Ali Abdelkareem, Tabbi Wilberforce, Abdul-Ghani Olabi, Sumsun Naher, A review of grout materials in geothermal energy applications, International Journal of Thermofluids, Volume 10, 2021, 100070, ISSN 2666-2027, https://doi.org/10.1016/j.ijft.2021.100070.

[23] Aiqiang Pan, John S. McCartney, Lin Lu, Tian You, A novel analytical multilayer cylindrical heat source model for vertical ground heat exchangers installed in layered ground, Energy, Volume 200, 2020, 117545, ISSN 0360-5442, https://doi.org/10.1016/j.energy.2020.117545.

[24] Guohui Gan, A numerical methodology for comprehensive assessment of the dynamic thermal performance of horizontal ground heat exchangers, Thermal Science and Engineering Progress, Volume 11, 2019, Pages 365-379, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2019.04.013.

[25] Yajiao Liu, Guangqin Huang, Jun Lu, Xiaofeng Yang, Chunlong Zhuang, Jiao Qin, A novel 2-D ring-tubes model and numerical investigation of heat and moisture transfer around helix ground heat exchanger, Geothermics, Volume 85, 2020, 101789, ISSN 0375-6505, https://doi.org/10.1016/j.geothermics.2019.101789.

[26] Hikmet Esen, Mehmet Esen and Onur Ozsolak, Modelling and experimental performance analysis of solar-assisted ground source heat pump system, Journal of Experimental & Theoretical Artificial Intelligence, Volume 29, 2015, Pages 1-17, Issue 1, https://doi:10.1080/0952813X.2015.1056242.

[27] Sarah Van Erdeweghe, Johan Van Bael, William D'haeseleer, Optimal configuration, design and control of a binary geothermal combined heat-and-power plant, Energy Conversion and Management, Volume 198, 2019, 111854, ISSN 0196-8904, https://doi.org/10.1016/j.enconman.2019.111854.

[28] Xuan Guan, Hualing Zhang, Chunyang Xue, A method of selecting cold and heat sources for enterprises in an industrial park with combined cooling, heating, and power, Journal of Cleaner Production, Volume 190, 2018, Pages 608-617, ISSN 0959-6526, https://doi.org/10.1016/j.jclepro.2018.04.099.

[29] Nattaporn Chaiyat, A multigeneration system of combined cooling, heating, and power (CCHP) for low-temperature geothermal system by using air cooling, Thermal Science and Engineering Progress, Volume 21, 2021, 100786, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2020.100786.

[30] M. Fallah, R. Akbarpour Ghiasi, N. Hasani Mokarram, A comprehensive comparison among different types of geothermal plants from exergy and thermoeconomic points of view, Thermal Science and Engineering Progress, Volume 5, 2018, Pages 15-24, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2017.10.017.

[31] Muhammad Imran, Hafiz Ali Muhammad, Farooq Sher, Muhammad Farooq, Young-Jin Baik, Zabdur Rehman, Chapter 19 - Exergoeconomic optimization of a binary geothermal power plant, Editor(s): Can Ozgur Colpan, Mehmet Akif Ezan, Onder Kizilkan, Thermodynamic Analysis and Optimization of Geothermal Power Plants, Elsevier, 2021, Pages 315-326, ISBN 9780128210376, https://doi.org/10.1016/B978-0-12-821037-6.00001-9.

[32] A. Ahmadi, M. El Haj Assad, D.H. Jamali, R. Kumar, Z.X. Li, T. Salameh, M. Al-Shabi, M.A. Ehyaei, Applications of geothermal organic Rankine Cycle for electricity production, Journal of Cleaner Production, Volume 274, 2020, 122950, ISSN 0959-6526, https://doi.org/10.1016/j.jclepro.2020.122950.

[33] Fereshteh Samadi, Neda Kazemi, Exergoeconomic analysis of zeotropic mixture on the new proposed organic Rankine cycle for energy production from geothermal

resources, Renewable Energy, Volume 152, 2020, Pages 1250-1265, ISSN 0960-1481, https://doi.org/10.1016/j.renene.2020.01.038.

[34] Lukman Adi Prananto, Firman Bagja Juangsa, Riza Mochamad Iqbal, Muhammad Aziz, Tubagus Ahmad Fauzi Soelaiman, Dry steam cycle application for excess steam utilization: Kamojang geothermal power plant case study, Renewable Energy, Volume 117, 2018, Pages 157-165, ISSN 0960-1481, https://doi.org/10.1016/j.renene.2017.10.029.

[35] Yusuf Başoğul, Environmental assessment of a binary geothermal sourced power plant accompanied by exergy analysis, Energy Conversion and Management, Volume 195, 2019, Pages 492-501, ISSN 0196-8904, https://doi.org/10.1016/j.enconman.2019.05.033.

