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Thermodynamic and mathematical analysis of geothermal power plants operating in different climatic conditions

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HIGHLIGHTS

- The geothermal power plant with different cooling systems at various climatic conditions has been modelled.
- Mathematic equations have been developed for each cooling systems.
- The power production and water consumption features of cooling systems at different weather conditions have been evaluated.
- The operation of cooling systems in different climates can be determined considering water reserves in the region.

ARTICLE INFO

Keywords:
Geothermal power plants
Hybrid cooling systems
Mathematical modelling
Thermodynamic analysis

ABSTRACT

This study presents the thermodynamic and mathematical analysis of the different cooling systems for geothermal power plants at various climatic conditions. The existing binary geothermal power plant data, located in Turkey, were processed in the commercial software Ebsilon Professional to model the power plant and cooling systems. The dry-bulb temperature and relative humidity values from data obtained from local meteorological stations in the analyzed region are used as variables in the model. The reference air-cooled cooling system, the wet-tower cooling system, the additional dry cooling system and three different hybrid cooling systems are modelled and analyzed separately in the reference geothermal power plant model. Accordingly, mathematical equations are developed to evaluate the power production and water consumption characteristics of cooling systems on the thermodynamic performance of geothermal power plants depending on weather conditions. The power production and water consumption of geothermal power plants can be calculated using hourly ambient temperature and relative humidity data at different climatic conditions with these equations. Considering water reserves in the area, operation periods of the hybrid cooling systems can be determined. Moreover, power plant operators can compare the performance and water consumption characteristics of hybrid cooling systems for different conditions with these equations.

Nomenclature

Letter

A	area, [m^2]
C_a	correction factor, [−]

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c_p	specific heat capacity, [J kg^{-1} °C^{-1}]
C_{wat}	water consumption factor, [–]
dA	differential heat transfer area, [m^2]
dP	differential pressure difference, [bar]
dT_{wat}	differential water temperature difference, [°C]
eff	effectiveness, [–]
h	specific enthalpy, [kJ kg^{-1}]
h_L	specific enthalpy of air from water, [kJ kg^{-1}]
h_{Lw}	specific enthalpy of saturated air at water surface temperature, [kJ kg^{-1}]
k	thermal conductivity, [W m^{-1} °C^{-1}]
LMTD	logarithmic mean temperature difference, [–]
$\dot{m}$	flow rate of working fluid, [kg s^{-1}]
$\dot{m}_{air}$	flow rate of air, [kg s^{-1}]
$\dot{m}_{des}$	water consumption rate at the reference design condition, [kg s^{-1}]
$\dot{m}_w$	flow rate of water, [kg s^{-1}]
P	working fluid pressure, [bar]
$\dot{Q}_{loss}$	heat loss, [kW]
RH	relative humidity of ambient air, [%]
t	time, [s]
T_{dry}	dry bulb temperature, [°C]
v	specific volume of the working fluid, [kg m^{-3}]
$\dot{W}$	power, [kW]
$\dot{W}_a$	generated power from the power plant at the given dry bulb temperature, [kW]
$\dot{W}_{des}$	generated power from the power plant at design condition, [kW]

Subscripts

aac	additional dry cooling system
air	air
evp	evaporative cooling system
fan	fan
fane	fan motor
hc	hybrid cooling system
hct	hybrid cooling tower system
in	inlet
mech	mechanical
ref	reference model
s	isentropic
wetct	wet cooling tower system

Greek

η	efficiency, [–]
ρ	density, [kg m^{-3}]
σ	mass transfer coefficient, [m s^{-1}]
ΔP	pressure difference, [bar]
$\Delta\dot{W}$	power gain, [kW]

1. Introduction

The energy demand of the world has been accumulating gradually with the development of energy-intensive technologies and the rise in the world population. For instance, the energy consumption rate of Germany and Britain in the 2050s is expected to be higher by about 23% compared to 2015, [1]. Therefore, the energy production rate needs to be elevated to meet the enhanced energy demand, [2]. Consequently, countries have to develop energy policies to use their own resources effectively considering the decreasing fossil fuel resources and increasing fuel prices, [3]. However, renewable energy sources should be considered in the energy production processes because of global warming, ozone depletion and environmental pollution issues, [4]. Renewable energy resources are the fastest-growing energy source, [5]. The share of renewable energy sources in the total energy supply is predicted to be 15% in 2040 with an annual increase rate of 2.8%, [6]. At this point, geothermal energy can play a significant role to meet the world's enhanced energy demand with its' clean, natural and sustainable features. Geothermal energy has become a remarkable component in energy systems and has played a role in energy production from renewable sources since the beginning of the twentieth century, [7]. Paulillo et al. [8] reported that global electricity production from geothermal energy sources was around 87 TWh (0.3%) in 2020. Rudiyanto et al. [9] reported that the total capacity of geothermal power plants in the world in 2015 was around 12.6 MW. However, this capacity reached 13.931 GW in 2019 which is a great increment, [10].

The cooling systems of power plants are important components since they have a large share of the total production cost and directly affect plant efficiency and energy production capacity. The climatic and water resources characteristics of the region where the power plant will be installed are needed to be considered in the selection of the cooling system [11]. In geothermal binary plants, the dry cooling systems are ineffective whereas the wet cooling systems consume a high amount of water. Furthermore, the environmental effects of cooling systems need to be considered since they may threaten organic life in water resources and decrease the water reserves to critical levels because of high water consumption Hence, the advanced hybrid cooling systems, which are based on mixed dry and wet cooling methods, having reduced cost and environmentally friendly features need to be implemented. Although there is a plethora of researches on the hybrid cooling systems of conventional power plants in the literature, only a few studies have studied hybrid cooling systems of geothermal power plants. The following literature survey is segregated into two major parts. First, the hybrid cooling system studies regarding the various power plants are presented. Then the hybrid cooling system studies in geothermal power plants are summarized.

