

Article

Cooling Potential for Single and Advanced Absorption Cooling Systems in a Geothermal Field in Mexico

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Abstract: Climate change is one of the main problems humanity is currently facing due to the use of fossil fuels. At present, 20% of the total electricity consumed in buildings worldwide is for air conditioning. The development and use of thermally driven cooling systems is very important, since they can be activated by renewable energies, such as geothermal, reducing the consumption of electricity produced by fossil fuels. In this paper, we analyze a geothermal field located in the state of Jalisco, Mexico, with the aim of comparing the performance of different advanced absorption cooling systems driven by a geothermal heat source. The analysis includes the influence of water temperature obtained from an abandoned geothermal well, using a U tube heat exchanger inside the well. The results show that this well can reach temperatures from 59 °C to 190 °C, depending on the depth of the U tube and the insulation thickness. At a $T_E = 8$ °C, the operating range temperatures were 59–80 °C, 77–110 °C, 135–162 °C, and 180–187 °C for the half-effect, single-effect, double-effect and triple-effect systems, respectively. The maximum cooling potential was 99,334 GW obtained with the double-effect system, followed by 92,995 GW with the triple-effect system, 70,939 GW with the single-effect system, and 38,721 GW with the half-effect system.

Keywords: absorption cooling systems; geothermal cooling; cooling potential



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1. Introduction

Climate change is one of the main problems humanity is currently facing due to the use of fossil fuels for different applications. Currently, 20% of the total electricity consumed in buildings worldwide is for air conditioning [1]. In 2018 it was estimated that refrigeration systems consumed 3900 TWh/year of electricity worldwide, and this energy is mainly produced by the consumption of fossil fuels [2]. Therefore, it is important that alternative cooling systems capable of operating with renewable energies, such as geothermal, are used.

A geothermal resource is the portion of heat released from the interior of the Earth that can be used in the appropriate technical and economic conditions available [3]. The potential of the Earth's geothermal resources is enormous, considering their current use and prospects, given the energy needs of humanity. The total heat content of the Earth is estimated to be around 1013 EJ (1 EJ = 10¹⁸ J) and it would take more than 109 years to deplete it, through a current global terrestrial heat flux of 40 million MW [4]. Therefore, the geothermal resource base is large enough and is practically everywhere. In 2020, the total global electricity production was 26,000 TWh, of which geothermal energy supplied an estimated 225 TWh (97 TWh of electricity and the rest in the form of heat) [5]. These amounts represent a low contribution to the total energy consumed around the world, even with the known potential.

The distribution of thermal energy used by category is approximately 58.8% for geothermal heat pumps, 18.0% for bathing and swimming, 16.0% for space heating, 3.5% for greenhouse heating, 1.6% for industrial applications, 1.3% for aquaculture pond and raceway heating, 0.8% for other applications [6].

In Mexico, the indirect use of geothermal energy almost entirely encompasses electricity production. Its direct uses are still restricted to bathing and swimming in recreational facilities and some of them for therapeutic uses (reported in 20 locations), including the geothermal field studied in this work, La Primavera. Likewise, Mexico's Federal Electricity Commission developed some of the direct uses of geothermal resources in the Los Azufre field, including a wood dryer, a fruit and vegetable dehydrator, a greenhouse, and a heating system of offices [7]. National studies related to geothermal energy and absorption cooling systems are presented next. Galindo-Luna et al. [8] proposed a hybrid system using a parabolic-trough collector field coupled to a low-enthalpy geothermal well to drive an absorption air conditioning system. A coefficient of performance (COP) of 0.71 was obtained for a generator temperature of 90 °C. The results of the modeling of a half-effect absorption system were reported using an ammonia/lithium nitrate mixture driven by a low-enthalpy geothermal source from two geothermal wells [9]. The results showed that at the wells temperatures, the cooling system can operate but obtaining low cooling effects. The potential that can be obtained from a geothermal well to operate a single-effect system was presented considering seasonal variations [10]. The COP values varied between 0.91 and 0.97. Ambriz-Díaz et al. [11] analyzed a cascade hybrid system operating in different modes. The system was composed in the first thermal level by an organic Rankine cycle to produce electricity, in the second level by an absorption refrigeration cycle to produce ice, and on the third level by a dehydrator for drying agricultural products. The results indicated that the dehydration process significantly improved the economic benefits of all the alternatives, achieving payback periods of around one year and reducing greenhouse emissions. Also, it was reported that the production of electricity alone was undesirable because it had the worst energy efficiencies and payback periods.

At an international level, several studies have been realized using geothermal energy to drive cooling systems or hybrid systems to produce cooling and an extra output. These studies are mainly based on single-effect, half-effect, and double-effect systems. Rogowska et al. [12] modeled a 10 kW single-stage absorption cooling system to produce air conditioning. The results were applied to a more advanced design of 500 kW refrigeration units in the studied region. An absorption refrigeration unit for the storage of agricultural products driven by geothermal hot water [13] reported a COP of 0.49, reaching a maximum value of 0.6 by using an extra heat exchanger. A dynamic simulation study to evaluate the performance of a new heating and cooling system was made based on the coupling between a low or medium-enthalpy geothermal source, an air handling unit, and a desiccant wheel [14]. The analysis demonstrated the technical and economic feasibility of the proposed system. The performance of an absorption refrigeration cycle and an ejector cycle using data from a low-enthalpy geothermal well were compared [15], deducing that at specific conditions, both systems were feasible as alternatives to conventional refrigeration systems. Arreola Núñez [16] modeled a commercial absorption machine driven by geothermal energy instead of gas. The results showed the thermodynamic feasibility of the system operating with geothermal energy but obtaining slightly lower coefficients of performance. Two works [17,18] have performed a parametric exergy and economic analysis of a combined cogeneration system to produce cooling and power using geothermal energy as a heat source. The results determined the most optimal operating conditions of the proposed systems. An integrated power and absorption cooling system was modeled including the waste heat from the power generation process as heat input in the cooling system, which in turn supplied the cooling water to the power plant [19]. It was concluded that it was not feasible, since it required a larger cooling capacity, increasing the total investment costs. Regarding advanced absorption cooling systems, a study presented a performance comparison of four different configurations for water/LiBr absorption cooling systems,

being these a single-effect, a half-effect, a double-effect in series, and a double-effect in reverse [20]. The maximum COPs achieved were 0.89, 0.4, 1.5, and 1.48, respectively. Similarly, calculations of the COPs and exergy analysis for single, double, triple, and half-effect water/LiBr absorption cycles were performed in another work [21]. It was observed that the COP significantly increased with the double and triple-effect cycles with values of 1.65 and 2.32, respectively. In the last two studies, the authors did not consider a special heat source, they only analyzed the performance of the systems at different operating temperatures. Shirazi et al. [22] analyzed the feasibility of single, double, and triple-effect heating and cooling absorption systems operating with solar collectors. The most favorable results were obtained with the double-effect system. The influence of various operating parameters on the COP and exergy efficiencies of the system were evaluated in [23]. The COPs for the single, double, and triple-effect chillers were in the range of 0.73–0.79, 1.22–1.42, and 1.62–1.90, respectively, while the maximum exergy efficiencies were in the range of 12.5–23.2%, 14.3–25.1%, and 17.7–25.2%, respectively. Best and Rivera [24] carried out a review analyzing both theoretical and experimental studies on absorption refrigeration systems operated with renewable energies such as geothermal. It was observed that the studies on absorption cooling systems driven partially or totally by geothermal energy in Mexico were very scarce. The installation of a thermo-chiller was described [25] operating with a double-effect ammonia absorption cycle at the Aurora Ice Museum in Chena Hot Springs, Alaska. The thermo-chiller was powered by thermal spring water providing 52.8 kW of cooling at $-29\text{ }^{\circ}\text{C}$. Han et al. [26], proposed a double-effect water/LiBr absorption cooling system, based on an enhanced geothermal system (EGS) using concentric circle wells. The influence of key parameters such as well depth and injection rates in the cooling system were analyzed. The results showed that the driven temperature of EGS hot water can reach more than $150\text{ }^{\circ}\text{C}$ steadily for 20 years. A thermodynamic analysis of a parallel-flow water/LiBr double-effect absorption system powered by geothermal energy was performed by using the Engineering Equation Solver (EES) software [27]. The results showed the behavior of the chiller at different operating conditions. A COP of up to 1.43 and a cooling load of 420 kW were obtained. Another thermodynamic study to utilize an existing low-temperature geothermal heat source was presented including six different models, with simple and half-effect systems [28]. The COP obtained with the half-effect system was approximately half (0.424) of the COP obtained with single-effect chillers (0.825). The results showed that the geothermal heat source can be used to drive both single and half-effect systems.

