Research Paper

Developing of a novel water-efficient configuration for shower cooling tower integrated with the liquid desiccant cooling system

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1. Introduction

Cooling towers are utilized in several industries including power plants, chemical industries, petrochemicals, and air conditioning instruments. The task of cooling towers is to cool the water temperature used in procedure equipment and reuse it. Evaporative cooling towers have a great surface area for air-to-water contact, which makes the procedure of evaporation easy and causes rapid cooling of the water. The cooling operation is carried out through the loss of latent heat of evaporation, and the air stream transfers heat energy to the outside air through absorption of heat and moisture from the water. In wet or evaporative cooling towers, the air cools the water by direct contact. In humid areas, evaporative cooling towers are not usable because the relative humidity in these areas is high. In such a situation, the evaporative tower efficiency decreases sharply and is practically unused. In these areas, a dry cooling tower (Heller tower) or air-cooled condensers are used. In a dry cooling tower, using a fan, cool air is blown on a bundle of tubes where steam or hot water stream flows through them, thereby the temperature of the fluid inside the tube decreases. Electric energy consumption in this kind of system is much more compared to the evaporative cooling towers. A proposed solution in these areas is to reduce the humidity corresponding to the room air before entering the evaporative cooling tower using moisture absorbing materials to allow evaporative cooling towers to be used. The liquid desiccant system is a system with the ability to absorb and removal of moisture. Absorption and removal of air humidity are done by the difference in water vapor partial pressure on the desiccant material surface and air stream. The moisture is absorbed from the air by the desiccant, and this continues to saturation condition. Adsorbed moisture

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ABSTRACT

A novel configuration of shower cooling tower (SCT) with counter-current flow pattern and a liquid desiccant system equipped with the internal cooling system, used lithium chloride as an adsorbent, is developed and modeled separately and mathematically. The governing relations are solved numerically using a finite difference scheme, and the performance of the hybrid system is parametrically studied. The precision of numerical modeling is increased considering the air humidity impact on the air temperature changes during the moisture absorbing and changes in the diameter of the water droplets along the SCT. A novel configuration is developed by the combination of an SCT and a liquid desiccant system as a hybrid system, and the influences of geometric, physical and environmental parameters on the efficiency of the system are scrutinized. The results show that when the initial diameter of water drops is decreased, the temperature of outlet water reduces; whereas, using the hybrid system reduces the process water temperature up to about 32 °C and 37 °C with and without dehumidifier system at 10 m height of shower cooling tower. In hot and humid regions, reducing the process water temperature of shower cooling tower utilized in the hybrid system is greater compared to a single cooling tower.

HIGHLIGHT

• Increasing height of cooling tower has a greater effect on water temperature reduction.
• In humid subtropical climate, performance of the hybrid system is more than the single cooling tower.
• Variation of ambient temperature have a greater impact on the water reduction in a single cooling tower.
• Increase of water temperature at the inlet of cooling tower does not affect its performance in hybrid mode.

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1.1. Review of research on evaporative cooling towers

The use of cooling towers was proposed in 1923 by Walker et al. [1]. Two years later, this issue was pursued more seriously by Merkel [2]. He studied the direct stream towers and obtained the mathematical equations governing the cooling tower’s behavior by taking into account the relation between heat and mass transfer procedures. Although the Merkel model predicted the outlet water temperature, the absolute humidity changes corresponding to the air was considered to be negligible, and therefore the moisture level of the outlet air from the tower was not predictable. In the years after that, Merkel proposed the base of the desiccant systems, a review of the studies on the application of every part of the proposed systems has been carried out separately.

In the current research, the capability to use humidity absorber system in combination with an evaporative cooling tower in humid areas is investigated. In other words, the aim is the feasibility study to use evaporative cooling towers in humid regions with the air humidity absorption system. As there has been no report on the hybrid application of the two mentioned systems, a review of the studies on the application of every part of the proposed systems has been carried out separately.