Rezaei et al. [12] modelled the hybrid cooling towers having different configurations to decrease water consumption by developing computer software. The first hybrid cooling tower had a series arrangement where the counter flow fill was divided into five sections. On the other hand, the authors utilized tubes with rotated triangular configuration for the second hybrid cooling tower arrangement. The experiments were performed to verify the simulation results. The authors calculated the total water loss and required heat exchanger surface area with 20%, 40%, 60% and 80% hybrid ratio where the hybrid ratio was defined as the ratio of the cooling load of dry system to the total cooling load. It was found that the best hybrid ratio is between 20% and 40% for summer. The payback time was found to be 12.1 years in the second hybrid system whereas it was 5.2 years for the first hybrid system. In another study, Carter et al. [13] developed a new dry cooling system called thermosiphon cooler which was integrated into power plants having a wet cooling system to save water consumption. In this system, the cooling water leaving the condenser passes through the pipes of the thermosiphon cooler heat exchanger. The coolant on the outside of the pipe evaporates the organic fluid. The evaporating fluid proceeds to the cooling tower where it is condensed and re-directs back to the heat exchanger. The results showed that the thermosiphon cooler can save a significant amount of water in power plants with wet cooling towers. Card [14] designed a parallel connected hybrid cooling system comprising an air-cooled condenser (ACC) and a conventional surface condenser connected to a wet cooling tower to save water consumption and improve the cooling performance. A thermodynamic simulation model was developed to analyze a combined cycle power plant (CCPP) with this hybrid cooling system. The model was simulated under different conditions to reveal the effects of various parameters on plant performance such as dry bulb temperature, wet bulb temperature, and number of fans operating in ACC. Asvapoositkul and Kuansathan [15] examined the performances of the wet, dry and hybrid cooling towers with sets of numerical simulations and experiments for a wide range of operating conditions. The proposed hybrid cooling tower consisted of a combination of dry cooling tower and wet cooling tower systems where the outlet airs of both systems were mixed in the mixing chamber. The authors reported that the characteristics of cooling towers are determined according to the air-water ratio in each cooling method. The analyses showed that the wet cooling tower dissipates five times more heat compared to the dry cooling tower. It was shown that the airflow rate is directly proportional to the fan power. Barigozzi et al. [16] developed a computer model of the hybrid cooling system of a waste-to-energy cogeneration plant. The hybrid cooling system was designed by connecting the ACC and the conventional condenser to the wet cooling tower in parallel. The results showed that the net power was maximum when dry and wet cooling systems are optimized properly. Zhai and Rubin [17] investigated the economic, technical and environmental characteristics of hybrid cooling systems in CCPPs and coal power plants. A hybrid cooling system in which the ACC and the surface condenser connected to the wet cooling tower in the parallel arrangement was modelled with computer-based software by the researchers. The results showed that the hybrid cooling system provides water savings with favorable initial cost in coal and natural gas CCPPs compared to conventional cooling systems. In another study, Hu et al. [18] compared the performance of the power plant with completely dry cooling, completely wet cooling and hybrid cooling systems for a wide range of dry bulb temperature and relative humidity conditions. The authors reported that the thermal performance of the hybrid cooling system is superior compared to other cooling systems and also provided less water consumption.

Jung and Wai [19] proposed a method to cool the inlet air of the ACC with the evaporative method of hot water used in the geothermal cycle. They reported that the cooling air temperature at 32.2° C can be reduced to 15.6° C with waste geothermal water at 71.1° C. The experimental results showed the inlet air can be cooled without sedimentation in the finned tubes using the proposed method. Furthermore, this method was found to be cheaper compared to the traditional ACC system with the wet cooling tower. In another study, Kozubal and Kutscher [20] analyzed the performance of a geothermal power plant using a hybrid cooling system where the ACC and conventional condenser were connected in series. They reported that the ACC could meet the total cooling load at temperatures below 12.2° C whereas the wet cooling system was activated at higher temperatures. The corresponding hybrid cooling system was found to increase the performance of the power plant. Buys et al. [21] stated that using evaporative methods to cool the inlet air of the ACC could provide power gain. However, the regions where geothermal power plants are established are arid regions. Therefore, they suggested that the wastewater in geothermal power plants can be used in evaporative cooling techniques. Subsequently, they investigated two types of evaporative cooling techniques, namely spray and munters. They found these methods are not suitable since 10% exergy loss occurred during flashing and extra pumping power needed to re-pump the geothermal water after the flashing process. They concluded that the geothermal water can be used directly with coated finned tubes in the evaporative cooling system to increase the plant performance by protecting equipment. In the technical report of Ashwood and Bharathan [22], alternative hybrid cooling systems with less water consumption and higher performance were investigated for binary geothermal power plants with ACC under hot weather conditions. They reported that the wet cooling component of the hybrid system could meet 30% of the

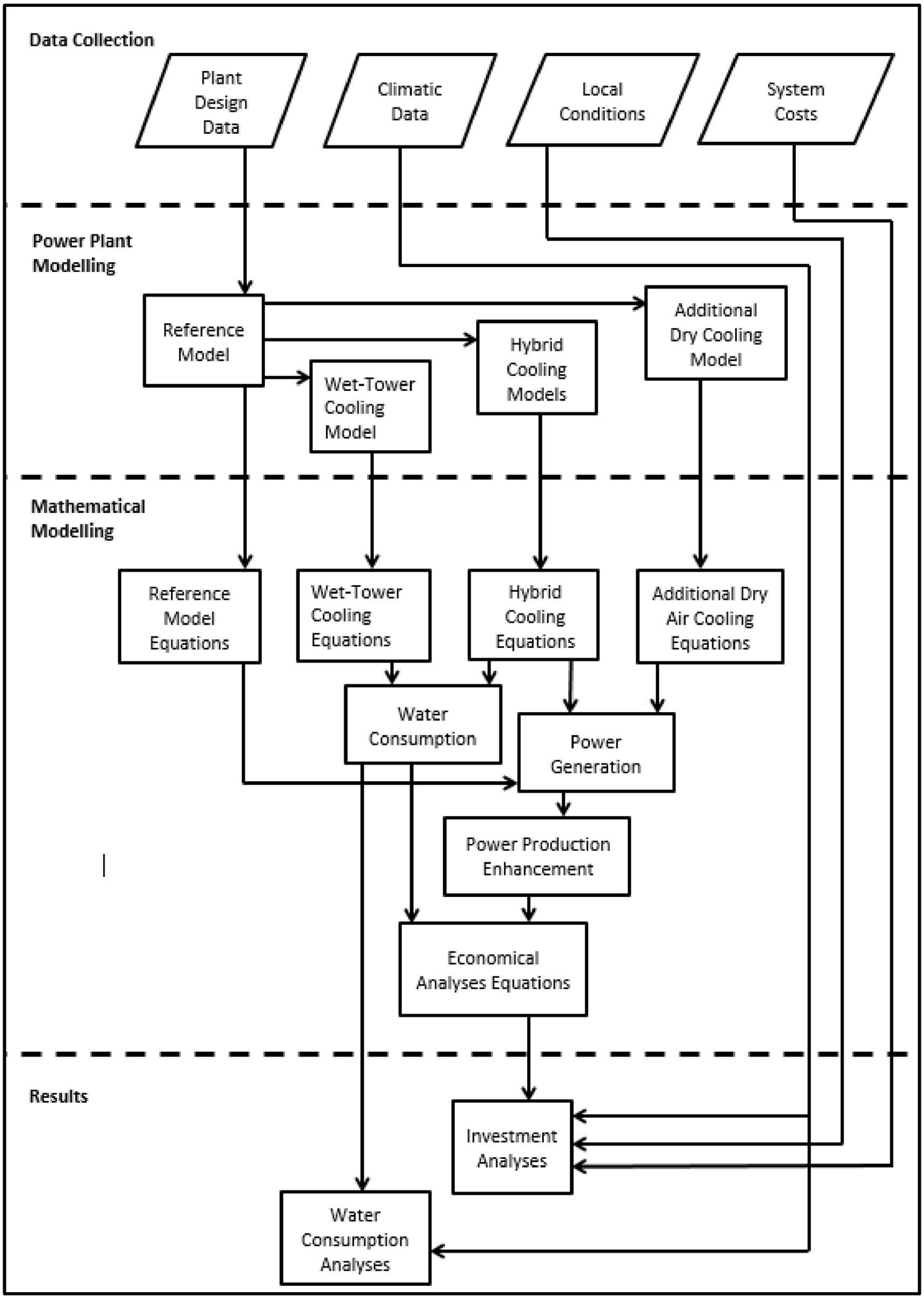

Fig. 1. The flowchart of analysis method.