This bibliographic review showed that several studies have been carried out to compare the performance of advanced systems (half, double, and triple-effect systems) operating with solar energy. However, studies related to advanced absorption systems directly operating with a geothermal heat source are very scarce. Specifically, no studies were found for double and triple-effect systems driven by geothermal energy in which the geothermal well was analyzed.

The purpose of this article is to compare different configurations of absorption cooling systems driven by a geothermal field located in Mexico. The analysis includes the temperatures that can be obtained from a geothermal well considering different cases, and the variation of the cooling potential and COP for each one of the systems over the course of the year as a function of the driven and ambient temperatures. Real ambient temperatures were used in the analysis obtained from a meteorological station installed in the geothermal field.

2. Systems Description

2.1. Single-Effect Cooling System

A single-effect water/LiBr refrigeration system is mainly integrated by two circuits: a refrigerant circuit formed by a condenser, an expansion valve, and an evaporator, and a solution circuit integrated by a generator, an absorber, a solution heat exchanger, a pump, and a valve. Figure 1 shows a schematic diagram of the system.

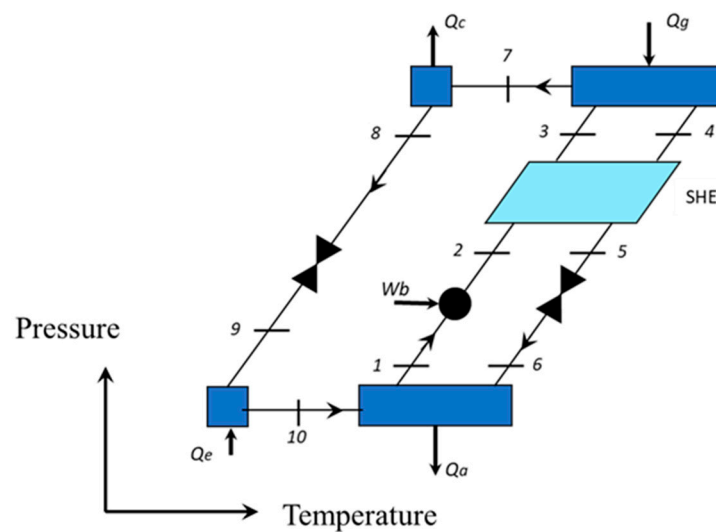


Figure 1. Schematic diagram of a single-effect absorption cooling system.

A solution with a high amount of refrigerant leaving the absorber (1) is pumped (2) to the generator, passing first through the solution heat exchanger, where it is preheated before entering the generator (3). In this component, an amount of heat Q_g is supplied to separate part of the water from the water/LiBr solution. The water in a vapor phase (7) goes to the condenser where it is condensed by means of an amount of heat removed Q_c . The liquid leaving this component (8) passes through an expansion valve reducing its pressure and temperature (9) and then to the evaporator producing the cooling effect Q_e . The water in a vapor phase leaving the evaporator (10) goes to the absorber where it is absorbed by the solution coming from the generator (4). The solution leaving the generator passes through the solution heat exchanger (5), preheating the solution going to the generator, and then through the valve reducing its pressure before entering the absorber (6). In the absorber, an amount of heat Q_a is delivered due to the exothermic reaction from the water absorption process. The solution with a high amount of refrigerant formed is then ready to be pumped to the generator starting the cycle again.

2.2. Half-Effect Cooling System

As can be seen in Figure 2, the half-effect cycle has one refrigerant circuit and two water/LiBr solution circuits, one of them at high pressure and the other one at low pressure. The refrigerant produced in the high-pressure generator passes through the condenser, the expansion valve, the evaporator, and the absorber similarly to the single-effect system. The solution low-pressure circuit also operates in the same way as previously described but with the difference that the refrigerant produced in the low-pressure generator is absorbed in the high-pressure absorber to produce a solution with a high amount of refrigerant which is used in the high-pressure circuit. Because in this system the heat is supplied into the two generators and produces only one stream of refrigerant which passes to the evaporator to produce the cooling effect, this system is called half-effect.

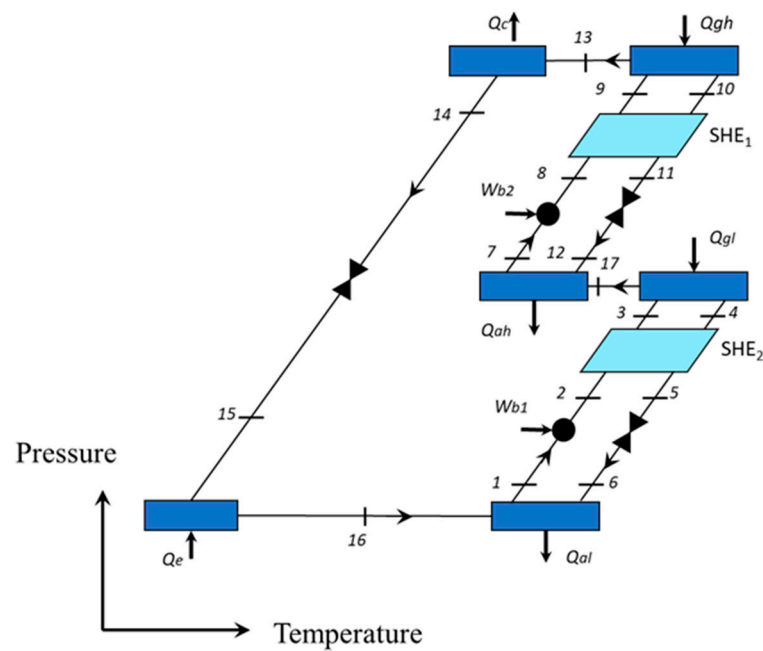


Figure 2. Schematic diagram of a half-effect absorption cooling system.

2.3. Double-Effect Cooling System

A double-effect system is shown in Figure 3. In the low part of the schematic diagram, it can be observed a single-stage system as described in Figure 1 but additionally, it has a second generator (Q_{g2}) operating at higher pressure and temperature, and a second solution heat exchanger (SHE_2). Although a second condenser appears in the diagram (Q_{c2}), this component and Q_{g1} is the same component. The shell part of a heat exchanger operates as a generator, while the side tubes operates as a condenser. As will be seen, the objective of adding the components above mentioned is to improve the system COP.