[36] N. Nasruddin, Irfan Dwi Saputra, Tania Mentari, André Bardow, Olga Marcelina, Susanto Berlin, Exergy, exergoeconomic, and exergoenvironmental optimization of the geothermal binary cycle power plant at Ampallas, West Sulawesi, Indonesia, Thermal Science and Engineering Progress, Volume 19, 2020, 100625, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2020.100625.

[37] Nan Wang, Su Zhang, Zhibin Fei, Wei Zhang, Limin Shao, Farshid Sardari, Thermodynamic performance analysis a power and cooling generation system based on geothermal flash, organic Rankine cycles, and ejector refrigeration cycle; application of zeotropic mixtures, Sustainable Energy Technologies and Assessments, Volume 40, 2020, 100749, ISSN 2213-1388, https://doi.org/10.1016/j.seta.2020.100749.

[38] Milad Feili, Hadi Rostamzadeh, Hadi Ghaebi, A new high-efficient cooling/power cogeneration system based on a double-flash geothermal power plant and a novel zeotropic bi-evaporator ejector refrigeration cycle, Renewable Energy, Volume 162, 2020, Pages 2126-2152, ISSN 0960-1481, https://doi.org/10.1016/j.renene.2020.10.011.

[39] N. Hassani Mokarram, A.H. Mosaffa, Investigation of the thermoeconomic improvement of integrating enhanced geothermal single flash with transcritical organic Rankine cycle, Energy Conversion and Management, Volume 213, 2020, 112831, ISSN 0196-8904, https://doi.org/10.1016/j.enconman.2020.112831.

[40] Man-Wen Tian, Towhid Parikhani, Kittisak Jermsittiparsert, Muhammad Aqeel Ashraf, Exergoeconomic optimization of a new double-flash geothermal-based combined cooling and power (CCP) system at two different cooling temperatures assisted by boosters, Journal of Cleaner Production, Volume 261, 2020, 120921, ISSN 0959-6526, https://doi.org/10.1016/j.jclepro.2020.120921.

[41] Mehran Abdolalipouradl, Farzad Mohammadkhani, Shahram Khalilarya, A comparative analysis of novel combined flash-binary cycles for Sabalan geothermal wells: Thermodynamic and exergoeconomic viewpoints, Energy, Volume 209, 2020, 118235, ISSN 0360-5442, https://doi.org/10.1016/j.energy.2020.118235.

[42] Sebastian Eyerer, Fabian Dawo, Christoph Wieland, Hartmut Spliethoff, Advanced ORC architecture for geothermal combined heat and power generation, Energy, Volume 205, 2020, 117967, ISSN 0360-5442, https://doi.org/10.1016/j.energy.2020.117967.

[43] Montaser Mahmoud, Mohamad Ramadan, Sumsun Naher, Keith Pullen, Mohammad Ali Abdelkareem, Abdul-Ghani Olabi, A review of geothermal energy-driven hydrogen production systems, Thermal Science and Engineering Progress, Volume 22, 2021, 100854, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2021.100854.

[44] Xinli Lu, Yangyang Zhao, Jialing Zhu, Wei Zhang, Optimization and applicability of compound power cycles for enhanced geothermal systems, Applied Energy, Volume 229, 2018, Pages 128-141, ISSN 0306-2619, https://doi.org/10.1016/j.apenergy.2018.07.033.

[45] Abdul Ghani Olabi, Montaser Mahmoud, Bassel Soudan, Tabbi Wilberforce, Mohamad Ramadan, Geothermal based hybrid energy systems, toward eco-friendly energy approaches, Renewable Energy, Volume 147, Part 1, 2020, Pages 2003-2012, ISSN 0960-1481, https://doi.org/10.1016/j.renene.2019.09.140.

[46] Chen Yue, Le Tong, Shizhong Zhang, Thermal and economic analysis on vehicle energy supplying system based on waste heat recovery organic Rankine cycle, Applied Energy, Volume 248, 2019, Pages 241-255, ISSN 0306-2619, https://doi.org/10.1016/j.apenergy.2019.04.081.

[47] A. Khosravi, S. Syri, X. Zhao, M.E.H. Assad, An artificial intelligence approach for thermodynamic modeling of geothermal based-organic Rankine cycle equipped with solar system, Geothermics, Volume 80, 2019, Pages 138-154, ISSN 0375-6505, https://doi.org/10.1016/j.geothermics.2019.03.003.

[48] Farzad Mohammadkhani, Mortaza Yari, A 0D model for diesel engine simulation and employing a transcritical dual loop Organic Rankine Cycle (ORC) for waste heat recovery from its exhaust and coolant: Thermodynamic and economic analysis, Applied Thermal Engineering, Volume 150, 2019, Pages 329-347, ISSN 1359-4311, https://doi.org/10.1016/j.applthermaleng.2018.12.158.