total cooling load with the maximum annual operating time of 1000 h. Moreover, the wet component only used 3.5% of the water consumption in the completely wet system. They proposed two different hybrid cooling systems by considering the cost analysis. The first proposed system was the evaporative cooling method of the inlet of the ACC. This payback of this system was found to be less than two years. In the second system, the authors installed a condenser and wet cooling tower parallel to the existing ACC system. It was found that the payback of this system was around 4.5–6.1 years. In another technical report, Bharathan [23] reported that the binary geothermal power plants with ACCs faced over 50% power degradation at temperatures above 30 $^{\circ}$C. Accordingly, the author investigated various hybrid cooling systems and found that the deluge cooling system was the most appropriate one among evaporative cooling systems due to the lower cost and higher performance. Moreover, the author stated that the water required to operate this hybrid system in geothermal power plants can be obtained from wastewater facilities and irrigation sources since the geothermal power plants were established in arid regions.

The above literature review shows that the researches on hybrid cooling systems in geothermal power plants are rather limited. However, hybrid cooling systems have come to the agenda recently due to the elevated energy demand and water consumption rates. In the literature, there exists a few studies considering climatic conditions analyzing hybrid cooling systems in geothermal power plants. The conventional power plant cycles are based on steam-water Rankine cycles whereas the analyzed geothermal power plant with binary cycles is using organic fluid as working fluid and therefore characteristics of the two cycles are different from each other. Furthermore, the highest temperature of the ORC cycle is lower compared to the conventional Rankine cycle and consequently, the ORC cycle is more affected by the ambient temperature. These differences necessitate analyzing the cooling system in geothermal power plants separately from the conventional Rankine cycle. In this study, mathematical and thermodynamic equations are developed for the pre-existing air-cooled system in the plant, wet-tower cooling system, additional dry cooling system and three different hybrid cooling systems to increase the performance of geothermal power plants in hot weather conditions. In this context, the software has been developed to model the real geothermal power plant with the pre-existing air-cooled system. After the model has been verified with the power plant data, different cooling systems have been integrated into the power plant to analyze power production and water consumption. The equations provide to analyze the effects of temperature and humidity on power and water consumption.

2. Materials and methods

The geothermal power plants have been generally established in the regions with insufficient water resources. Therefore, the dry (air) cooling systems are used for cooling purposes to save water. However, the performance of plants with air-cooled systems is lower compared to plants using hybrid cooling systems. In this study, the mathematical models have been developed to calculate the power plant performance and water consumption as functions of dry bulb temperature and relative humidity for different cooling systems. The reference air-cooled cooling system, the wet-tower cooling system, the additional dry cooling system and three different hybrid cooling systems namely the evaporative cooling system, hybrid-tower cooling system and hybrid cooling system have been modelled. There are two reference models have been utilized in the current study. The air-cooled condenser is the first reference model where the cooling is performed only with an air-cooled condenser. The wet-tower cooling system is the second reference model which is the reference for water consumption. It calculates the maximum water consumption if dry air cooling was not used. This model has been developed in order to compare the amount of water consumption in hybrid cooling models. For cooling process, surface condenser and wet type cooling tower are placed instead of air cooled condenser.

In the reference air-cooled cooling model, the environment air is directed to the finned tubes through which the working fluid passes by the fan located in the center of the cell. The air passing through the finned tubes cools the working fluid and allows it to condense inside the tubes. If the temperature of the cooling air can be reduced, the work fluid can also be cooled to lower temperatures. As a result, the working pressure of the condenser decreases and the amount of power generated in the cycle increases. The details of the system description can be found in Refs. [24,25]. It has been suggested in the literature that evaporative cooling techniques can be used to cool the air entering the air-cooled condenser [26].

In evaporative cooling systems, inlet air is cooled with liquid phase water by various methods. Some of the liquid water passes into the vapor phase by using the energy of the air and thus cools the inlet air. This method is called evaporative cooling. This configuration can be considered as hybrid cooling where wet and dry cooling are applied together, as additional cooling is performed using water in the dry cooling system.

The hybrid-tower cooling system uses dry and wet cooling components together. These components are connected in series in the cooling tower.

On the other hand, the wet cooling system using water is connected in series to the air-cooled condenser in the hybrid cooling model. In this model, a heat exchanger connected to the wet cooling tower enters to the process before the air-cooled condenser.

Although it is not directly related to this study, the additional dry air cooling model has also been developed in order to ensure the integrity of the subject and to compare it with hybrid cooling systems. In this model, it is assumed that an additional dry air cooling model corresponding to approximately 25% of the capacity of the pre-existing air-cooled condenser is added. The details of the description of the above-mentioned systems can be found in Ref. [25].

The developed mathematical models enable to compare the different cooling models at different climatic conditions. The methodology of this study consist of three major parts: collecting data, modelling the power plant and evaluating mathematical equations. The analysis method is presented in Fig. 1 as a flowchart. Firstly, the geothermal power plant design data, the climatic data and the regional conditions have been collected for analyses. The power plant data includes the flow chart presenting the thermodynamic cycle and equations of state data. In the modelling procedure, the equations of state values have been considered at the reference environment conditions. The dry bulb temperature and relative humidity data have been used as climatic values in the area where the

