

Full Length Research Paper

Thermo-economic study of hybrid cooling tower systems

Sonia Parsa¹, Hossein Shokouhmand^{2*}, Amirmohammad Sattari², Mohammad Nikian¹ and Fatemeh Entezari Heravi²

¹Islamic Azad University of Takestan, Iran.

²School of Mechanical Engineering, College of Engineering, University of Tehran, Tehran, Iran.

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In this paper, the effects of ambient temperature and relative humidity on performance of wet, dry, and two combined cooling systems are studied for a typical 250 MW power plant in Hamedan, Iran. Although there have been many works in background of effects of different parameters on performance of cooling towers, there is no detail analysis about different alternatives of hybrid cooling systems due to the water consumption. In this study, four alternatives were considered for the cooling system: wet, dry, and two combined wet and dry cooling systems. One of the hybrid systems demanding only half of the cells of the existing wet cooling system so has a limitation for water consumption. Another one has no limitation for utilizing all of the cells of the existing wet cooling towers and as a result does not have any limitation for water consumption. Investigating mentioned cases are significant due to the lack of water in middle-east countries. Also, by means of monthly profiles of ambient temperature, the amount of annual power loss is computed for each case. The water consumption for each case, is computed as well. Finally, the best alternative was determined by computing both the capital and annual costs, and annual water consumption.

Key words: Wet cooling tower, dry cooling tower, hybrid cooling tower, power plant, economic evaluation.

INTRODUCTION

Cooling towers are among the most important components of power plants. They are utilized to reduce the augmented temperature of water in power plants and return the cold water into the main plant cycle. Cooling agent in these towers can be either air or water that, with either direct or indirect contact, reduces the temperature of hot water coming from the condenser. Cooling towers are classified into three different types based on their

heat transfer approach, wet, dry, and combined cooling towers. Wet cooling towers operate upon evaporative cooling. The working fluid and the evaporated fluid are the same in these towers. Dry cooling towers operate with a surface that intercepts the working fluid from the ambient air, which is used for convective heat transfer. The dry cooling towers do not use evaporation as a means for cooling; therefore, they consume much less

*Corresponding author. E-mail: hshokoh@ut.ac.ir.

water. According to the water shortage problem in Iran, replacement of wet cooling towers with dry ones seems an attractive approach. So, the influence of different parameters on the performance of cooling towers is a major issue for designers.

Among different parameters which affect the performance of cooling towers, ambient temperature and relative humidity are the most influential. So, there are many works done in this background. In 1946, one of the first investigation of performance of cooling towers has been conducted by Simpson and Sherwood (1946). They found constants for evaluating coefficient of mass transfer. Berman was the first one who described how the Log-Mean Enthalpy Method (LMED) may be applied to cooling tower design (Berman, 1961). Besides, Moffat (1966), for the first time derived the ε -NTU equation for a counter-flow cooling tower. Furthermore, Jaber and Webb (1989) developed the ε -NTU design method for cooling towers. They presented sample calculations for counter and cross-flow cooling towers. Also, the authors summarized the LMED method introduced by Berman, and show that this is totally consistent with the ε -NTU method. Bernier (1995) reviewed the heat and mass transfer processes in cooling towers. He also presented a practical correlation for evaluating water evaporation rate affiliated with mass transfer at the water-air interface. In other research Söylemez (2004), for the first time optimized the water to air mass ratio for counter flow cooling towers that included the ambient pressure and average of tower and basin temperature of cooling towers in detail. Muangnoi et al. (2008) investigated the influence of temperature and humidity on performance of counter-flow wet cooling towers by using exergy analysis and finally by utilizing optimization methods, best temperature and humidity for achieving the highest efficiency were computed. Also, Lucasa et al. (2010) due to disadvantages of the moisture that goes out from cooling towers, worked on the effect of psychrometric environmental conditions on the amount of the aforementioned moisture. They concluded that by increasing dry-bulb temperature, the amount of the moisture escaping from the tower declines. In another research, Papaefthimiou et al. (2012) studied thermodynamic effect of ambient temperature on specifications of the cooling towers. They resulted that by decreasing inlet wet-bulb temperature, the temperature will be more reduced in the tower; as well as the amount of waste of water. In other work He et al. (2014) investigated the influence of environmental conditions and water flow on performance of cooling towers with pre-cooled air. They found that influence of water flow on performance of cooling towers is negligible. Also, it was concluded that employing this type of cooling towers, under the conditions of high ambient temperature and low humidity is very helpful. And finally the evaporation rate of water in this type of towers is lower than wet cooling towers. Thereafter Ma et al. (2015) surveyed the effect of both the ambient

temperature and cross wind on efficiency of cooling towers. They concluded that effect of outlet water temperature has non-linear and linear relationship with the wind velocity and the ambient temperature respectively. Finally, He et al. (2015) performed experimental study on application of two trickle media for inlet air pre-cooling of natural draft cooling towers. They optimized size of trickles which has the less pressure drop and the highest performance. Another major parameter that affect performance of cooling towers, especially natural-draft cooling towers, is wind that widely considered in many works (Harnach and Niemann, 1980; Dachun and Chenxin, 1987; du Preez and Kröger, 1995; Su et al., 1999; Al-Waked and Behnia, 2004; Ke and Ge, 2014). Since the mean wind speed of the target site of the current research is low, and dry cooling system is one of the alternatives of this research, effect of wind speed is not considered.

Another topic that attracted many attentions is to utilizing saline water in cooling tower systems. For instance, Kinnon et al. (2010) showed that NaCl is the main salt in the saline water from coal-bed methane production. Sadafi et al. (2015a) monitored the saline water droplet size at different ambient conditions using microscope digital camera. They showed that for 500 μm radius droplets with 3 and 5% initial NaCl mass concentrations the net energy required to evaporate the droplet falls by 7.3 and 12.2%, respectively (compared to a pure water droplet). Also, in a subsequent study, Sadafi (2015b) investigates the performance of saline water, compared to pure water in spray cooling and demonstrates the existence of several advantages.

In the current research, four cases are considered for the cooling system of a typical power plant, with a nominal capacity of 250 MW placed in Hamedan city in Iran. These cases are utilizing wet cooling towers, dry cooling towers and two combined wet and dry systems (hybrid systems), which in one case there is a limitation of using wet cooling towers due to lack of water sources in the region, while there is no limitation in another case. Utilizing dry and combined cooling systems are important issues due to lack of water in the world, especially in Iran. Hence, in the present work, at first, thermodynamics of wet and dry cooling towers are studied. Then, the effects of ambient temperature and relative humidity on the performance of wet and dry cooling towers are investigated. Using profiles of ambient temperature and relative humidity of Hamedan power plant, diagrams relevant to the performance of wet, dry and combined cooling towers are extracted. Finally, four mentioned cases are compared by the economical aspect and the best one is determined.