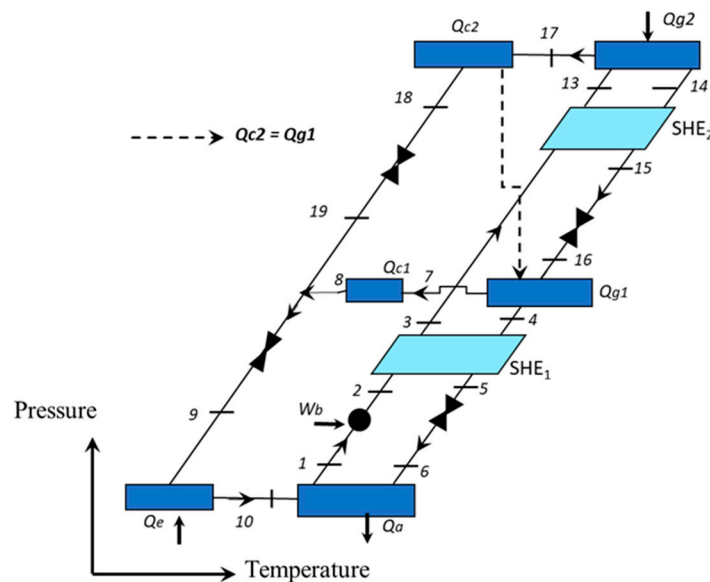


Figure 3. Schematic diagram for a double-effect absorption cooling system.

In this system, an amount of heat is supplied to Q_{g2} at a higher temperature than that supplied to a single-stage system to produce the refrigerant. The water in a vapor phase is then condensed in Q_{c2} leaving as a saturated liquid. The heat removed from the condensation processes is used as heat input to Q_{g1} to produce an extra amount of

refrigerant which is then condensed in Q_{c1} . Both condensed streams join before passing through the expansion valve and the evaporator. The rest of the cycle operates similarly to the single-stage system with the difference that the solution leaving from the absorber is pumped to Q_{g2} to produce the refrigerant at the highest pressure, and the solution leaving this component passes to Q_{g1} to produce more refrigerant. The two solution heat exchangers SHE_1 and SHE_2 are used to preheat the solution going from the absorber to the generator to reduce the heat supplied to Q_{g2} .

As can be seen from Figure 3, and the above explained, the heat is supplied to this system only in Q_{g2} , since Q_{g1} uses the heat delivered from the condensation process in Q_{c2} . Because the heat is supplied in only one component and produces two refrigerant streams, this system is denominated as a double-effect.

2.4. Triple-Effect Cooling System

A triple-effect system is shown in Figure 4. This system is similar to the double-effect system previously described but additionally, it has a third generator (Q_{g3}) and a third solution heat exchanger (SHE_3). The operation of this system is analogous to the double-effect system but in this case, the heat source is supplied to Q_{g3} at an even higher temperature than the second-stage system to produce the first stream of refrigerant. The other two streams of refrigerants are produced in Q_{g2} and Q_{g1} , using the heat delivered from the condensation processes in Q_{c3} and Q_{c2} . Because this system has the capability of producing three streams of refrigerant by supplying heat in only one component, this system is called a triple-effect system.

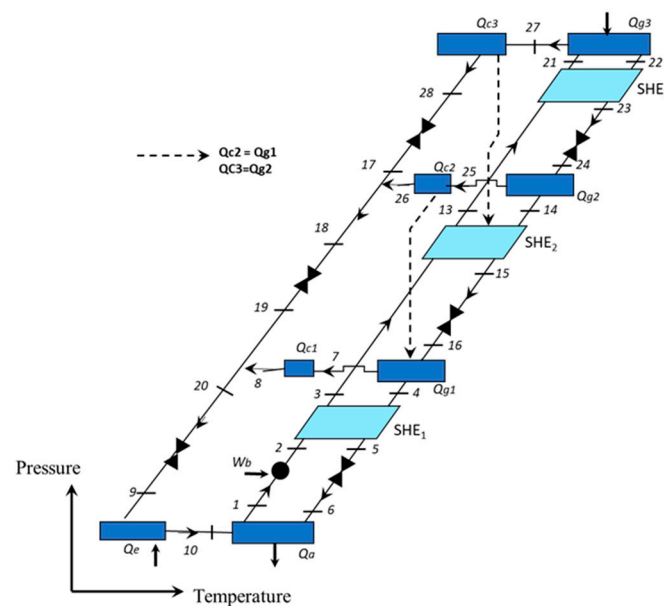


Figure 4. Schematic diagram of a triple-effect absorption cooling system.

3. Location and Solution Methods

3.1. Geothermal Source

The “Cerritos Colorados” geothermal field is located in the central-southern portion of the “La Primavera forest” [29] which is located 20 km west of the city of Guadalajara, in the state of Jalisco, between the extreme coordinates $103^{\circ}28'$ to $103^{\circ}42'$ West longitude and $20^{\circ}32'$ to $20^{\circ}44'$ North latitude (Figure 5).

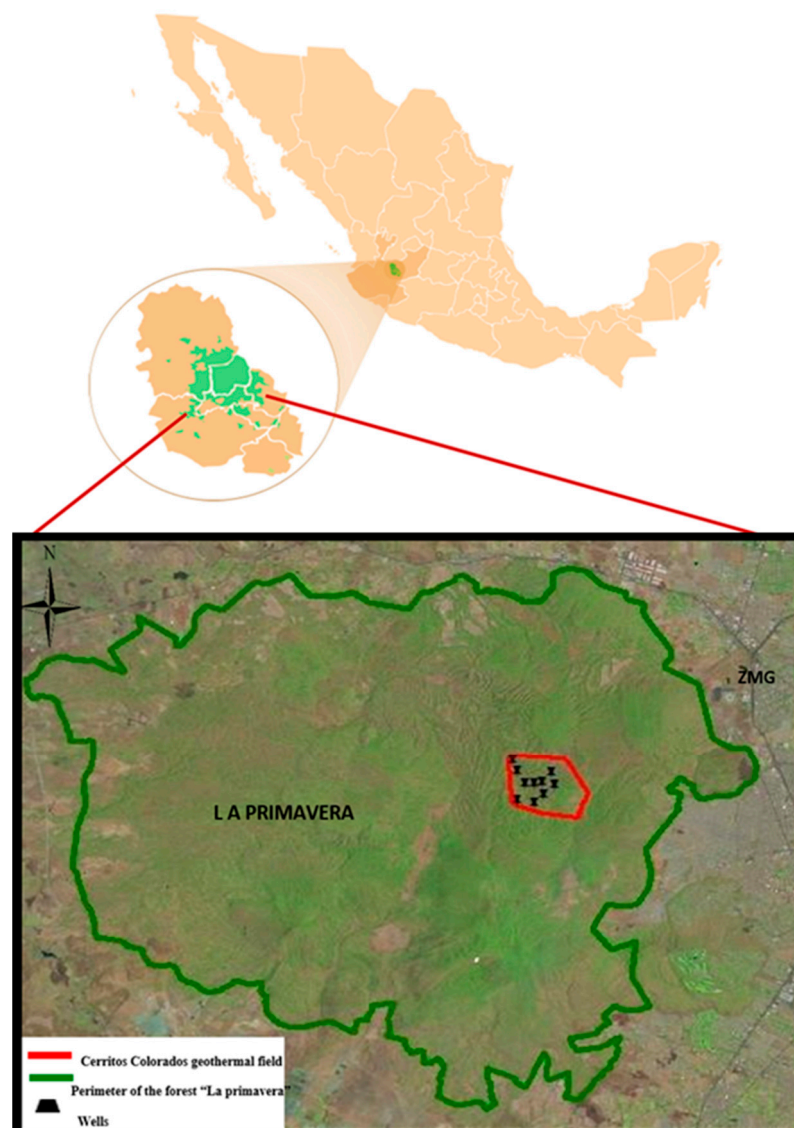


Figure 5. Location map of the Cerritos Colorado geothermal field [30].

In the present study, the data of the PR2 well are used. This well was chosen due to all the necessary data for analysis were available [31]. In Table 1 the characteristics of the well are presented, and Figure 6 presents its lithology.

Table 1. Data from PR2 geothermal well in the geothermal field Cerritos Colorados [29,31].