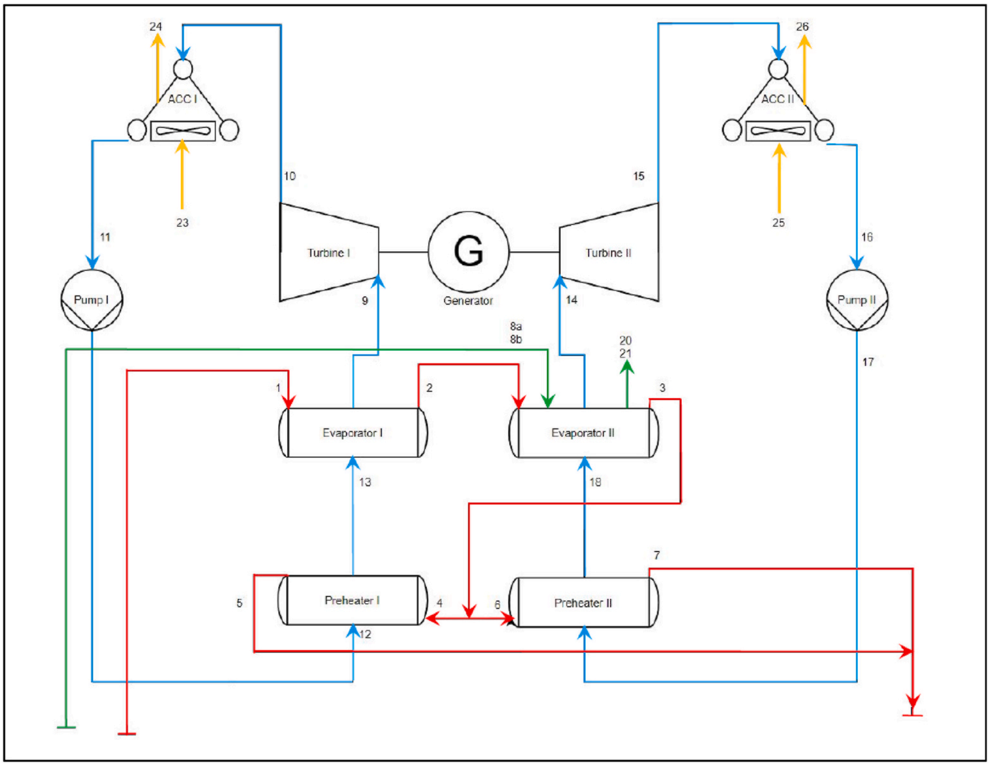

Fig. 2. The flowchart of the binary geothermal power plant.

power plant was established using the measurements of the local meteorology station. Since the accuracy of the calculation increases by using the small-time period, the hourly measurement data have been used in this study. During the analyses, it was assumed that the dry bulb temperature and relative humidity data remained constant for 1 h.

After the collection of necessary data, the geothermal power plant has been modelled using a commercial software Ebsilon Professional©. The model has been verified with the reference design data of the power plant. Thereafter, the different cooling systems are integrated to the verified model. The effects of dry bulb temperature and relative humidity at different climatic conditions for different models have been evaluated. The regression method has been used to develop mathematical models using the results from the models. The mathematical formulations have been employed to determine the power plant performance and water consumption data at different climatic conditions for analyzed cooling systems.

3. Thermodynamic model

The flowchart of the binary geothermal power plant is demonstrated in Fig. 2. The reference geothermal power plant is a real power plant (Dora II Geothermal power plant) located in Turkey, see Ref. [11]. There are two independent binary cycles in the power plant and the working fluid was taken as the n-pentane (organic fluid) in both cycles. Therefore, both cycles can be modelled as the Organic Rankine Cycle. However, there is a single generator that is used for two cycles. The pressure stages of the two cycles are different. The cycle where the geothermal water first enters the evaporator is a high-pressure cycle whereas the other one is a low-pressure cycle. The geothermal water drawn from the production wells enters the cycle by entering evaporator 1 and evaporator 2. The geothermal water makes the n-pentane which is in the first cycle into the saturated vapor in evaporator 1 and thereafter enters evaporator 2. The geothermal water coming out of the evaporator 2 is divided into two streams and these two streams enter preheater 1 and preheater 2. The working fluid n-pentane becomes saturated liquid in preheaters. The geothermal water coming out of the preheaters is directed into reinjection wells to be sent back to the source. It can be observed that the working fluid is firstly pressurized by the pump and then becomes saturated liquid in the preheater. Afterwards, the working fluid becomes saturated vapor in the evaporator and then it expands in the turbine. The working fluid completes the cycle by finally condensing in the ACC. As can be seen from Fig. 2, there are various equipment such as heat exchangers, turbines and ACCs in the power plant. The thermodynamic calculation equations of the equipment are presented below.

3.1. Heat exchanger

The mass and heat balance equations of the heat exchangers namely the evaporators and condensers are given in the following equations.

Table 1
Actual flow measurements of the reference binary geothermal power plant.

State number	Explanation	Fluid	State	Temperature T (°C)	Pressure P (kPa)	Specific enthalpy h (kJ/kg)	Flow Rate ṁ (kg/s)
0	–	Brine	Dead state	17.1	101.3	70.75	
0′	–	n-Pentane	Dead state	17.1	101.3	−44.05	
0″	–	CO_2	Dead state	17.1	101.3	206,12	
0″	–	Air	Dead state	17.1	101.3	290.25	
1	Evaporator I inlet	Geothermal water	Liquid	169	1296.96	715.11	231.94
2	Evaporator II inlet	Geothermal water	Liquid	124.4	1296.96	523.15	231.94
3	Evaporator II outlet	Geothermal water	Liquid	100	1246.3	424.077	231.94
4	Preheater I inlet	Geothermal water	Liquid	100	1246.3	424.077	115.97
5	Preheater I outlet	Geothermal water	Liquid	81.8	1215.9	343.36	115.97
6	Preheater II inlet	Geothermal water	Liquid	100	1246.3	424.077	115.97
7	Preheater II outlet	Geothermal water	Liquid	83.77	1246.3	351.75	115.97
8a	Evaporator II inlet	Geothermal water	Vapor	169	1246.3	2767.7	2.49
8b	Evaporator II inlet	CO_2	Vapor	169	1246.3	343.91	2.10
9	Turbine I inlet	n-Pentane	Vapor	133.08	1185.5	510.23	118.61
10	Turbine I outlet	n-Pentane	Vapor	83.7	141.85	444.46	118.61
11	ACC I outlet	n-Pentane	Liquid	41.87	141.85	20.66	118.61
12	Pump I outlet	n-Pentane	Liquid	43.3	1185.5	24.89	118.61
13	Preheater I inlet	n-Pentane	Liquid	89.3	1185.5	133.88	118.61
14	Turbine II inlet	n-Pentane	Vapor	95.3	547.15	452.83	97.24
15	Turbine II outlet	n-Pentane	Vapor	69.6	121.5	418	97.24
16	ACC II outlet	n-Pentane	Liquid	45.4	121.5	23.06	97.24
17	Pump II outlet	n-Pentane	Liquid	47.55	547.15	27.85	97.24
18	Preheater II inlet	n-Pentane	Liquid	92.3	547.15	142	97.24
19	Geothermal water reinjection	Geothermal water	Liquid	83	1215.9	348.4	231.94
20	Evaporator II, outlet, waste	Geothermal water	Vapor	112.5	1215.9	2695.25	0.10
21	Evaporator II, outlet, waste	CO_2	dissolved	140	1215.9	290.23	2.71
22	Evaporator II outlet	Geothermal water	Liquid	96.9	1215.9	405.96	2.39
23	ACC I inlet	Air	Gas	17.1	1013	290.25	4350
24	ACC I outlet	Air	Gas	29.1	1013	302.3	4350
25	ACC II inlet	Air	Gas	17.1	1013	290.25	4350
26	ACC II outlet	Air	Gas	26.1	1013	290.25	4350

$$\dot{m}_{in} = \dot{m}_{out} \tag{1}$$

$$P_{in} = P_{out} + dP \tag{2}$$

$$(kA) \times LMTD = \dot{m}_{in}h_{in} - \dot{m}_{out}h_{out} - \dot{Q}_{loss} \tag{3}$$