PRESENT MODEL

In this paper, four alternatives are considered for cooling

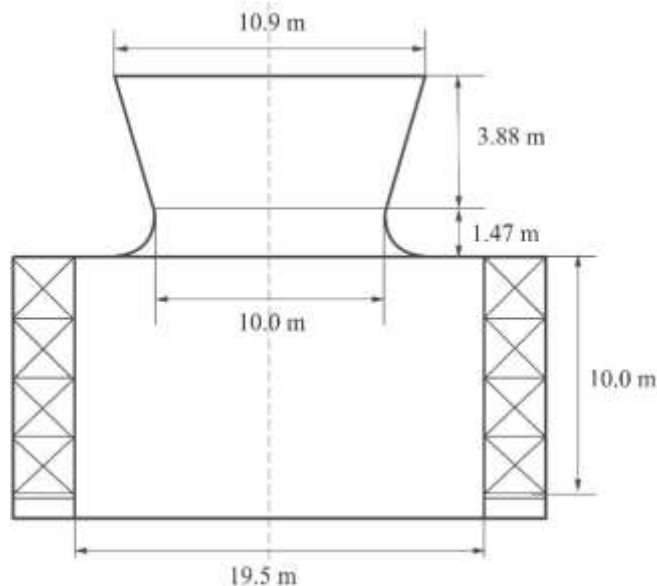


Figure 1. Scheme of the present wet cooling tower.

Table 1. General specifications in the design of the wet cooling tower.

Parameter	Value
Number of cells	12
Water inlet temperature	38.2°C
Water outlet temperature	27.8°C
Wet bulb temperature	14.5°C
Maximum air flow rate	889150 (cfm/cell)
Quantity of make-up water (at 35% RH)	675.6 (m ³ /h)

system of a typical power plant with 250 MW nominal capacity. These alternatives are

- A) Wet cooling towers
- B) Dry cooling towers
- C) Wet/dry system with 50% capacity of wet cooling towers
- D) Wet/dry system which in any conditions power plant work with full load capacity.

Case A

This case contains a series of wet cooling systems including twelve wet cooling towers. Dimensions of present wet cooling towers and specifications are presented in Figure 1 and Table 1 respectively.

Case B

In this case, cooling system contains a dry cooling tower.

Geometry and general specifications of dry cooling tower of the current work are illustrated in Figure 2 and Table 2 respectively. It should be noted that there is no auxiliary system (such as peak coolers) in studied dry cooling towers.

Case C

In this case, as shown in Figure 3, cooling water, first, enters a dry cooling tower, and, next, enters a shell and tube heat exchanger. Half of the present wet cooling towers (6 cells) supply the cooling water of the heat exchanger. So, in this case, limitation of water consumption, due to presence of wet cooling towers, confines the accessibility to the full capacity of wet cooling system (12 cells). So it is expected that, in this particular case, power losses occur at high ambient temperatures. It should be noted that the specifications of wet and dry cooling towers in this case are similar to those of cases A and B.

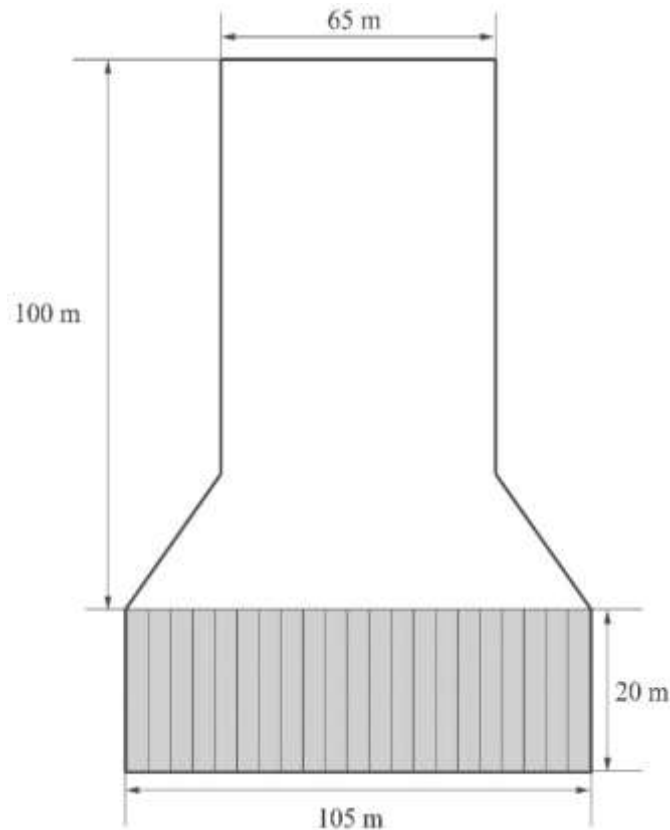


Figure 2. Geometry model of the dry cooling tower of the work.

Table 2. General specifications of the dry cooling tower.

Parameter	Value
Number of deltas	120
Height of deltas	20 m
Tower height (from base)	120 m
Base diameter of tower	105 m
Upper diameter of tower	65 m
Cooler surface	8,000 m ²
Cooling capacity	28,000 m ³ /h

Case D

Difference of this case with case C is that the limitation of make-up water is not considered. So, in case D, at high ambient temperature conditions, more wet cooling towers come into the cooling system and compensate amount of power losses which happens in case C. It should be noted since the dry cooling tower is present in this case, all of the wet cooling towers will not be utilized even at the high temperatures. So, the amount of water usage in this case is lower than case A. The number of cells utilized in any ambient temperature is dependent on the

requisite cooling capacity to achieve to the full load capacity of the power plant (250 MW). It means that in higher ambient temperature more cells will be utilized in order to achieve to the desired cooling capacity.

MATHEMATICAL MODELS

Dry cooling tower modeling

Governing equations that are utilized in this work for thermodynamic analysis of dry cooling tower are heat transfer in heat exchangers and energy conservation. A brief description of mentioned governing equations is presented in the following.