Well	Latitude	Longitude	Geothermal Gradient ($^{\circ}\text{C}/\text{km}$)	Depth (m)	Temperature ($^{\circ}\text{C}$)
PR2	20.66423	−103.53115	114.9	1988	320

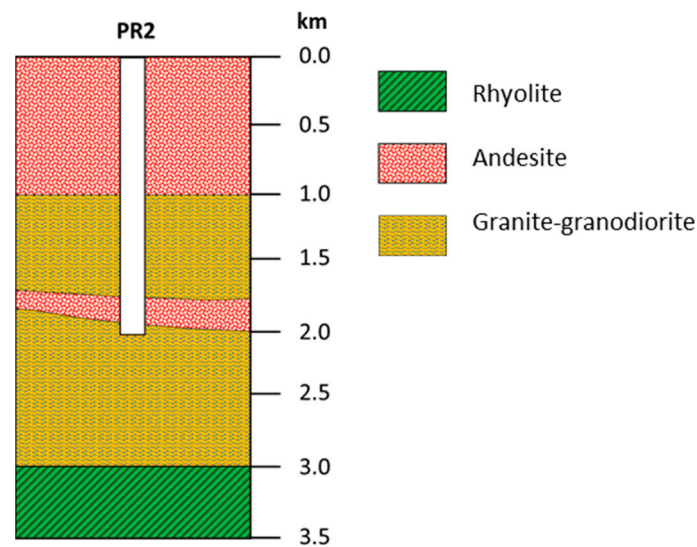


Figure 6. Simplified lithology based on core drilled wells from Cerritos Colorados geothermal field.

3.2. Calculation of Total Thermal Energy

To determine the energy gained by the fluid circulating through the well, the volumetric method was used. This method takes into consideration the energy contained in a certain volume of rock containing a hot water reservoir which supplies the energy to the fluid circulating through the pipe. The energy gained by the fluid is determined by using heat transfer equations and energy balances [32]. The procedure is repeated again for each control volume.

The energy stored in the rock volume (Q_r) can be determined by

$$Q_r = AH(1 - \phi)\rho_r C_r (T_i - T_m) \quad (1)$$

where A is the area of the control volume; H is the thickness of the reservoir; ρ_r is the density of the rock; C_r the specific heat of the rock; ϕ the porosity of the rock, defined as the relationship between the volume of empty spaces in the host rock or spaces occupied by the water in it and the total volume of the material [33]; and T_i and T_m , the initial and minimum temperatures, respectively.

The fluid energy Q_f can be estimated by Equation (2)

$$Q_f = AH\rho_f C_f \phi (T_i - T_m) \quad (2)$$

where ρ_f and C_f represent the density and specific heat of the rock, respectively.

Thus, the total energy is calculated as:

$$Q_t = Q_f + Q_r \quad (3)$$

Some of these parameters can be known with acceptable accuracy, but others, such as the area and thickness of the well, are uncertain even with extensive drilling, due to underground phenomena and morphology. Therefore, study areas of 1 km², 2 km², and 3 km², were considered as a minimum, probable, and maximum areas, respectively. For the majority of the reservoirs, the uncertainties regarding the depth are small compared with those of the respective area [34].

In Equations (1) and (2) H was considered as 2000 m which is an average value of divers well of the geothermal field. ϕ was established as 10%, which is related to the type of rock which is predominantly constituted by a volcanic igneous rock of intermediate composition [35]. The values of ρ_r and C_r of each type of rock were used according to the stratigraphic column of the Cerritos Colorados geothermal field (Figure 6). In the case of the fluid, the specific heat water values $C_f = 4.18$ kJ/kg°C and a density $\rho_f = 1000$ kg/m³

are used. T_m is set according to the technology used. For geothermal power plants, a temperature of 135 °C for single cycles and 90 °C for binary cycles are assumed in general [36]. For the present study, T_m varied according to the requirements of each absorption cycle. From the modeling of each one of the systems, it was found that for the specified conditions, T_m was 58 °C, 77 °C, 134 °C and 170 °C for the half, single, double, and triple-effect systems, respectively, and T_i was taken from the recorded temperatures of the analyzed geothermal well. It is important to mention that although a T_m of 58 °C for the half-effect cycle seemed to be a low temperature to drive an absorption cooling system, the double solution circuit used in this cycle (see Figure 2), allows the system to operate at such low temperature.

To define the reservoir initial temperature (T_i), an average of the temperatures recorded in the well PR2 was taken based on the data in Table 1.

3.3. Calculation of the Cooling Potential (CP) for the Absorption Cooling Systems

The power (P) produced by a geothermal power plant can be calculated by Equation (4) using the volumetric method [36].

$$P = \frac{Q_t R_f C_e}{F_p t} \quad (4)$$

where Q_t is the total energy obtained by Equation (3), R_f is the recovery factor, which represents the fraction of the energy that could be recovered, C_e is the efficiency factor, F_p is the plant factor, and t the operation time in hours. Values of R_f up to 25% have been reported for good conditions of porosity and permeability into the well but normally lower values are reported [35]. For the present study, an R_f of 12.5% and an F_p of 90% were considered since they are typical values reported in the literature [36].

Based on this equation, the cooling potential (CP) can be estimated by Equation (5) as:

$$CP = \frac{Q_t R_f COP}{F_p t} \quad (5)$$

where the efficiency factor (C_e) is replaced by the COP for each one of the cooling analyzed systems.

3.4. Simulation of the Water/LiBr Absorption Cooling Systems

To obtain the COP of each system, mathematical models were developed based on the first-law of thermodynamics, and then the simulation was carried out by using the Engineering Equation Solver (EES) software.

The following assumptions were considered in the modeling of each one of the cooling cycles:

- There is thermodynamic equilibrium in the cycles.
- The cycles operate under steady-state conditions.
- The solution is saturated at the exit of the generators and absorbers.
- The refrigerant is saturated at the exit of the condensers and evaporators.
- Heat losses and pressure drops in piping and components are negligible.
- The flow through the valves is isenthalpic.
- The effectiveness of the heat exchangers was 0.8.
- A $\Delta T = 7$ °C was considered between the ambient temperature and the temperature of the condensers and absorbers.
- The evaporation temperature T_E was 8 °C.

The value of $T_E = 8$ °C was chosen since it was the optimum value of the evaporation temperature obtained from the analysis of diverse absorption cooling systems [10]. These authors also reported that the values of CP and COP are directly proportional to T_E . However, higher values of T_E were not considered in the present study, since higher temperatures cannot provide good air condition temperatures.

Appendix A shows the tables with the energy balances for each one of the components integrating the four analyzed systems [20].

3.5. Heat Transfer Analysis in a U-Tube Heat Exchanger

The heat transfer analysis inside the “U” type heat exchanger was performed using a mathematical model developed with the finite volume numerical method and implemented in Fortran 90 programming language. This computational mathematical method consists of dividing the study domain into a finite quantity of control volumes. The equation which models the physical phenomena is discretized and solved for each control volume [37]. The main advantage of this mathematical model is that allows the estimation of fluid temperature through the heat exchanger based on pipe diameter, fluid velocity, boundary temperatures, the total length of the heat exchanger, the total length of the insulator, and soil thermo-physical characteristics. The study domain includes the soil, the well, the heat exchanger walls, the insulator material, and the space inside the heat exchanger where the fluid flows. Figure 7 shows a schematic diagram of the underground U tube heat exchanger.