Fig. 2 represents the flowchart of the binary geothermal power plant. The liquid geothermal water leaving from the production well enters evaporator I (states 0–1) and evaporator II (states 2–3) respectively. Subsequently, the liquid geothermal water enters preheater I (states 4–5) and preheater II (states 6–7). Then the liquid geothermal water flows to re-injection geothermal wells. The geothermal water that is drawn from the production wells has some portion of geothermal vapor and CO_2 which enters the evaporator II (state 8a, 8b). Most of the geothermal condense in evaporator II and is drained as liquid geothermal water. The remaining geothermal vapor is vented from evaporator II (state 20). The condensed CO_2 is drained from evaporator II. There are two different identical cycles for power generation which use organic fluids as working fluids. Because of that, this cycle is called as Organic Rankine Cycle. The organic fluid is n-pentane in this study. The difference between these two cycles is the pressure difference; one is high pressure and the other operates with lower pressure. The turbines of the cycles are connected to one generator. The ORC starts with liquid n-pentane (state 11–16). The liquid n-pentane is pressurized with pump I and pump II (state 12- state 17). The pressurized n-pentane enters preheater I and preheater II (state 12 and state 17). The n-pentane exit preheaters in the saturated liquid phase (state 13-state 18). Then saturated

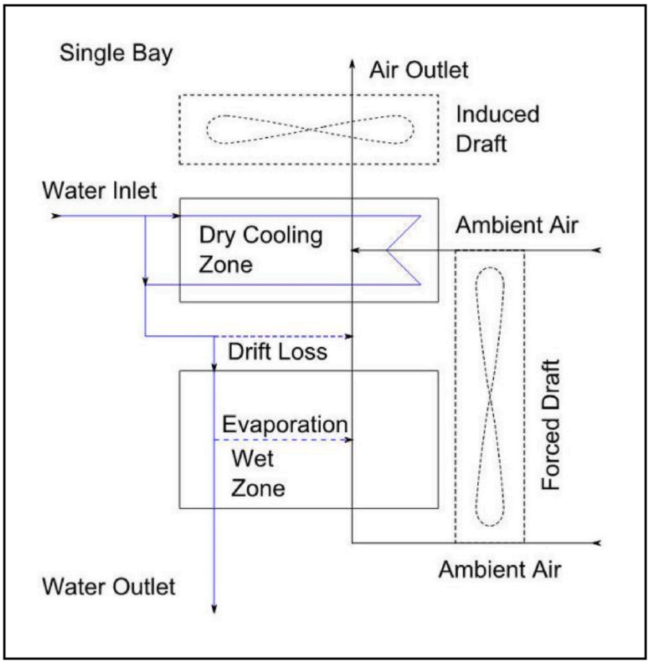

Fig. 3. The flowchart of the hybrid cooling tower.

liquid n-pentane enters evaporator I and evaporator II where it will become saturated vapor phase (state 9-state 14). Saturated vapor n-pentane enter turbines at both cycles. The saturated working fluid expands at the superheated region. This is the property of organic fluids. The expanded n-pentane in the turbine is discharged (state 10-state 15) and flows through ACCs for condensing. The condensed n-pentane is directed to pumps and the cycle has been completed. ACCs utilize air as a cooling medium. Air enters ACCs (state 23, state 25) and cools and condenses n-pentane and exits ACCs (state 24-state 26). The details of the states of the fluids are also given in Table 1 below. The reason why two wells are used in the reference binary geothermal power plant can be explained by the separation of the steam of the geothermal water after exiting the well to the earth to support the evaporator II.

3.2. Turbine

The expansion of the working fluid is considered the energy of the turbine. The Stodola law has been used for the thermodynamic calculation of the turbine model, see Ref. [27]. The isentropic efficiency has been considered for the power output of the turbine.

$$\text{Stodola Constant} = \frac{\dot{m}_{in}}{\sqrt{\frac{P_{in}}{v_{in}}}} \tag{4}$$

$$\dot{W} = [\dot{m}_{in}(h_{in} - h_{out})] \times \eta_{mech} \tag{5}$$

$$\dot{W} = \left[\dot{m}_{in}\left(h_{in} - h_{out,s}\right) \times \eta_s\right] \times \eta_{mech} \tag{6}$$

3.3. ACC

The working fluid leaving from the turbine is condensed through the ACC. ACCs consist of an array of cells with a fan in each. The number of cells is determined according to the cooling load. The power consumption in the fans can be calculated using Eq. (7). The heat transfer area of the ACC can be calculated with a similar method to those for the heat exchangers. The mechanical, fan and fan engine efficiencies are taken from the power plant data and are used as $\eta_{mech} = 99\%$, $\eta_{fan} = 98\%$ and $\eta_{fane} = 67\%$ in the model.

$$\dot{W}_{fan} = \frac{\dot{m}_{air}}{\rho_{air}} \times \Delta P \times \frac{1}{\eta_{fan}} \times \frac{1}{\eta_{fane}} \tag{7}$$

3.4. Evaporative cooler

The inlet air is cooled with the evaporation of the water in the evaporative cooler before the air passes to the ACC. The evaporation heat of water is drawn from the air stream therefore the temperature of the air decreases. The effectiveness of the evaporative cooler is calculated using Eq. (8).

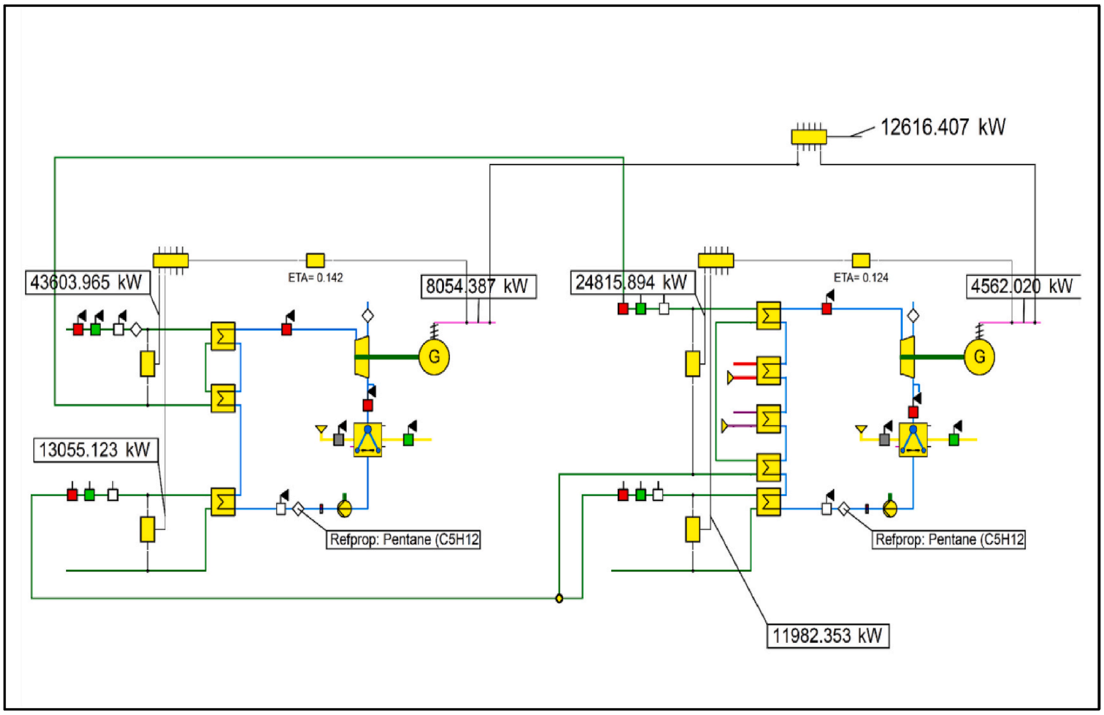

Fig. 4. The model of the reference binary geothermal power plant.