1.1.1. Review of research on evaporative cooling towers

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In 1995, Givoni and Al Hemiddi [10] introduced a shower cooling tower (SCT) and used this system to cool the air in hot areas. The system contained water spray nozzles at the top part of the tower and the basin at the bottom. They examined the efficiency of this system in Saudi Arabia. The findings of this research presented that the inlet air of 45°C after passing the cooling tower reaches 29°C. After one year, Satoshi and others [11] studied the efficiency of an SCT and the effect of various parameters on its efficiency in Japan. The results of this research indicated that the outlet air temperature depends on parameters such as tower height, mass flow rates of inlet air and water, environmental conditions and extent of water droplet diffusion. Givoni [12] in another study examined the efficiency of the SCTs in different weather zones (Los Angeles, Riyadh, and Yokohama). He examined the effect of the height of the tower on the outlet air temperature. The results show...
that increasing the tower's height from 0.5 m to 1.0 m significantly reduces the air temperature, but increasing the height from 2.0 m to 3.0 m has no considerable much effect on decreasing the outlet air temperature. Mohiuddin and Kant [13] studied the design of wet cooling towers following different environmental and operational conditions. Then, in another study, they evaluated the transfer coefficients in cooling towers [14]. Gosh assay et al. [15] obtained the mass transfer rate and pressure drop in a filled bed cooling tower, including plastic fillers made of PVC. They showed that the mass transfer coefficient and the pressure drop were higher in terms of the application of unlined and leveling fillers compared with flat fillers. Alrahman-khan and Zubair [16] evaluated the drop in a filled bed cooling tower, including plastic fillers made of PVC. They solved the governing relations. They launched a numerical simulation of evaporative cooling process, finite difference scheme was applied for one-dimensional model using the relations governing procedures of heat and mass transfer. To predict the velocity of water droplets and the evaporative cooling process, finite difference scheme was applied for solving the governing relations. They [18] launched a numerical simulation of the SCTs using a neural network approach and provided an intelligent program for designing these towers. Engineers using this program could get the water outlet temperature from the tower by entering a set of laboratory information into the software. The average error of the proposed software was obtained 1.31% compared to the empirical results while using the governing equations-based model for heat transfer and mass transfer procedures, the absolute value of mean error was 9.42%. Xiaoni and Zhenyan [19] in another study in the same year, in order to progress the model and decrease its error, ignored the assumptions of saturated air in contact with water, the Luss coefficient to be one and constant considering the mass stream rate of water (regardless of the evaporation of water). Then, they compared the results of the new model with the empirical results and stated that the new model predicts the experimental results with high precision. Moreover, Xiaoni et al. [20] suggested a new method called projection pursuit regression (PPR) for quick, easy, and cost-effective simulation of SCTs. The findings indicated that the PPR model has higher accuracy than the heat transfer and mass transfer model. Wang and Li [21] investigated the thermal efficiency of a wet cooling tower with countercurrent flow patterns and exergy changes in it. The results indicated that the exergy efficiency is always below 25%. Also, the value of exergy transfer in the tower is affected by the temperature of the environment. Ricardo and Variyar [22] presented a new method for calculating the flow pattern in countercurrent flow patterns cooling towers by revising Merkel's equations. The findings of this method indicated that the proper height in the filling section is not affected by the rate of the water stream and the tower diameter, and only depends on the air stream mass flow rate. Shauny et al. [23] presented a new mathematical model for predicting the efficiency of SCT. This model was applied for predicting temperature and exergy changes across the tower. The results of this research indicated that with increasing tower height, water exergy decreases. Also, the distribution of exergy loss at the bottom of the tower is high and decreases as it moves towards the top of the tower. Also, by decreasing the water droplets diameter, the time and surface of mass transfer will increase. Jiang et al. [24] studied numerically and experimentally wet cooling towers with cross-flow patterns. In this study, a wet cooling tower was comprised of a plate-fin heat exchanger, and its performance was studied in different conditions. Heat transfer and mass transfer coefficients were obtained by numerical solution of the two-dimensional mathematical model and also empirical relations. The error between numerical resolution results and laboratory results was less than 8%. Asvapoositkul and Kuansathan [25] carried out a comparison between the efficiency of the hybrid cooling towers (wet/dry). In this research, a numerical technique was utilized for predicting system behavior. They presented that the results of solving the obtained equations are completely by empirical data. Muangnoi et al. [26] experimentally and numerically studied the performance of water-jet cooling tower. Also, the influence of parameters of the height of the spray area, water stream to air ratio, diameter, and velocity of water droplets and air velocity on system performance were investigated. Xiaoni et al. [27] studied numerically the efficiency of a SCT which uses seawater stream. They compared the temperature corresponding to the outlet water of the cooling tower in two modes of using freshwater and seawater. The results indicated that the performance of the cooling tower in reducing the temperature corresponding to the outlet water in the use of fresh water is better comparing with the use of seawater in similar conditions. In addition, by enhancing the water droplets diameter and the level of salt in the sea water, the efficiency of the cooling tower decreases. Zunaid et al. [28] presented a 2D numerical mathematical model for a SCT. In the proposed model, the mass, energy, exergy, motion and water droplet equations were solved to calculate the temperature corresponding to the exhaust air, and the validation of results then performed using empirical data. It was found that the temperature corresponding to the inlet droplets shows a considerable impact on the temperatures of outlet air and water, the tower's performance and exergy of the system. Moreover, they indicated that the maximum performance of the SCT at an inlet water temperature of 36 °C was 55.28%.