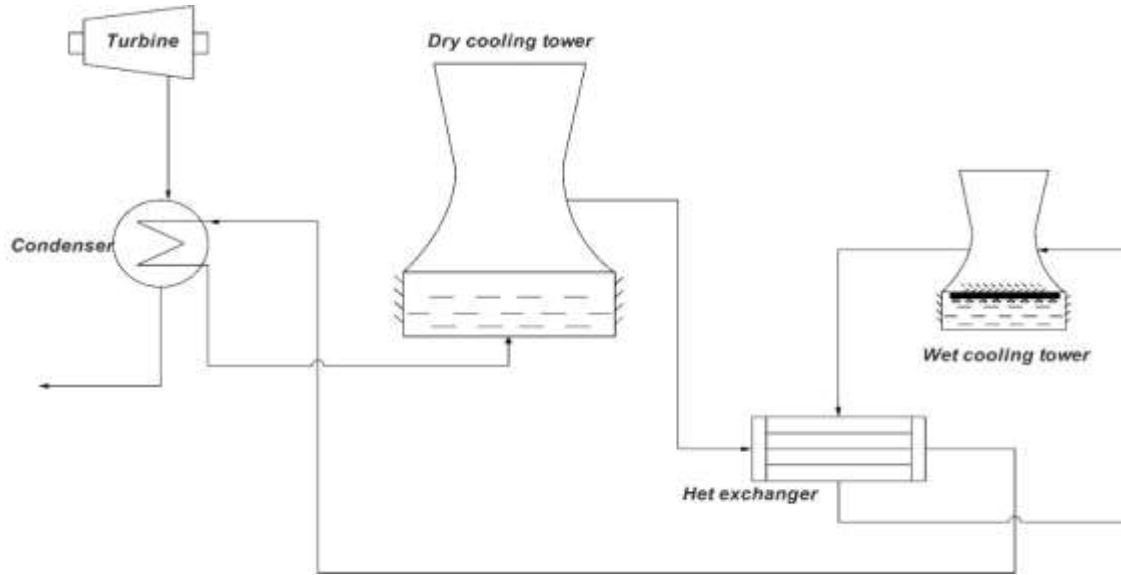


Figure 3. Schematic of the current combined system.

Heat transfer equations

It is common to use Forgo T60 type heat exchangers in dry cooling towers. Total heat transfer coefficient includes three different heat transfer coefficients including convection of internal water flow, conduction from tubes, and convection between air and tubes. So, Equation (1) can be considered for determining the total heat transfer coefficient.

$$\frac{1}{U} = \frac{1}{h_{w,i}A_i} + \frac{1}{\eta h_{a,f}A_f} + \frac{\delta}{KA_t} \tag{1}$$

Calculating heat transfer of dry cooling towers based on front surface is a common approach. So, Equation (1) can be rewritten as Equation (2).

$$\frac{1}{U_c(A_f/A_{fr})} = \frac{1}{h_{w,i}(A_i/A_{fr})} + \frac{1}{\eta h_{a,f}(A_f/A_{fr})} + \frac{\delta}{K(A_t/A_{fr})} \tag{2}$$

Considering that dimensions of Forgo T60 heat exchangers are specified, in order to simplify equations, Equations (3) to (5) are assumed,

$$h_w = h_{w,i} \left(\frac{A_i}{A_{fr}} \right) \tag{3}$$

$$h_a = \frac{1}{\left[\frac{1}{\eta h_{a,f} \left(\frac{A_f}{A_{fr}} \right)} + \frac{\delta}{K \left(\frac{A_t}{A_{fr}} \right)} \right]} \tag{4}$$

$$U = U_c \left(\frac{A_f}{A_{fr}} \right) \tag{5}$$

So:

$$\frac{1}{U} = \frac{1}{h_w} + \frac{1}{h_a} \Rightarrow U = \frac{h_w h_a}{h_w + h_a} \tag{6}$$

In above equations, total heat transfer was considered for clean tubes, and effect of fouling has not been included. For considering fouling, total heat transfer can be modified as Equation (7).

$$\frac{1}{U_{dirty}} = \frac{1}{U_{clean}} + R_f \tag{7}$$

As a result,

$$U_{dirty} = \frac{U_{clean}}{U_{clean} R_f + 1} \tag{8}$$

R_f is sediment coefficient. For Forgo heat exchangers in dry cooling systems R_f is considered to be $0.00009 \frac{m^2c}{w}$ (Know HowDocuments, 1984).

Combining Equations 6 and 8, total heat transfer can be calculated through Equation 9.

$$U = \frac{h_w \cdot h_a}{h_w + h_a + R_f \cdot h_w \cdot h_a} \tag{9}$$

In order to calculate water and air side heat transfer coefficients (h_w , h_a), Equation (10) has been presented by the manufacturer of Forgo T60 heat exchangers (Know HowDocuments, 1984)

$$\begin{aligned} h_w &= [317.3 + 2.82(T_{wi} + T_{wo})]Q_{ow}^{0.8} \\ h_a &= 1180 \left[\frac{\dot{m}_a}{A_f} \left(\frac{\rho_{oa}}{\rho_{am}} \right)^{0.64} \right]^{0.515} \end{aligned} \tag{10}$$

ρ_{ma} is average air specific mass through a heat exchanger that is equal to:

$$\rho_m = \frac{\rho_{a,i} + \rho_{a,o}}{2} \quad (11)$$

Energy conservation

Forgoing heat losses in plumbing and water transfer route from the condenser to the tower, heat released from vapor to the cooling liquid in condenser is equal to the heat released from cooling tower water. The released heat from cooling water is equal to the absorbed heat by the passing air from tower. So,

$$\dot{Q}_c = \dot{m}_{w,t} c_{p,w} \Delta T_w = \dot{m}_{a,t} c_{p,a} \Delta T_a \quad (12)$$

\dot{Q}_c is released heat from condenser which based on number of deltas and pass flow rate from a column can be written as follows:

$$\dot{Q}_c = 2N_d \dot{m}_w c_{p,w} \Delta T_w = 2N_d \dot{m}_a c_{p,a} \Delta T_a \quad (13)$$

Wet cooling tower modeling

The control volume of a counter flow cooling tower is illustrated in Figure 4. The most important assumptions are summarized as follows:

- 1) Heat and mass transfer occur only in the perpendicular direction of the flows.
- 2) Losses of heat and mass transfer through tower walls are neglected.
- 3) Mass transfer coefficient is constant all over the tower.
- 4) The water temperature distribution is uniform in each cross section.
- 5) The tower cross section is constant in every height.
- 6) The Louis factor is considered as a variable in the modeling.