$$\text{eff} = \frac{T_{in} - T_{out}}{T_{in} - T_{dry}} \tag{8}$$

The mass balance eqn. is given in Eq. (9) below.

$$\sum_{in} \dot{m} = \sum_{out} \dot{m} \tag{9}$$

3.5. Wet cooling tower

The thermodynamic calculation of the wet cooling tower model is performed using the Merkel equation, see Ref. [28].

$$\int \frac{c_p dT_{wat}}{h_{LW} - h_L} = \int \frac{\sigma dA}{\dot{m}_w} \tag{10}$$

The hybrid cooling tower model with wet and dry cooling components has also been generated. The flowchart of the hybrid cooling tower is presented in Fig. 3. As can be seen from Fig. 3, the wet and dry components are connected in series in the hybrid cooling tower. The cooling water coming from the condenser is first cooled in the dry part, then the cooling process is completed in the wet part. Thus, the cooling load is shared between wet and dry units and this sharing rate can be adjusted. When the load of the dry cooling part increases, the water consumption decreases and the cooling performance decreases. Similarly, if the ratio of the wet part to share the cooling load increases, the water consumption increases and the power plant performance increases.

3.6. Surface condenser

The working fluid is condensed in the surface condenser by transferring its heat to the cooling water. Accordingly, the mass and heat balance equations are presented below.

$$\dot{m}_{in} = \dot{m}_{out} \tag{11}$$

$$P_{out} = P_{in} - dP \tag{12}$$

$$(kA) \times LMTD = \sum_{in} \dot{m}h - \sum_{out} \dot{m}h \tag{13}$$

Accordingly, the detailed thermodynamic model of the binary geothermal power plant data has been constructed in the Ebsilon Professional© software as can be seen in Fig. 4. The equipment in the thermodynamic model has been calibrated considering the phase change points of the working fluid. Please note that this model represents the original reference model of the plant which uses a complete air cooling method.

Table 1 shows the actual flow measurements taken from the reference binary geothermal power plant for various states. Accordingly, these values have been defined into the reference model. Please note that the first column of Table 1 refers to the numbering presented in Fig. 2.

Subsequent to generating the reference model, five different cooling systems namely; the wet-tower cooling system, the additional dry cooling system and three different hybrid cooling systems are separately adopted to the reference geothermal power plant model. The power production and water consumption data of the power plant for each model depending on the ambient temperature and relative humidity values can be evaluated precisely.

4. Mathematical model

This section presents analyses for different cooling systems as well as the air-cooled cooling system which is the pre-existing original cooling system of the plant. In the analysis, the RH has been considered as constant in air-cooled cooling model and additional air cooling model at 60% since the RH is negligible in air cooling only. Firstly, the power production of the plant has been calculated for the reference model at different temperatures. The correction coefficients have been calculated for different temperatures based on the power production under design conditions (17 °C). The correction coefficient has been taken as 1 in design conditions. Accordingly, when the correction coefficient is less than 1, it means that the production of the plant is less than the design conditions. Similarly, the production of the plant is more than the design conditions when the correction coefficient is greater than 1. Subsequently, the power characteristic curve of the plant has been sketched with these coefficients. A mathematical model has been developed to calculate the power production requirement of the plant at different environment temperatures using the regression method. This model enables to calculate of the power production of the plant at any dry bulb temperature.

The above-mentioned analyses have been also performed separately for the wet-tower cooling system, the additional dry cooling system and three different hybrid cooling systems in addition to the reference air-cooled cooling system. Water consumption is an important parameter for cooling systems using water. Furthermore, the RH of the environment has been taken as a variable for the cooling systems using water since it seriously affects power generation and water consumption. In addition, water consumption coefficients for all cooling models have been calculated and consequently, water consumption characteristic curves have been obtained for these models.

4.1. Power production characteristics

According to the reference model, the plant produces 10.51 MW of net electricity at 17 °C environment temperature. Please note that the net electricity production data have been considered for all models since the results are more reasonable using net production data. The power correction coefficients have been obtained at different temperatures using Eqs. (14)–(19) for different cooling systems utilized in this study.

Air-cooled cooling model (Reference Model)

$$C_{a,ref} = 1 - \left(\dot{W}_{des,ref} - \dot{W}_{a,ref}\right)/\dot{W}_{des,ref} \tag{14}$$

Wet-tower cooling model

$$C_{a,wetct} = 1 - \left(\dot{W}_{des,wetct} - \dot{W}_{a,wetct}\right)/\dot{W}_{des,wetct} \tag{15}$$

Evaporative cooling model

$$C_{a,evp} = 1 - \left(\dot{W}_{des,evp} - \dot{W}_{a,evp}\right)/\dot{W}_{des,evp} \tag{16}$$

Hybrid-tower cooling model

$$C_{a,hct} = 1 - \left(\dot{W}_{des,hct} - \dot{W}_{a,hct}\right)/\dot{W}_{des,hct} \tag{17}$$

Hybrid cooling model

$$C_{a,hc} = 1 - \left(\dot{W}_{des,hc} - \dot{W}_{a,hc}\right)/\dot{W}_{des,hc} \tag{18}$$

Additional dry cooling model

$$C_{a,aac} = 1 - \left(\dot{W}_{des,aac} - \dot{W}_{a,aac}\right)/\dot{W}_{des,aac} \tag{19}$$

4.2. Water consumption characteristics

Besides power production, water consumption is an also important parameter in the design and development of cooling systems. The wet-tower cooling model, the evaporative cooling model, the hybrid-tower cooling model and the hybrid cooling model consume water in this study. The wet-cooling tower model using complete wet cooling has been developed in order to compare the amount of water consumption in hybrid cooling models. The wet-cooling tower model is the model with the highest water consumption. The amount of water used in hybrid systems should be compared with the water flow rate consumed in this model. The water consumption correction coefficients have been obtained for different water consumption using Eqs. (20)–(23) for different cooling systems utilized in this study. Please note that the hybrid cooling model consists of wet tower cooling model and air-cooled cooling model, connected together in series.

Table 2
Net power correction coefficients of the reference model.

Dry bulb temperature (°C)	Correction coefficient [−]
0	1.324361
5	1.226574
10	1.130669
15	1.031977
17	1
20	0.93989
25	0.842965
30	0.749808
35	0.660839
40	0.57532

Table 3
Net power correction coefficients of the wet-tower cooling model.