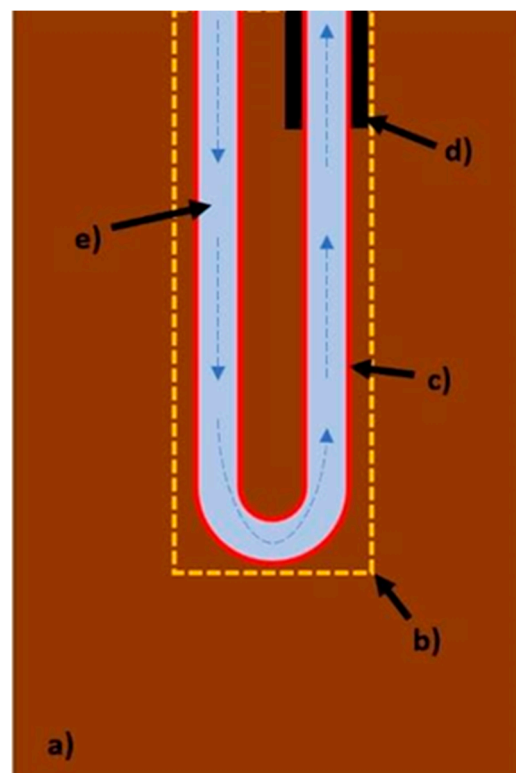


Figure 7. Study domain: (a) soil (brown), (b) well (yellow), (c) heat exchanger walls (red), (d) insulator (black), and (e) space inside the heat exchanger (blue).

In total, 1400 control volumes were used to represent the study domain. These control volumes are organized irregularly, presenting a greater density near the pipe walls of the heat exchanger.

The assumptions considered for the mathematical model are:

- The fluid viscosity is neglected, thus, the fluid velocity profile inside the heat exchanger is constant.
- The space inside the heat exchanger is represented as one-dimensional due to the constant fluid velocity.
- Inside the heat exchanger, the convection is the main heat transfer phenomenon; therefore, the conduction heat transfer among the fluid and the pipe walls is neglected.

- Outside the heat exchanger, the conduction is the main heat transfer phenomenon; therefore, the convection heat transfer through the soil porous is neglected.
- The heat exchanger walls thickness is neglected as the thermal dynamic is led by the soil.
- The soil thermal diffusivity changes as a function of the soil type presented in the lithology [32].
- The boundary temperature is equal to the temperature calculated with the geothermal gradient.
- The inlet fluid temperature remains constant.
- Environmental factors are neglected in soil surface temperature.
- Condensation and phase change for the fluid are omitted.

The energy equation (Equation (6)) is used for modeling purposes. All the terms in this equation (temporal, convective, and diffusive) are discretized for each node in the mesh represents the study domain [38].

$$\frac{\partial T}{\partial t} + \nabla \cdot (vT) = \alpha \cdot \nabla^2 T \quad (6)$$

In the previous equation, v and α represent the fluid velocity and thermal diffusivity, respectively. When Equation (6) is discretized, and the coefficients are arranged for each mesh node, a linear equation system results. This equations system is solved using an iterative Tridiagonal Matrix Algorithm (TDMA), which solves each dimension separately [39].

4. Results and Discussion

To determine the CP and COP for each of the systems driven with the energy provided by the geothermal well, first it is necessary to know the ambient temperatures in the zone since the geothermal well temperatures and the systems cooling performance depend upon these.

4.1. Ambient Temperature Data per Hour

The ambient temperature data used for the analysis were requested from the National Water Commission, CONAGUA. The data from the meteorological station “La Primavera” was used since it is the nearest station to the geothermal field.

Figure 8 shows the average ambient temperature per hour for the entire year 2018 for Spring, Summer, Autumn, and Winter. As can be seen, the temperatures varied between 10.3 °C and 33 °C for the whole year.

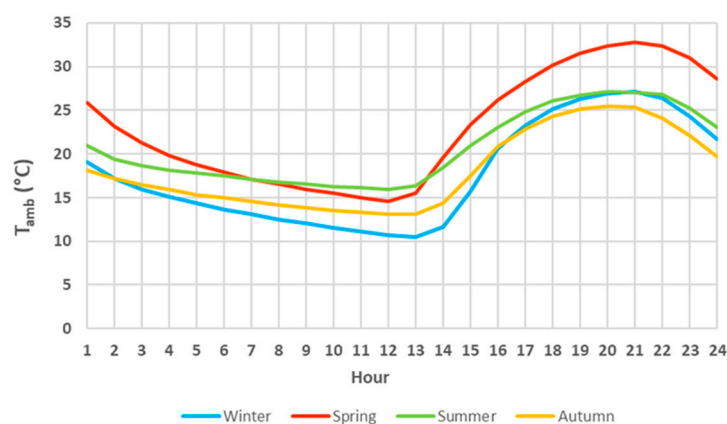


Figure 8. Average temperature per hour for each season.

4.2. Determination of the Geothermal Well Temperature

As stated in the assumptions, the initial temperature distribution is equal to the geothermal gradient for the specific site, which is equal to the boundary temperature.

Equations (7) and (8) show temperature correlations used in this study as a function of depth z .

$$T_{(z,t=0)} = 20 + (0.115 \cdot z), 0 < z < 2000 \quad (7)$$

$$T_{(z,t=0)} = 480 - (0.115 \cdot z), 2000 < z < 4000 \quad (8)$$

Six different cases were simulated changing the fluid velocity, the heat exchanger total length, the insulator length, and thickness. The thermal diffusivity for each type of soil used in the simulations was calculated using data from the stratigraphic column presented previously (Figure 6) and is constant for each test. The thermal diffusivity of the insulator material was considered equal to $1.08 \times 10^{-9} \text{ (m}^2/\text{s)}$. For all the tests carried out, the fluid inlet temperature was fixed at 20°C , which is the average value of the seasonal environmental temperature. Finally, a “U” tube heat exchanger with a 3” pipe diameter was considered for the 6 cases. This value is considered because this is the maximum allowable diameter to cover the largest contact area inside the well. Table 2 shows the values of total length (TL), insulator length (IL), insulator thickness (IT), and fluid velocity (v) for each one of the 6 tests. The outlet fluid temperature is also reported (T_O).

Table 2. Values considered for the simulations of the geothermal well.

Case	TL (m)	IL (m)	IT(in)	v (m/s)	T_O ($^\circ\text{C}$)	Cooling System Possible to Activate
1	4000	-	-	6	60.8	half-effect
2	4000	1000	1	6	174.7	triple-effect
3	4000	500	2	6	116.2	single-effect
4	4000	1000	1	4	160	double-effect
5	3000	1000	1	6	145.9	double-effect
6	3000	-	-	6	59.6	half-effect

The base case (first) was done without insulation, resulting in a temperature of 60.8°C . This outlet temperature is enough to be used in a half-effect cooling system and was set as the starting point to compare subsequent results. Afterward, distinct values of total length, insulator length, insulator thickness, and air velocity were used to analyze their effects on the outlet fluid temperature, which is used to drive the absorption systems.

For the second case, insulator material was considered with 1000 m length and 1” of thickness, keeping the remaining parameters as in the first case. The result shows that the fluid outlet temperature was 174.7°C .

The insulator material was considered again for the third case but changing the length and thickness. As result, the outlet fluid temperature was 116.2°C . Compared with the first one, simulating 500 m less insulator material, allows the fluid to continue exchanging energy with its surroundings.

The fourth case considers 1000 m of insulator material, 1” of insulator thickness but a fluid velocity of 4 m/s. The result shows that the change in the velocity caused less turbulence, but the outlet fluid temperature was greater compared with the first case achieving a temperature of 160°C .

Figure 9 illustrates the results obtained for cases 1 to 4. It shows the initial temperature condition in blue, equal to the temperature at the boundary obtained with the geothermal gradient. The fluid temperature profile is also shown, along with the heat exchanger for examples 1 to 4 (all 4000 m in total length). It is observed that the energy gain occurs in the first 2000 m, which is the downward section of the exchanger, while, depending on the simulator conditions, the energy gained was different.