The conservation of mass equation for the entering water into the air in the steady state is indicated by Equation (14).

$$\dot{m}_a W + h_D A_V dV (W_{s,w} - W) = \dot{m}_a \left[W + \left(\frac{\partial W}{\partial V} \right) dV \right] \quad (14)$$

In which dV is the control volume element which can be seen in Figure 4.

Equation (14) is simplified as,

$$\dot{m}_a dW = h_D A_V dV (W_{s,w} - W) \quad (15)$$

The general energy balance equation of the moist air could be expressed as Equation (16).

$$\dot{m}_a h + h_c A_V dV (T_w - T) + h_D A_V h_{f,g,w} dV (W_{s,w} - W) = \dot{m}_a \left[h + \left(\frac{\partial h}{\partial V} \right) dV \right] \quad (16)$$

After simplification, Equation (16) would be rewritten as

$$\dot{m}_a dh = h_c A_V dV (T_w - T) + h_D A_V h_{f,g,w} dV (W_{s,w} - W) \quad (17)$$

The energy balance for water could be expressed as a function of heat (h_c) and mass (h_D) transfer coefficients,

$$\left[\dot{m}_w + \left(\frac{\partial \dot{m}_w}{\partial V} \right) dV \right] \left[h_{f,w} + \left(\frac{\partial h_{f,w}}{\partial V} \right) dV \right] = \dot{m}_w h_{f,w} + h_c A_V dV (T_w - T) + h_D A_V h_{f,g,w} dV (W_{s,w} - W) \quad (18)$$

The simplified equation would be,

$$\dot{m}_w dh_{f,w} + \dot{m}_a dW h_{f,w} = h_c A_V dV (T_w - T) + h_D A_V h_{f,g,w} dV (W_{s,w} - W) \quad (19)$$

By inserting the Louis factor, $Le_f = \frac{h_c}{h_D c_{p,a}}$ in Equation 19, and a

little bit simplification, the Equation (20) is obtained.

$$\begin{aligned} \dot{m}_w dh_{f,w} + \dot{m}_a dW h_{f,w} &= h_D A_V dV [Le_f c_{p,a} (T_w - T) \\ &+ h_{f,g,w} (W_{s,w} - W)] \end{aligned} \quad (20)$$

Combining the aforementioned equations, Equation (21) is derived.

$$\frac{dh}{dW} = \frac{(Le_f c_{p,a} (T_w - T) + h_{f,g,w} (W_{s,w} - W))}{W_{s,w} - W} \quad (21)$$

Again, a simplification would result in derivation of Equation (22).

$$h_{s,w} - h = c_{p,a} (T_w - T) + h_g^0 (W_{s,w} - W) \quad (22)$$

By assuming constant specific heat capacity for air in the preceding equations, the Equation (23) is obtained.

$$h_{s,w} - h = c_{p,a} (T_w - T) + h_g^0 (W_{s,w} - W) \quad (23)$$

Ultimately, inserting Equation 23 in Equation 22 and succeeding simplifications will result in the Equation 24.

$$\frac{dh}{dW} = Le_f \frac{(h_{s,w} - h)}{(W_{s,w} - W)} + (h_{f,g,w} - h_g^0 Le_f) \quad (24)$$

Equation 25 indicates the steady state energy balance between the water and air.

$$\dot{m}_a dh = \dot{m}_w dh_{f,w} + \dot{m}_a dW h_{f,w} \quad (25)$$

Be careful that the last term in Equation 25 indicates the impact of water evaporation on the energy equation and \dot{m}_w represents the mass flow rate of water in any altitude of the tower. Commonly, because of the low percentage of water vapor in the air, the diminution in water flow rate is neglected in the modeling (ASHRAE, 1975) and $\dot{m}_w = \dot{m}_{w,in} = \dot{m}_{w,out}$. However, in the present work, in order to increase the accuracy of evaluating losses, these changes have been considered.

$$\dot{m}_a dh = (\dot{m}_{w,in} - \dot{m}_a (W_{out} - W)) dh_{f,w} + \dot{m}_a dW h_{f,w} \quad (26)$$

It is clear that $dh_{f,w} = c_{p,w} dT_w$. By substituting this in Equation (26) and then in Equation 24, Equation 27 is achieved.

$$\left(\dot{m}_{w,in} - \dot{m}_a (W_{out} - W) \right) \frac{dh_{f,w}}{dW} = \frac{dh}{dW} - h_{f,w} = \frac{dh_{f,w}}{dW} = \frac{1}{(\dot{m}_{w,in} - \dot{m}_a (W_{out} - W))} \left[Le_f \frac{(h_{s,w} - h)}{(W_{s,w} - W)} + (h_{f,g,w} - h_g^0 Le_f) - h_{f,w} \right] \quad (27)$$

Finally, Equations 28 to 30 will be used for the simulation of the cooling tower core:

$$\dot{m}_a dW = h_D A_V dV (W_{s,w} - W) \quad (28)$$

$$\frac{dh}{dW} = Le_f \frac{(h_{s,w} - h)}{(W_{s,w} - W)} + (h_{f,g,w} - h_g^0 Le_f) \quad (29)$$

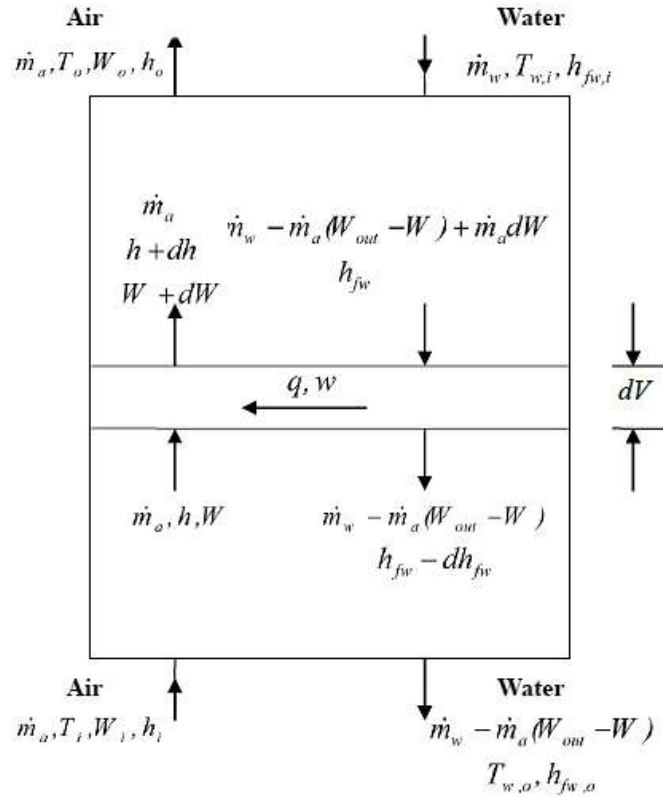


Figure 4. Control volume for conservation of mass and energy balance in a counter flow cooling tower.