[−]		Dry bulb temperature [°C]			
		10	20	30	40
Relative Humidity [%]	**10**	1.10324	1.04787	0.99318	0.93313
	20	1.09379	1.0321	0.96558	0.89153
	30	1.08454	1.01685	0.93948	0.85325
	40	1.07548	1.00209	0.91473	0.81775
	50	1.0666	0.98658	0.89119	0.7848
	60	1.0579	0.97136	0.86874	0.75415
	70	1.04937	0.9566	0.84724	0.72559
	80	1.04099	0.94229	0.82664	0.69875
	90	1.03276	0.92838	0.80692	0.67276
	100	1.02468	0.91488	0.78805	0.64876

Wet-tower cooling model

$$C_{wat,wetct} = 1 - \left(\dot{m}_{des,wetct} - \dot{m}_{w,wetct}\right)/\dot{m}_{des,wetct} \tag{20}$$

Evaporative cooling model

$$C_{wat,evp} = 1 - \left(\dot{m}_{des,evp} - \dot{m}_{w,evp}\right)/\dot{m}_{des,evp} \tag{21}$$

Hybrid-tower cooling model

$$C_{wat,hct} = 1 - \left(\dot{m}_{des,hct} - \dot{m}_{w,hct}\right)/\dot{m}_{des,hct} \tag{22}$$

Hybrid cooling model

$$C_{wat,hc} = 1 - \left(\dot{m}_{des,hc} - \dot{m}_{w,hc}\right)/\dot{m}_{des,hc} \tag{23}$$

5. Results and discussions

The power correction coefficients at different temperatures can be evaluated using Eqs. (14)–(19). The net power correction coefficients for the reference model and the wet-tower cooling model are provided in Tables 2 and 3 respectively as an example. The net power correction coefficients of other models can be calculated using Eqs. (14)–(19) with the similar procedure.

Subsequently, net power production equations as functions of the dry bulb temperatures and relative humidity have been obtained using the regression method based on the correction coefficients, please see Eqs. (24)–(29). Please note that the regression equations of the reference model and the additional dry cooling model are only a function of the dry bulb temperature since the RH is assumed to be constant to 60% in these models as explained before.

Air-cooled cooling model (Reference Model)

$$\dot{W}_{a,ref} = \dot{W}_{des,ref} \times \left(1.319632475 - \left(0.018829979 \times T_{dry}\right)\right) \tag{24}$$

Wet-tower cooling model

$$\dot{W}_{a,wetct} = \dot{W}_{des,wetct} \times \left(1.236267002 - \left(0.00865679 \times T_{dry}\right) - \left(0.0016747 \times RH_a\right)\right) \tag{25}$$

Evaporative cooling model

$$\dot{W}_{a,evp} = \dot{W}_{des,evp} \times \left(1.382866 - \left(0.014906 \times T_{dry}\right) - \left(0.002094 \times RH_a\right)\right) \tag{26}$$

Hybrid-tower cooling model

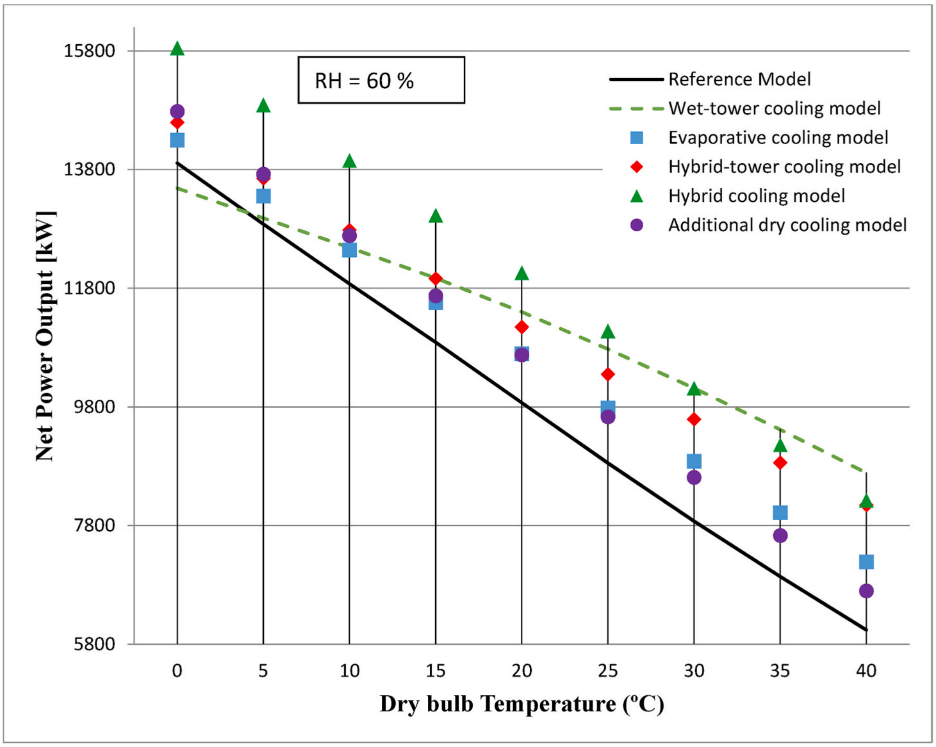

Fig. 5. The net power output data of all cooling models with respect to dry bulb temperature.

Table 4
Water consumption correction coefficients of the wet-tower cooling model.

[−]		Dry bulb temperature [°C]			
		10	20	30	40
Relative Humidity [%]	**10**	0.99003	1.15179	1.31192	1.46733
	20	0.97515	1.12635	1.2707	1.40383
	30	0.96053	1.10171	1.23146	1.34483
	40	0.94615	1.07777	1.19403	1.28916
	50	0.93203	1.05451	1.1582	1.23689
	60	0.91814	1.03189	1.12386	1.18761
	70	0.90448	1.00987	1.09079	1.14093
	80	0.89104	0.98843	1.05872	1.09655
	90	0.87782	0.96754	1.0278	1.05392
	100	0.8668	0.95098	1.00517	1.02728

$$\dot{W}_{a,hct} = \dot{W}_{des,hct} \times \left(1.315595 - \left(0.0130743 \times T_{dry}\right) - \left(0.001474 \times RH_a\right)\right) \tag{27}$$

Hybrid cooling model

$$\dot{W}_{a,hc} = \dot{W}_{des,hc} \times \left(1.27919 - \left(0.0148457 \times T_{dry}\right) - \left(0.0005128 \times RH_a\right)\right) \tag{28}$$

Additional dry cooling model

$$\dot{W}_{a,aac} = \dot{W}_{des,aac} \times \left(1.307096643 - \left(0.017975821 \times T_{dry}\right)\right) \tag{29}$$