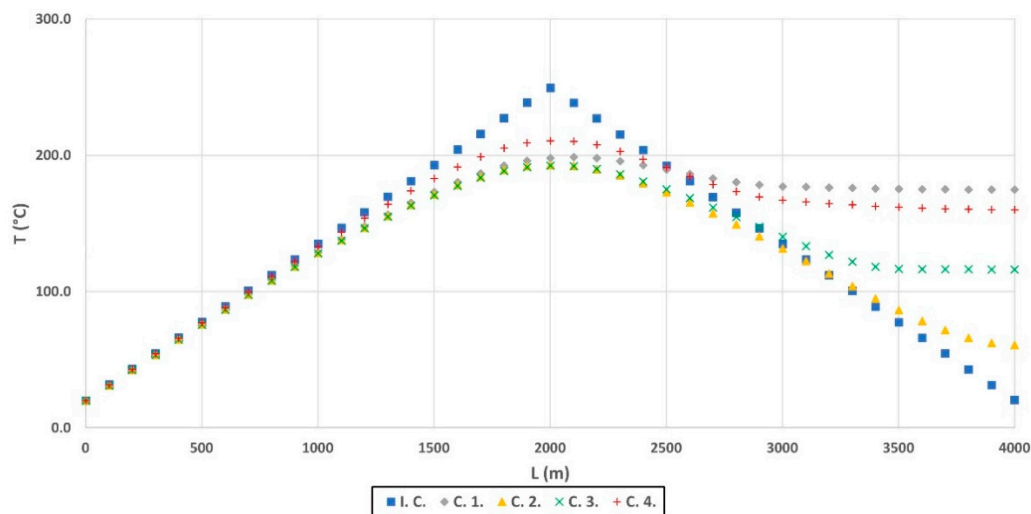


Figure 9. Fluid temperature inside the exchanger: I.C. initial condition (blue), C.1 (orange), C.2 (grey), C.3 (green), C.4 (red).

For the fifth case, the total length of the heat exchanger was reduced, while in the sixth one the insulator material was omitted. The results showed that both cases' temperature for were 145.9 °C and 59.6 °C, respectively. As in the first 2 cases, there is a difference of almost 100 °C in the final temperature of the fluid caused by the absence of the insulator material. Figure 10 shows the results for cases 5 and 6 (3000 m total length).

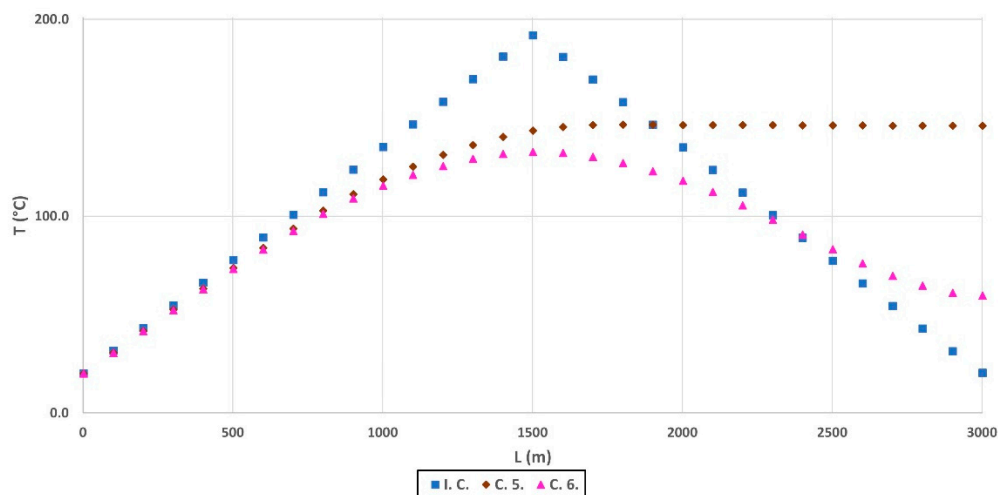


Figure 10. Fluid temperature inside the exchanger: I.C. initial condition (blue), C.5. (coffee), C.6. (pink).

4.3. Determination of the Cooling Potential (CP)

The CP can be determined for each one of the proposed systems by using Equation (5). The hourly results for the four seasons are presented in Figure 11a–d at a $T_E = 8$ °C. It can be observed that the CP is higher for the double-effect system with a maximum cooling potential of 99,334 GW, followed by the triple-effect with 92,995 GW, 70,939 GW for simple-effect, and 38,721 GW for the half-effect system. These maximum potentials are obtained in winter when the ambient temperature is lower. For the double and triple-effect system, when the environmental temperatures are very high, the systems cannot operate. This happened since the absorber and condenser temperatures were fixed 7 °C above the ambient temperature, and at considerably high absorber temperatures and low pressures, a crystallization phenomenon occurs.

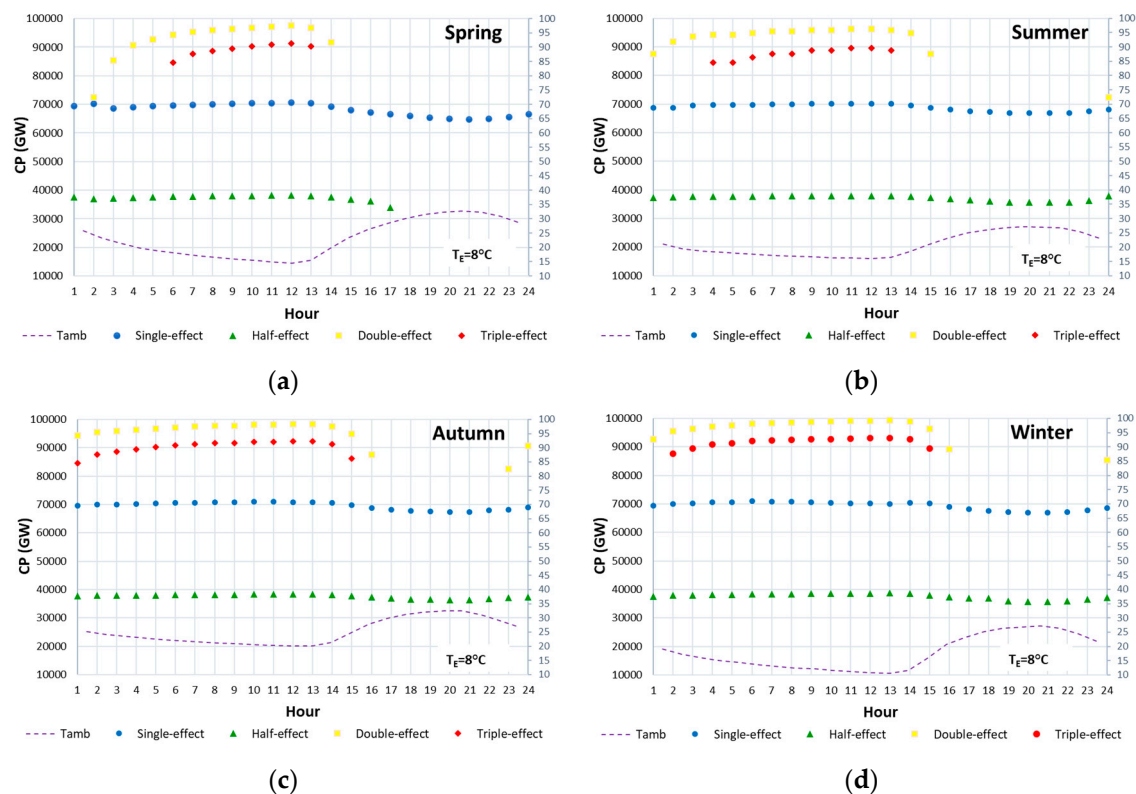


Figure 11. Cooling potential per hour for each absorption system for the season, (a) Spring, (b) Summer, (c) Autumn, and (d) Winter.