$$\frac{dh_{f,w}}{dW} = \frac{1}{(\dot{m}_{w,in} - \dot{m}_a(W_{out} - W))} \left[Le_f \frac{(h_{s,w} - h)}{(W_{s,w} - W)} + (h_{fg,w} - h_g^0 \cdot Le_f) - h_{f,w} \right] \quad (30)$$

For evaluating the temperature distribution, the Equation (31) is used.

$$dT_w = - \frac{1}{c_w} \frac{(dh - dW \cdot h_{f,w})}{\dot{m}_w - (W_{out} - W)} \quad (31)$$

In the Equations (28) to (31), the coefficient of mass transfer is unknown. This problem is often resolved using Equation (32).

$$\frac{h_D \cdot A_V \cdot V}{\dot{m}_{w,in}} = c \left(\frac{\dot{m}_{w,in}}{\dot{m}_a} \right)^n \quad (32)$$

In which n and c are the experimental coefficients used for the tower design. Braun has fitted the curve of the cited n and c . This work has been executed based on Simpson and Sherwood (1946) measurements, for different tower designs operating under design conditions (Braun et al., 1989). The values of c and n in present model are considered 1.405 and -0.727 respectively based on Braun work.

By multiplying both sides of Equation (32) in $\left(\frac{\dot{m}_{w,in}}{\dot{m}_a} \right)$ and considering the definition of Number of Transport Units (NTU), the experimental value of NTU is achieved as Equation (33) represents.

$$NTU = \frac{h_D \cdot A_V \cdot V}{\dot{m}_a} \Big|_{em} = c \left(\frac{\dot{m}_{w,in}}{\dot{m}_a} \right)^{n+1} \quad (33)$$

The effectiveness of the cooling tower is defined as the ratio of the actual energy to the maximum possible energy and is calculated using Equation (34).

$$\varepsilon = \frac{h_{out} - h_{in}}{h_{s,w} - h_{in}} \quad (34)$$

Also, it is necessary to define the non-dimensional temperature difference, or temperature ratio in the cooling tower literature, as the ratio of the actual loss to the maximum value through Equation (34).

$$R_{ct} = \frac{T_{w,in} - T_{w,out}}{T_{w,in} - T_{wb,in}} \quad (35)$$

In extracting Equations (14) to (34), it is assumed that there is no resistance against the heat flow in the air-water interface. In other words, the common interface temperature is assumed to be equal to the water bulk temperature. In this way, all of the terms in Equations (14) to (34) which have the subscripts (s,w) are substituted by (s,int) . Jabir and Webb assumed that T_w is approximately near to $T_{int} + 0.5$ (Jaber and Webb, 1989).

Figure 5 illustrates both enthalpies of the saturated water-air mixture and the operating line of the tower as a function of water temperature. By assuming a low difference between the values of $h_{s,w}$ and $h_{s,int}$ on the saturated line as a linear function, Equation (36) could be derived.

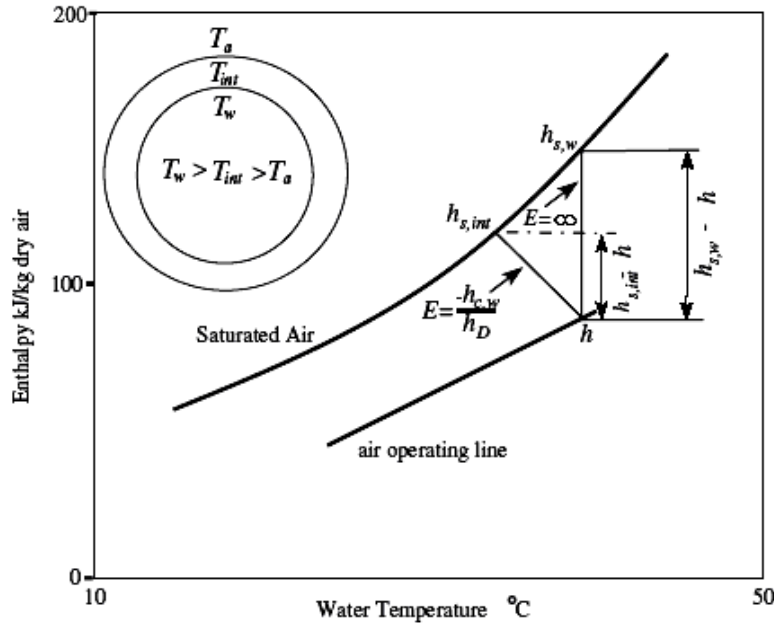


Figure 5. Water operating line on the temperature-enthalpy diagram, indicating the effect of $E = -h_{c,w}/h_D$ on the saturated air enthalpy.

$$h_{s,w} - h = h_{s,w} - h_{s,int} + E(T_w - T_{int}) \tag{36}$$

The slope E is calculated utilizing the Equation (37).

$$E = -\left(\frac{h_{c,w}}{h_D}\right) \tag{37}$$

Equation (37) could be used to determine the temperature at the interface. For large values of E, interface and bulk temperatures are approximately equal.

Based on what has been said so far, the term of the average mass transfer coefficient ($h_D A_V$), could be calculated using experimental results of Simpson and Sherwood (1946). In the present paper, besides the experimental exit temperatures, the water mass transfer coefficient, k'_a , and the total heat transfer coefficient, K'_a , have been calculated too. The two aforementioned coefficients could be correlated through Equation (38)

$$\frac{h_{s,w} - h}{h_{s,int} - h} = -\frac{k'_a}{K'_a} \tag{38}$$

By assuming that the interface and bulk temperatures are equal, the last two coefficients also would have the same value. The experimental data will hand in the value of the average mass transfer coefficient, K'_a .

RESULTS AND DISCUSSION

In this work, effects of ambient temperature and relative humidity on the performance and water consumption of a power plant for four mentioned cooling systems are investigated for case study of Hamedan Power plant. Also, cases are economically compared and the best choice is distinguished.

Effect of ambient temperature

The most important parameter that influences performance of cooling towers is the ambient temperature. Changes of ambient temperature cause changes in heat transfer rate of towers. It is clear that the increase of the ambient temperature generally causes an increase in condenser temperature, and, as a result, reduces the output power of the turbine. Furthermore, performance of dry cooling towers is more dependent on ambient temperature in comparison with wet cooling towers, since heat release mechanism in wet cooling towers are mostly through evaporation.