The above equations provide the net power generation of the plant for different six cooling systems. Consequently, Fig. 5 presents the power generation comparisons for six cooling systems at the constant RH value of 60% as a demonstration. In Fig. 5, the RH value has been set to constant to provide net power generation comparisons for additional dry cooling and the reference model. As can be seen from Fig. 5, the hybrid cooling model yielded the highest net power generation when the dry bulb temperature is less than 30 °C. On the other hand, the wet-tower cooling model exhibited better performance compared to other models for the dry bulb temperature values greater than 30 °C. Since the hybrid system is a system that cools with air, it operates more effectively at low temperatures. In wet cooling systems, the latent heat also is transferred as the weather gets warmer and therefore they are less affected by this situation. Therefore, the cooling system should be determined according to the climatic conditions in the region where the geothermal power

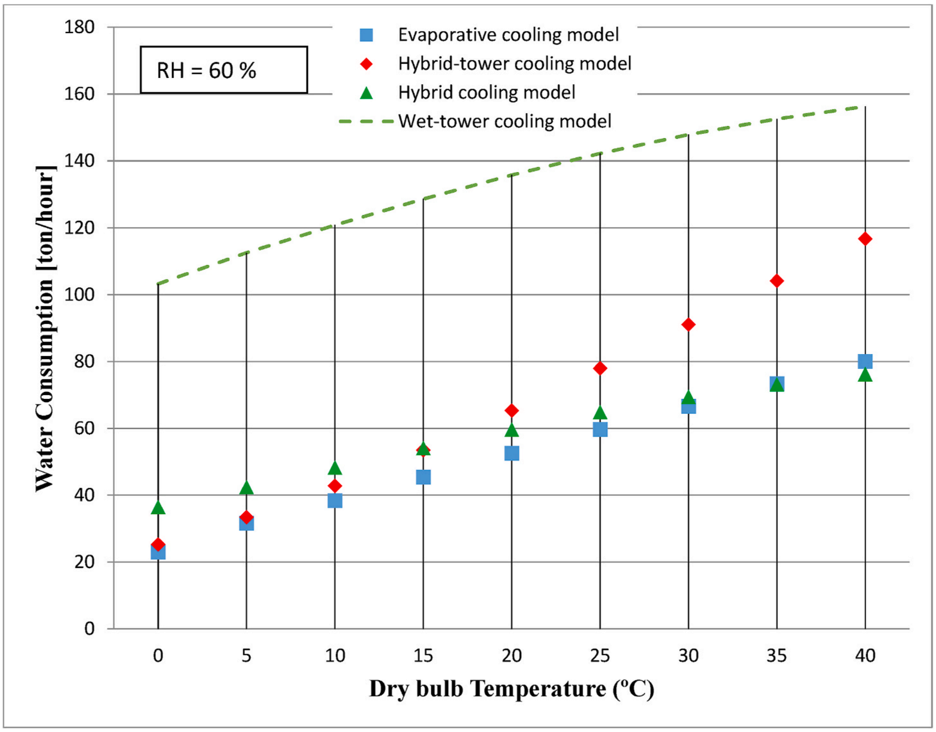

Fig. 6. The water consumption data of four cooling models with respect to dry bulb temperature.

plant is located. It should be noted that alternative cooling models showed better net power generation than the reference model for a wide range of dry bulb temperature values (5 °C < T_{dry}).

The water consumption correction coefficients for the wet-tower cooling model is provided in Table 4 as an example. The water consumption correction coefficients of other models can be calculated using Eqs. (20)–(23) with the similar procedure.

Accordingly, the water consumption equations as functions of the dry bulb temperatures and relative humidity have been obtained using the regression method based on the correction coefficients, please see Eqs. (30)–(33).

Wet-tower cooling model

$$\dot{m}_{w,wetct} = \dot{m}_{des,wetct} \times \left(0.9385845 + \left(0.01155635 \times T_{dry}\right) - (0.002543 \times RH_a)\right) \tag{30}$$

Evaporative cooling model

$$\dot{m}_{w,evp} = \dot{m}_{des,evp} \times \left(2.196934 + \left(0.044253 \times T_{dry}\right) - (0.0320872 \times RH_a)\right) \tag{31}$$

Hybrid-tower cooling model

$$\dot{m}_{w,hct} = \dot{m}_{des,hct} \times \left(1.066693 + \left(0.046796 \times T_{dry}\right) - (0.0137656 \times RH_a)\right) \tag{32}$$

Hybrid cooling model

$$\dot{m}_{w,hc} = \dot{m}_{des,hc} \times \left(0.908907 + \left(0.020344 \times T_{dry}\right) - (0.00461456 \times RH_a)\right) \tag{33}$$

The above equations provide the water consumption flow rate of the plant for different four cooling systems. Consequently, Fig. 6 presents the water consumption comparisons for four cooling systems as an example. Please note that the RH value has been set to constant in Fig. 6. As can be seen from Fig. 6, the evaporative cooling model provided the minimum water consumption when the dry bulb temperature is less than 35 °C. Since the water consumption is limited by the relative humidity of the air, it is obvious that the evaporative cooling model has lower water consumption compared to others in this condition. However, the hybrid cooling model yielded the minimum water consumption when the dry bulb temperature is greater than 35 °C. Therefore, the climatic conditions in the region where the geothermal power plant is located should be taken into account in terms of water consumption as in the net power generation.

6. Conclusions

This study set out to present the mathematical and thermodynamic analysis of a geothermal power plant having different cooling systems for various meteorological conditions. The existing binary geothermal power plant is modelled with commercial software. The mathematical equations are developed to determine the power production and water consumption data of geothermal power plants for six different cooling systems using the dry bulb temperature and relative humidity values from hourly data obtained from local meteorological stations. The following conclusions can be drawn from this study.

- The net power production and net water consumption of the plants for different cooling models can be determined accurately at different temperatures and relative humidity values using derived mathematical equations.
- The hybrid cooling model exhibited the highest net power production when the dry bulb temperature is less than 30 °C.
- The wet-tower cooling model showed better net power production performance compared to other models for the dry bulb temperature values greater than 30 °C.
- The alternative cooling models different from the reference cooling model showed better net power generation than the reference model for a wide range of dry bulb temperature values (5 °C < T_{dry}).
- The evaporative cooling model has the minimum water consumption when the dry bulb temperature is less than 35 °C.
- The hybrid cooling model exhibited the minimum water consumption when the dry bulb temperature is greater than 35 °C.
- In overall, this study proposes that the climatic conditions in the region where the geothermal power plant is located should be taken into account in terms of net water consumption and net power generation.

This study provides a general model and it is a useful study for geothermal power plant operators and engineers to calculate the net power generation and water consumption values of a geothermal power plant in any region. Furthermore, this study enables the operation periods of the hybrid cooling systems can be determined according to the water resources and climatic conditions in the region. A further study could assess operation optimization in terms of water consumption and power generation for power plants with hybrid cooling models at different climatic regions.

CRediT authorship contribution statement

Tugrul Başaran: Conceptualization, Data curation, Formal analysis, Software, Resources. **Burhanettin Çetin:** Investigation, Methodology, Project administration, Supervision. **Mehmed Rafet Özdemir:** Validation, Visualization, writing – original draft, Writing – review & editing.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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