[49] Liang-Hui Zhi, Peng Hu, Long-Xiang Chen, Gang Zhao, Thermodynamic analysis of an innovative transcritical CO2 parallel Rankine cycle driven by engine waste heat

and liquefied natural gas cold, Energy Conversion and Management, Volume 209, 2020, 112583, ISSN 0196-8904, https://doi.org/10.1016/j.enconman.2020.112583.

[50] Xingchao Wang, Edward K. Levy, Chunjian Pan, Carlos E. Romero, Arindam Banerjee, Carlos Rubio-Maya, Lehua Pan, Working fluid selection for organic Rankine cycle power generation using hot produced supercritical CO2 from a geothermal reservoir, Applied Thermal Engineering, Volume 149, 2019, Pages 1287-1304, ISSN 1359-4311, https://doi.org/10.1016/j.applthermaleng.2018.12.112.

[51] Jian Song, Xiaoya Li, Kai Wang, Christos N. Markides, Parametric optimisation of a combined supercritical CO2 (S-CO2) cycle and organic Rankine cycle (ORC) system for internal combustion engine (ICE) waste-heat recovery, Energy Conversion and Management, Volume 218, 2020, 112999, ISSN 0196-8904, https://doi.org/10.1016/j.enconman.2020.112999.

[52] Xurong Wang, Yiping Dai, Exergoeconomic analysis of utilizing the transcritical CO2 cycle and the ORC for a recompression supercritical CO2 cycle waste heat recovery: A comparative study, Applied Energy, Volume 170, 2016, Pages 193-207, ISSN 0306-2619, https://doi.org/10.1016/j.apenergy.2016.02.112.

[53] Kerri L. Hickenbottom, Leslie Miller-Robbie, Johan Vanneste, Junko Munakata Marr, Michael B. Heeley, Tzahi Y. Cath, Comparative life-cycle assessment of a novel osmotic heat engine and an organic Rankine cycle for energy production from low-grade heat, Journal of Cleaner Production, Volume 191, 2018, Pages 490-501, ISSN 0959-6526, https://doi.org/10.1016/j.jclepro.2018.04.106.

[54] Matteo Marchionni, Giuseppe Bianchi, Savvas A. Tassou, Techno-economic assessment of Joule-Brayton cycle architectures for heat to power conversion from high-grade heat sources using CO2 in the supercritical state, Energy, Volume 148, 2018, Pages 1140-1152, ISSN 0360-5442, https://doi.org/10.1016/j.energy.2018.02.005.

[55] Gequn Shu, Lingfeng Shi, Hua Tian, Xiaoya Li, Guangdai Huang, Liwen Chang, An improved CO2-based transcritical Rankine cycle (CTRC) used for engine waste heat recovery, Applied Energy, Volume 176, 2016, Pages 171-182, ISSN 0306-2619, https://doi.org/10.1016/j.apenergy.2016.05.053.

[56] Hussam Jouhara, Navid Khordehgah, Sulaiman Almahmoud, Bertrand Delpech, Amisha Chauhan, Savvas A. Tassou, Waste heat recovery technologies and applications, Thermal Science and Engineering Progress, Volume 6, 2018, Pages 268-289, ISSN 2451-9049, https://doi.org/10.1016/j.tsep.2018.04.017.

[57] José J. Fierro, Ana Escudero-Atehortua, César Nieto-Londoño, Mauricio Giraldo, Hussam Jouhara, Luiz C. Wrobel, Evaluation of waste heat recovery technologies for

the cement industry, International Journal of Thermofluids, Volumes 7–8, 2020, 100040, ISSN 2666-2027, https://doi.org/10.1016/j.ijft.2020.100040.

[58] Frank P. Incropera; David P. Dewitt; Theodore L. Bergman; Adrienne S. Lavine, Fundamentals of Heat and Mass Transfer, 8e, New York: Wiley, 2017.

[59] Ehsan Izadpanah, Ahmad Zarei, Saeed Akhavan, Marzie Babaie Rabiee, An experimental investigation of natural convection heat transfer from a helically coiled heat exchanger, International Journal of Refrigeration, Volume 93, 2018, Pages 38-46, ISSN 0140-7007, https://doi.org/10.1016/j.ijrefrig.2018.06.008.

[60] Gequn Shu, Lingfeng Shi, Hua Tian, Shuai Deng, Xiaoya Li, Liwen Chang, Configurations selection maps of CO2-based transcritical Rankine cycle (CTRC) for thermal energy management of engine waste heat, Applied Energy, Volume 186, Part 3, 2017, Pages 423-435, ISSN 0306-2619, https://doi.org/10.1016/j.apenergy.2016.03.049.