In order to determine the amount of power loss due to changes in ambient temperature for each case in Hamedan power plant, the following steps have been taken. First, the generated power in a 250 MW power plant was computed as a function of ambient temperature based on the equations mentioned in the theory modeling. The range of temperature is considered between 0 and 40°C. Figure 6 illustrated output power of the power plant versus ambient temperature for four cases.

As shown in Figure 6, the augmentation of the ambient temperature leads to the diminution of the plant's power generation, since the first causes the diminution of the dry cooling tower effectiveness and then the augmentation of the exit water temperature. This augmentation in temperature makes the turbine inlet pressure higher and, as a result, lowers its generated power. But as shown in this figure, ambient temperature does not affect

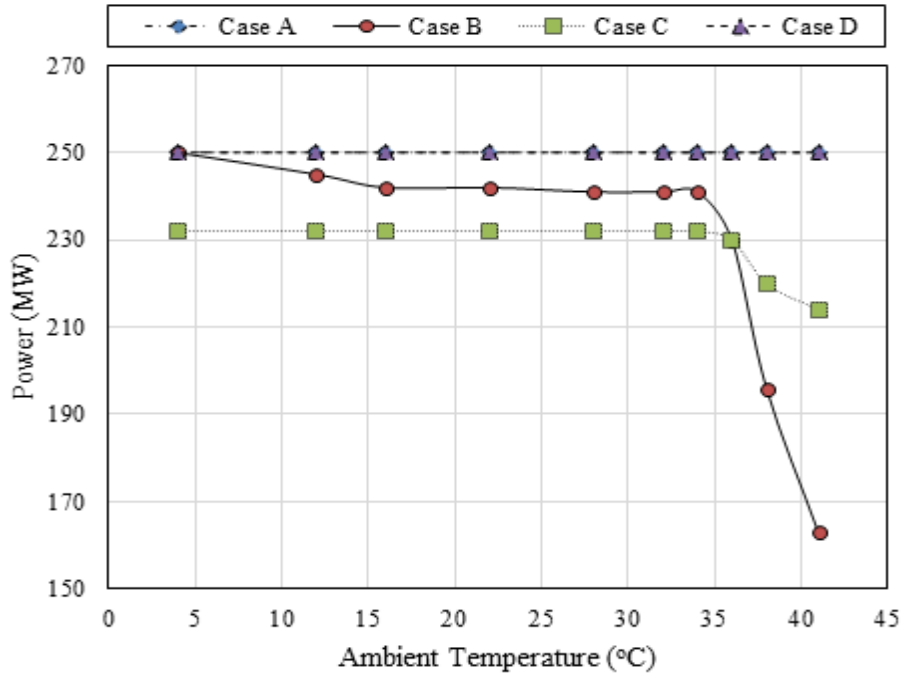


Figure 6. Power generations for mentioned cases as a function of ambient temperature.

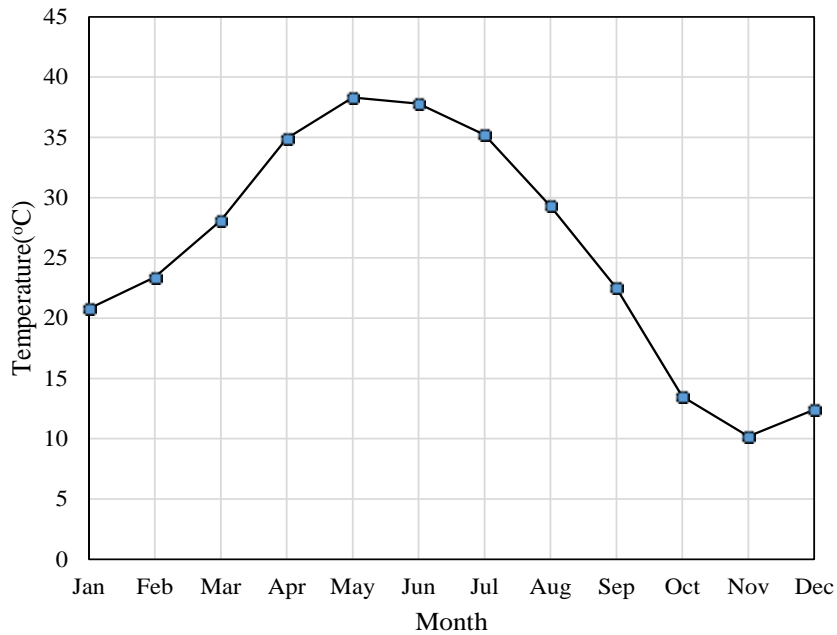


Figure 7. Monthly temperature of Hamedan Power plant.

performance of cases A and D, because of the presence of wet cooling towers.

As mentioned before, case study of current paper is Hamedan Power plant in Iran. So for calculating the amounts of the annual energy loss due to the ambient temperature, diagram of temperature for Hamedan Power

plant versus month was collected (Figure 7). Based on Figures 6 and 7, the annual amount of power produced and that of any month can be calculated.

By utilizing Figures 6 and 7, energy production of the power plant for each case versus month can be extracted which is shown in Figure 8. As shown in this figure, there

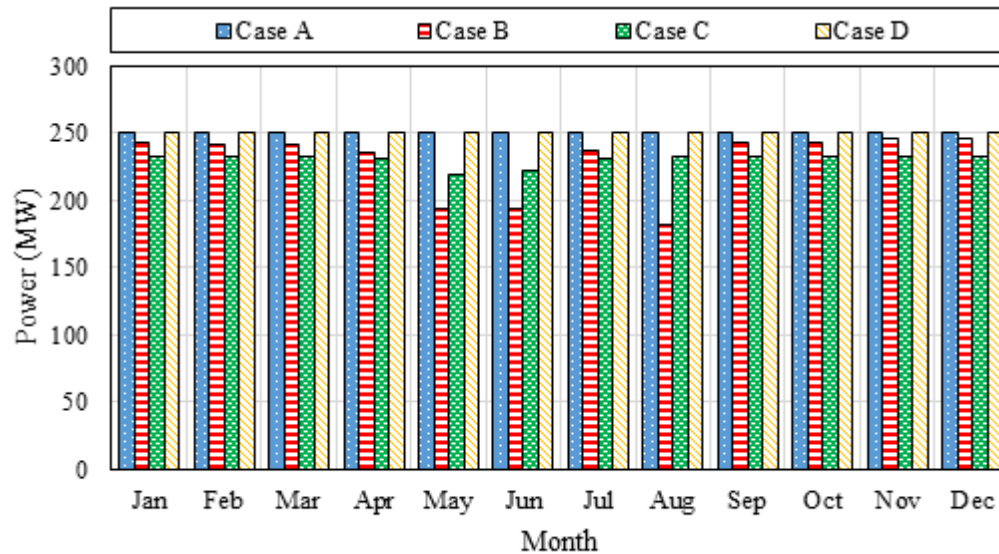


Figure 8. Monthly power production due to the ambient temperature of Hamedan power plant for four mentioned cases.