From the analysis carried out in Section 4.2, it was clear that the geothermal well can produce useful heat at temperatures up to 175 °C, depending on the analyzed case as it was shown in Table 2 and in Figures 9 and 10. Figure 12a–d show the cooling potential for each one of the proposed systems at a $T_E = 8$ °C for Spring, Summer, Autumn, and Winter, respectively. First, it is noticeable that the triple-effect system cannot operate in the Spring and Summer seasons due to the high ambient temperatures. The rest of the systems may operate for all seasons. Although it was expected that with the triple-effect system the highest cooling potential could be reached, due to its limited operating ranges the CP values are slightly lower than those obtained with the double-effect system. So, the highest potentials are obtained with the double-effect system, followed by the triple-effect system, and then for the single-effect system. The lowest CP values are obtained with the half-effect system, but with the advantage that these type of system may operate at very low temperatures from 59 °C to 79 °C. The single-effect system operates at temperatures between 77 °C and 112 °C, while the other systems require temperatures higher than 140 °C to operate, which can be only obtained for the analyzed cases 2, 4, and 5.

Figure 13a–d show the coefficients of performance for each one of the systems for Spring, Summer, Autumn, and Winter, respectively. As it was expected, according to the data reported in the literature [21], the highest COPs are obtained with the triple-effect system. However, this system has a more limited operating temperature range in contrast to the other systems. The lowest COP values were obtained for the half-effect system varying from 0.43 to 0.49. For the single-effect system, the COPs varied between 0.89 and 0.97, while for the double-effect and triple-effect systems varied from 1.27 to 1.47, and from 1.78 to 1.95, respectively. The maximum COPs are obtained in the Winter and Autumn seasons.

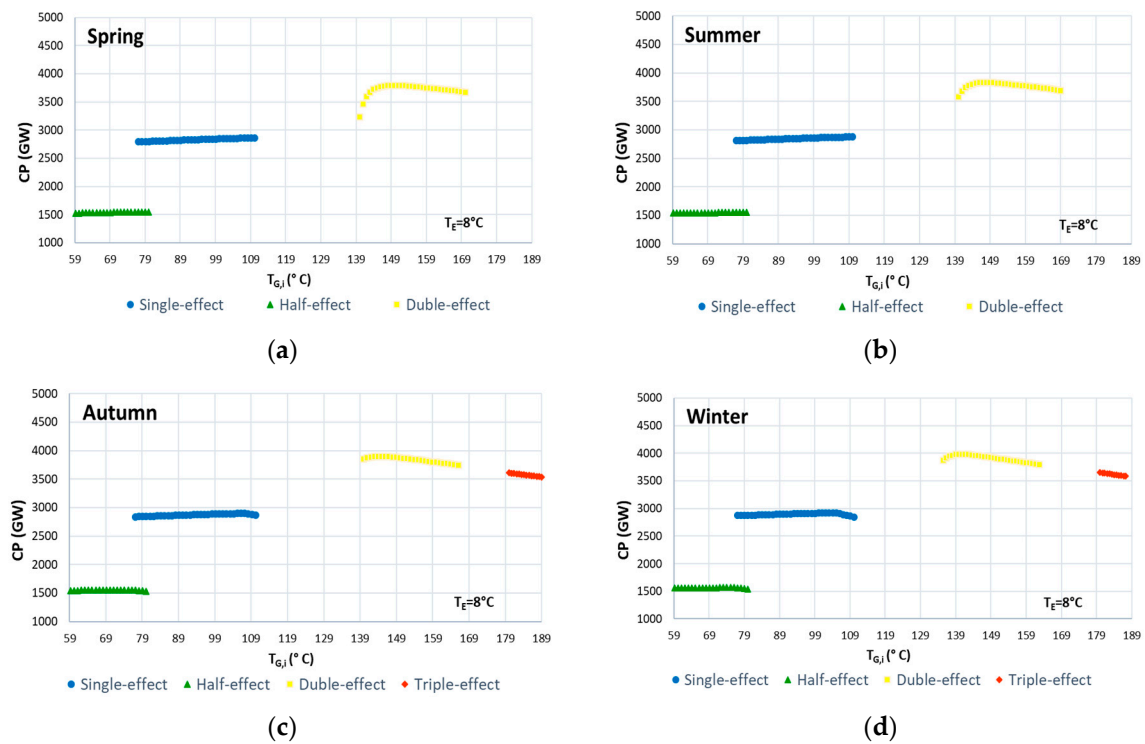


Figure 12. Cooling Potential per hour for each absorption system for the season, (a) Spring, (b) Summer, (c) Autumn, and (d) Winter.

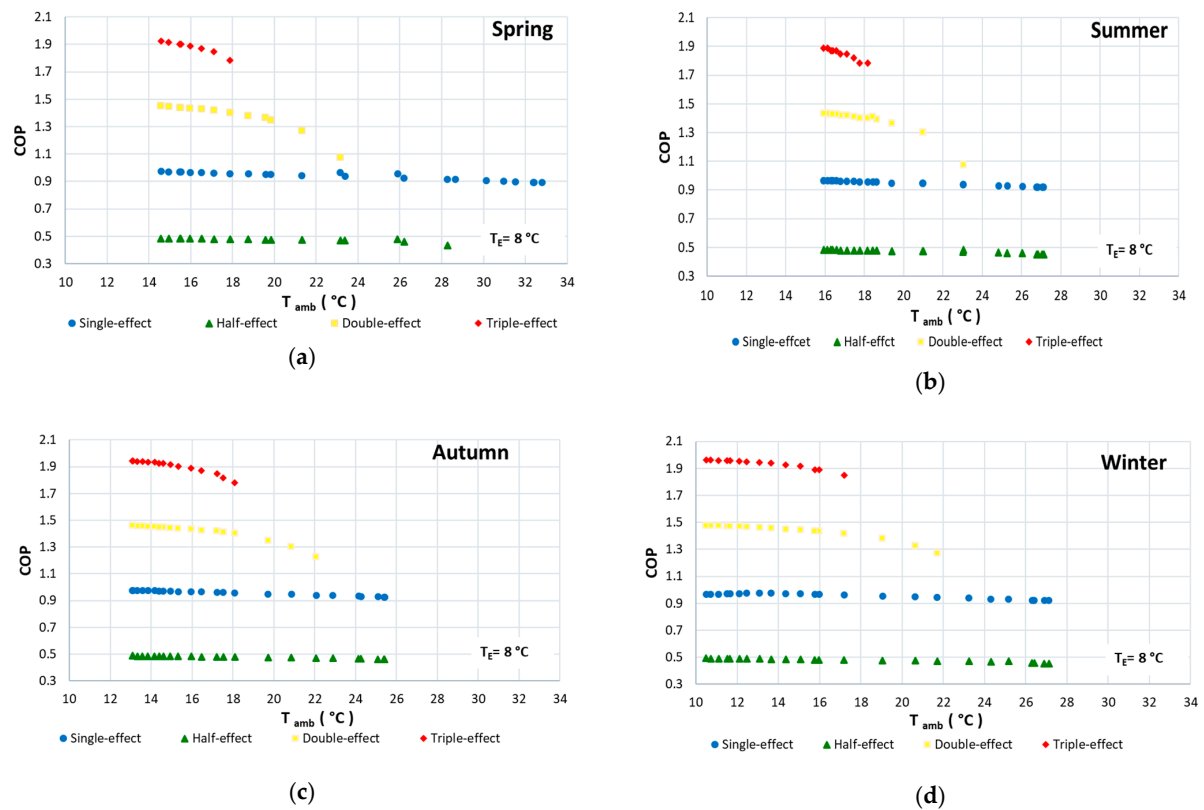


Figure 13. COP as a function of the T_{amb} for each absorption system for (a) Spring, (b) Summer, (c) Autumn, and (d) Winter.