Table 3. Comparison between energy losses during a year of different methods due to ambient temperatures.

Case	Energy loss during a year (MWh)	Annual investment loss (\$)
A	0	0
B	186,624	2,799,360
C	174,240	2,613,600
D	0	0

is no power loss in cases A and D since full capacity of wet cooling towers is achieved and evaporation heat transfer mechanism can supply the amount of heat transfer required for full capacity power generation of the power plant. After these cases, case C has the most power generation due to its utilization of wet cooling towers. Finally, dry cooling towers produce the higher back pressure of turbine that, as a result, has more power loss in comparison with other alternatives.

So the amount of annual power loss due to the ambient temperature is calculated. By utilizing amounts of power generation at each case, as well as regarding the cost of electricity in Iran that is 15\$ per MWh, the amount of total annual investment loss can be calculated too. These results are shown in Table 3.

Effect of humidity

In wet cooling towers, by increasing relative humidity, the efficiency of the system decreases due to mechanism of cooling system which is mainly arising from evaporation. So, the more decreases in relative humidity of ambient

air, the more capability of evaporation that causes increasing of cooling capacity. But in dry cooling towers, because of the convective heat transfer mechanism, relative humidity does not have noticeable effect and is considered as a second factor (Mehdi, 2000)

In order to determine the power loss in power plants due to changes in relative humidity, first of all, the generated power in a 250 MW power plant was computed as a function of relative humidity due to the relations mentioned in the theory modeling. The range of relative humidity is considered between 0 and 1. Figure 9 illustrated power of power plant versus relative humidity for four cases.

As illustrated in Figure 9, and as mentioned before, relative humidity does not have noticeable effect on performance of dry cooling towers. On the other hand, despite of the presence of wet cooling towers in other alternatives, since present wet cooling towers are including forced-draft fans, lack of heat transfer can be compensated by changing pitch angle of the fans and also more flow rate of the cooling water. So, in these cases power loss is negligible. On the other word, however relative humidity has noticeable effect on

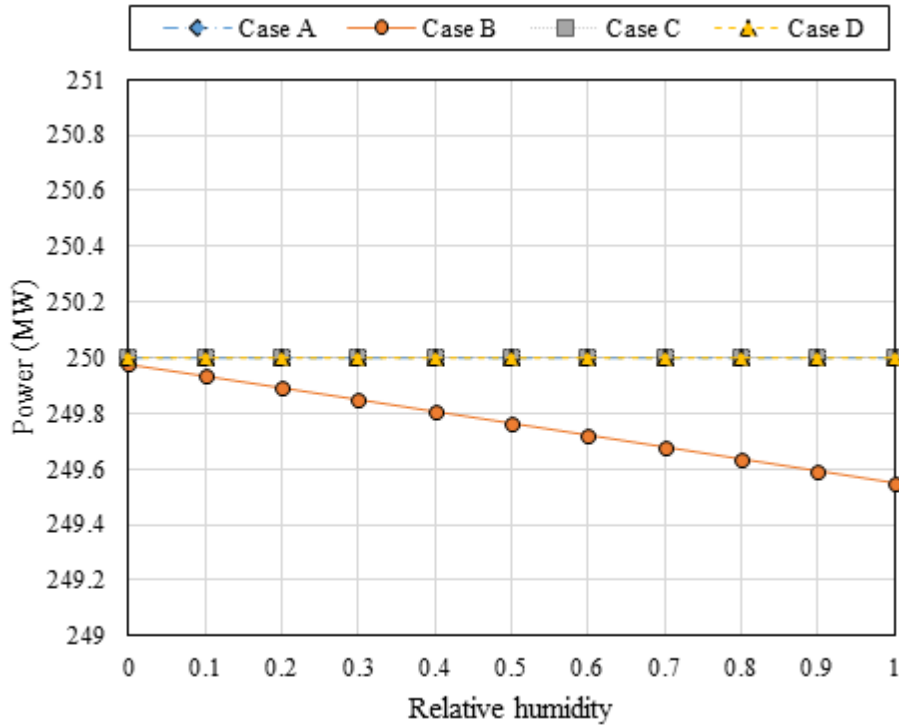


Figure 9. Power generations as a function of relative humidity.

Table 4. Comparison amounts of water consumption between different methods.

Case	Total annual water demand (m ³ /y)	Costs of water consumption (\$)
A	11,000,000	1,320,000
B	850,000	102,000
C	1,375,000	165,000
D	2,750,000	330,000

performance of wet cooling towers, in present power plant, wet cooling towers have been designed in a way that they can overcome to the relative humidity by increasing flow rate of the cooling water and pitch angle and speed of the fans. This fact was observed directly from data sheets of the power plant during the year. But, as discussed before, by increasing relative humidity more circuit water is needed in order to overcome power loss and this needs more cooling water in hot days of the year. This is going to be considered in estimating the amount of water consumption.

Water consumption

Using thermodynamic analysis for Hamedan power plant in Iran, average annual amounts of water consumption for four alternatives was calculated and is shown in Table 4. Amounts of water consumption are different in four

cases due to the structure of cases. It is clear that dry cooling towers have the least amount of water consumption and are suitable for places which confront the problem of water shortage, even though performance and power generation is lower than other cases. Wet cooling towers have the most water demand; but their performance is the most appropriate for regions that have enough sources of water. It was an interesting comparison with combined cooling system cases. In fact, these cooling systems can be utilized in places that have midrange sources of water and also need high performance and power generation. Case D has more annual water demand than case C. Instead, approximately there is no power loss in this case. On the other hand, in comparison with wet cooling towers, it has lower annual water demand and has the same performance, but need more capital cost due to the utilization of dry cooling towers. On the other side, considering that the water price is 0.12\$ per cubic meter in industrial

Table 5. Total investment cost amounts of different methods.

Case	Capital cost (\$)	Annual running cost (\$)
A	38,514,149	3,999,400
B	18,373,851	2,781,400
C	56,888,000	2,844,400
D	56,888,000	3,174,400

applications in Iran, amount of water consumption cost as running cost can be computed for each case that is shown in Table 4.

Economic analysis

Utilizing mentioned descriptions and, also, an exact investigation of required utilities and instruments of each case, amounts of capital and annual running costs of each case was computed. It should be reminded that capital cost of cases C and D are similar due to similarity of structures and just running costs are different.

Cost details of this project consists of the expenditure of purchasing requirement equipment and building structures as capital cost, operating and maintenance costs and water demand as running cost. In addition, power production sales are considered as income of the work. Capital and annual running costs of different alternatives (for building and operating of cooling systems) are listed in Table 5.

In this study, three commonly methods are applied for economic evaluation: Net present value (NPV), internal rate-of-return (IRR) and normal payback method (NP). A brief mathematical model of them will be presented.

The net present value (NPV) method recognizes the surplus of benefits over costs, where all measures are reduced for their time value. (If costs exceed benefits, net damages result). Also, The NPV method is often called the net present worth or net savings method. When this method is applied for measuring a cost-reducing investment, the cost savings are the benefits, and it is often called the net savings (NS) method. NPV from an investment, such as an investment is calculated by following equation:

$$NPV_{A1:A2} = \sum_{t=1}^N \frac{B_t - C_t}{(1 + d)^t} \quad (39)$$

where $NPV_{A1:A2}$ is NB, for example present value benefits (savings) net of present value costs for alternative A1 as compared with alternative A2, B_t is benefits in year t , which may be specified to contain energy savings, C_t is costs in year t related with alternative A1 as compared with a mutually exclusive alternative A2, and d is the reduce rate.

The internal rate-of-return (IRR) method solves for the discount rate for which dollar savings are just equal to dollar costs over the analysis duration; that is the rate for which the NPV is zero. This discount rate is the rate of Payback the investment. It is compared to the investor's minimum plausible rate of return to specify whether the investment is favorable. Unlike the preceding three techniques, the internal rate of return does not call for the inclusion of a prespecified discount rate in the calculation; rather, it solves for a discount rate.

The rate of return is usually computed by a process of trial and error, by which diverse compound rates of interest are applied to discount cash flows until a rate is found for which the NPV of the investment is zero. The method has the following procedure: (1) Compute NPV using Eq. (39), except substitute a trial interest rate for the discount rate, d , in the equation. A positive NPV means that the IRR is greater than the trial rate; a negative NPV means that the IRR is less than the trial rate. (2) Based on the information, try another rate. (3) By a series of iterations, find the rate at which NPV is zero.

In this study, economic analysis was conducted based on following assumptions in order to render the analysis more traceable:

- i) Construction time of the project is considered 2 years.
- ii) Operation time of the project is considered 20 years.
- iii) Inflation is assumed 15% based on reports of Central Bank of Iran.
- iv) Tax is considered to be 5% of the benefits.
- v) Load factor is assumed to be 0.65 for all cases.

The above assumptions are based on typical value in Iran. Economic analysis results for different alternatives are listed in Table 6.

As shown in Table 6, the best choice from the economic aspect is case B, that is, utilizing dry cooling towers. On the other hand, negative NPV in case C shows that this case does not have economic justification. But it should be noted that although dry cooling towers have the most economic benefit, they have relatively large power loss, especially in summer times. But, on the other hand, they have the least water consumption among other alternatives. So, it should be considered that based on which one of the following parameters, power production or water consumption, is more important in a specific region, the final decision will be different. As

Table 6. Economic analysis results.

Case	IRR (%)	NPV (\$)	NP (year)
A	34.44	24,320,072	3.95
B	48.28	17,516,140	3.71
C	20.09	-22,570,974	5.98
D	21.02	3,431,506	5.78

a matter of fact, if a region is faced with lack of water sources, like case study of current paper, Hamedan Power plant, it is preferable to use dry cooling towers alone. But if national power grid needs full load capacity of the power plant, case D, that is, utilizing combined cooling system that generates full load power, is more suitable. For places near water sources like lakes and rivers, certainly case A, that is, using wet cooling towers, will be the best choice.

Conclusion

The cooling towers are among the most crucial components of every thermal power plant. Their performance directly affects the outlet power and the plant efficiency. Although there have been many works in background of effects of different parameters on performance of cooling towers, there is no detail analysis about different alternatives of hybrid cooling systems due to the water consumption, to the knowledge of the authors. So, in the present work, the effects of the ambient temperature and relative humidity on dry, wet and two cases of combined cooling towers (which designed based on water consumption) performance were studied. Results showed that the plant power generation generally declines with the increase in the ambient temperature and relative humidity, which have matching with pervious works. All of the simulations were executed for a 250 MW plant capacity. Finally, the amount of water consumption of the dry, wet and hybrid towers is evaluated. The results maintained that the need for the make-up water increases as an outcome of augmentation of either ambient temperature or relative humidity. Finally, by an economic analysis, dry cooling system was determined to be the best choice, from both the economical aspect and the amount of water consumption of the case study, Hamedan Power plant in Iran.

Conflict of Interests

The authors have not declared any conflict of interests.

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Nomenclature

A:	Surface area
A_f:	The outer surface of the pipe
A_i:	Inner surface of the tube
A_t:	Average heat transfer area of the tube
A_{fr}:	Front surface of the heat exchanger
h:	Heat transfer coefficient
h_a:	External flow heat transfer coefficient (air side)
h_c:	Convective heat transfer coefficient
h_D:	Convective mass transfer coefficient
h_w:	Internal flow heat transfer coefficient (water side)
K:	Thermal conductivity of the pipe
\dot{m}_a:	Mass flow rate through a column of heat exchanger
$\dot{m}_{a,t}$:	Total mass flow rate of air
$\dot{m}_{w,t}$:	Total mass flow rate of water
Q_{ow}:	Flow rate of water through a column of heat exchanger
Q_c:	Released heat from condenser
R_f:	Sediment coefficient
T_{wi}:	Inlet water temperature
T_{wo}:	Outlet water temperature
U:	Total heat transfer coefficient
U_c:	Total heat transfer coefficient based on cold surface
W:	Absolute humidity

Greek symbols

δ:	Thickness of the tube
η:	Efficiency of the fin and tube
ρ_{oa}:	Air specific mass at standard conditions
ρ_{ma}:	Average air specific mass through a heat exchanger.