5. Conclusions

A geothermal field located in Mexico was studied with the purpose to assess the performance of the four advanced absorption cooling systems driven by a geothermal heat source.

From the analysis of the geothermal well, it was found that without placing an insulation layer in the tube of return, it is possible to obtain temperatures around 60 °C, which are enough to drive a half-effect system; however, it is necessary the insulation to reach higher temperatures to drive the other systems, especially the double-effect and triple-effect systems.

At a $T_E = 8$ °C, the operating range temperatures were 59–80 °C, 77–110 °C, 135–162 °C, and 180–187 °C, with the half-effect, single-effect, double-effect, and triple-effect systems, respectively.

The range of the coefficients of performance for the four systems were: 0.43–0.49 for the half-effect system, 0.89–0.97 for the single-effect system, 1.27–1.47 for the double-effect, and 1.78–1.95 for the triple-effect system. The maximum values of the coefficients of performance were obtained in Autumn and Winter when the ambient temperatures were the lowest. This occurs since at low ambient temperatures, the condenser and absorber temperatures are also lower, thus reducing the operating pressures of the systems. The lower operating pressures cause an increase of the refrigerant production and therefore an increase of the coefficients of performance.

The maximum cooling potential was obtained with the double-effect system, achieving values up to 99,334 GW, followed by the triple-effect system with 92,995 GW. With the single-effect and half-effect systems the cooling potentials were up to 70,939 GW and 38,721 GW, respectively.

From the analysis, it was demonstrated that the triple-effect system does not present any advantage over the double-effect system, since it is more complex, it requires higher operating temperatures, it has a more limited operating range, and achieved lower cooling potentials.

The technical feasibility to produce cooling at temperatures as low as 58 °C by using geothermal energy was also demonstrated. This fact is very important because around the world there are numerous geothermal wells which can be used for these purposes. However, it is important to consider both the distance from the wells to towns and the possible applications. This is relevant since long distance result in the economic infeasibility because the geothermal areas (hydrothermal) are, in many cases, located far from urban areas.

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Nomenclature

A	area
C_e	efficiency factor
C	specific heat
COP	coefficient of performance
CP	cooling potential

F_p	plant factor
h	enthalpy
H	thickness of the reservoir
IL	insulator length
IT	insulator thickness
$\dot{m}$	mass flow
p	pressure
P	geothermoelectric potential
Q	heat load
Q_t	total energy
R_f	recovery factor
SHE	solution heat exchanger
T	temperature
TL	total length
t	time
W	work
v	fluid velocity
Z	depth

Subscripts

a	absorber
b	pump
c	condenser
e	evaporator
f	fluid
g	generator
G, I	introduced to the generator
i	initial
m	minimum
o	outlet
r	rock
w	wall

Greek letters

α	thermal diffusivity
ρ	density
ε	effectiveness
v	volume
$\emptyset$	porosity of the rock

Appendix A**Table A1.** Energy balances for each one of the components of the single-effect absorption cooling system.

Component	Balances
Generator	$Q_g = \dot{m}_4 h_4 + \dot{m}_7 h_7 - \dot{m}_3 h_3$
Condenser	$Q_c = \dot{m}_7 (h_7 - h_8)$
Evaporator	$Q_e = \dot{m}_{10} (h_{10} - h_9)$
Absorber	$Q_a = \dot{m}_{10} h_{10} + \dot{m}_6 h_6 - \dot{m}_1 h_1$
Effectiveness of the SHE	$\varepsilon_{SHE} = \frac{h_3 - h_2}{h_4 - h_2}$
Pump Work	$W_b = v_1 (P_8 - P_{10})$
COP	$COP = \frac{Q_e}{Q_g + W_b}$

Table A2. Energy balances for each one of the components of the half-effect absorption cooling system.

Component	Balances
Generator	$Q_{gh} = \dot{m}_{13}h_{13} + \dot{m}_{10}h_{10} - \dot{m}_9h_9$ $Q_{gl} = \dot{m}_4h_4 + \dot{m}_{17}h_{17} - \dot{m}_3h_3$
Condenser	$Q_c = \dot{m}_{13}(h_{13} - h_{14})$
Evaporator	$Q_e = \dot{m}_{16}(h_{16} - h_{15})$
Absorber	$Q_{al} = \dot{m}_6h_6 + \dot{m}_{16}h_{16} - \dot{m}_1h_1$ $Q_{ah} = \dot{m}_{17}h_{17} + \dot{m}_{12}h_{12} - \dot{m}_7h_7$
Effectiveness of the SHE	$\varepsilon_{SHE1} = \frac{h_3-h_2}{h_4-h_2}$ $\varepsilon_{SHE2} = \frac{h_9-h_8}{h_{10}-h_8}$
Pump Work	$W_{b1} = v_1(P_2 - P_1)$ $W_{b1} = v_7(P_8 - P_7)$
COP	$COP = \frac{Q_e}{Q_{gh}+W_{b1}+Q_{gl}+W_{b2}}$

Table A3. Energy balances for each one of the components of the double-effect absorption cooling system.

Component	Balances
Generator	$Q_{g2} = \dot{m}_{14}h_{14} + \dot{m}_{17}h_{17} - \dot{m}_3h_3$
Condenser	$Q_{c1} = \dot{m}_7h_7 - \dot{m}_8h_8$
Generator-Condenser	$Q_{g1} = \dot{m}_7h_7 + \dot{m}_4h_4 - \dot{m}_{16}h_{16}$ $Q_{c2} = \dot{m}_{17}h_{17} - \dot{m}_{18}h_{18}$
Evaporator	$Q_e = \dot{m}_{10}(h_{10} - h_9)$
Absorber	$Q_a = \dot{m}_6h_6 + \dot{m}_{10}h_{10} - \dot{m}_1h_1$
Effectiveness of the SHE	$\varepsilon_{SHX} = \frac{h_3-h_2}{h_4-h_2}$ $\varepsilon_{SHX} = \frac{h_{13}-h_3}{h_{14}-h_3}$
Pump Work	$W_b = v_1(P_2 - P_1)$
COP	$COP = \frac{Q_e}{Q_{g2}+W_b}$

Table A4. Energy balances for each one of the components of the triple-effect absorption cooling system.

Component	Balances
Generator	$Q_{g3} = \dot{m}_{22}h_{22} + \dot{m}_{27}h_{27} - \dot{m}_{21}h_{21}$
Condenser	$Q_{c1} = \dot{m}_7h_7 - \dot{m}_8h_8$
Generator-Condenser	$Q_{g2} = \dot{m}_{25}h_{25} + \dot{m}_{14}h_{14} - \dot{m}_{24}h_{24}$ $Q_{c3} = \dot{m}_{27}(h_{27} - h_{28})$ $Q_{g1} = \dot{m}_7h_7 + \dot{m}_4h_4 - \dot{m}_{16}h_{16}$ $Q_{c2} = \dot{m}_{25}(h_{25} - h_{26})$
Evaporator	$Q_e = \dot{m}_{10}(h_{10} - h_9)$
Absorber	$Q_a = \dot{m}_6h_6 + \dot{m}_{10}h_{10} - \dot{m}_1h_1$
Effectiveness of the SHE	$\varepsilon_{SHE1} = \frac{h_3-h_2}{h_4-h_2}$ $\varepsilon_{SHE2} = \frac{h_{13}-h_3}{h_{14}-h_3}$ $\varepsilon_{SHE3} = \frac{h_{21}-h_{13}}{h_{22}-h_{13}}$
Pump Work	$W_b = v_1(P_2 - P_1)$
COP	$COP = \frac{Q_e}{Q_{g3}+W_b}$

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