1.2. Review of research on liquid desiccant system

The application of desiccant systems has recently enhanced for absorbing humidity from the air flow. Airflow humidity is absorbed with desiccant systems before entering a cooling system. Thus, the consumption of energy in cooling systems is reduced and appropriate, and conditioned air is provided.

Humidity-absorbing systems are considered as low-cost systems due to the low energy consumption in the absorbent regeneration procedure, and the required energy can be provided with the aid of a variety of energy sources [29]. The humidity is absorbed by desiccant from the air flow, and this procedure is along with latent heat release. Liquid desiccants often are sprayed in the environment or on the surface to absorb water vapor in the environment [30]. If the partial pressure of the vapor in the desiccant is lower compared to that in air flow, the desiccant adsorbs humidity of the air, and this procedure continues until reaching a balance and saturation mode is achieved for desiccant. Material reaches cannot absorb humidity after reaching saturation mode and must be regenerated for reuse. The desiccant is exposed to the airflow with greater temperature, in the regeneration procedure. The vapor partial pressure in a warmer air is lower comparing its amount on the surface of desiccant material. Under these conditions, due to the partial vapor pressure difference at the interface between desiccant and air, humidity from absorbent material is transferred to warm air; therefore, the desiccant material is regenerated [31]. Desiccant systems are divided into two kinds of liquid and solid. Liquid desiccants are more utilized because of their effective flexibility and the capability to absorb pollution and bacteria in the air. A slight amount of desiccant solution as droplets is usually transferred via air flow in the liquid desiccant systems. It should be mentioned that liquid desiccants are more used in industrial applications that require a high volume of humidity-absorbing material.

Several studies have been conducted on the application of liquid desiccant systems. For example, Yadav introduced laboratory design corresponding to the combined solar air conditioning system comprising vapor compression system and liquid desiccant cycle [32]. In this system, the output heat from the condenser was utilized for regenerating the desiccant to increase system efficiency factor. He
showed that solar thermal energy could be utilized because of the low temperature required for regenerating the desiccant. Kinsara et al. [31] studied the impact of inlet desiccant solution temperature and performance of the heat exchanger on humidity-absorbing system efficiency was evaluated. The results of this study showed that, with increasing temperature of inlet desiccant solution, system efficiency is improved. Furthermore, system performance is intensely affected by the performance of the heat exchanger, and the humidity absorption can be considerably enhanced with an excellent design. Khan et al. [33] investigated a hybrid ECS integrated with a liquid desiccant system by developing a mathematical model. The findings indicated that, if a lithium chloride solution is utilized as a liquid desiccant, thermal efficiency corresponding to the liquid desiccant system is affected by the level of desiccant solution and mass flow rate of process air. A similar study is also conducted by Saman and Alizadeh [34]. Ali and Vafa [35] theoretically studied mass and heat transfer processes between airflow and liquid desiccant film in two modes of inclined parallel flow and inclined counterflow. They studied the impact of plates slope in the desiccant system on the humidity of air absorbed and regenerated. The results indicated that, when the Reynolds number of the air flow decreases, the level of humidity absorbed with the liquid desiccant enhances. Moreover, it was stated that, if the temperature of inlet air enhances and the liquid desiccant concentration reduces, the dehumidifying process is improved. Chen et al. [36] employed a mathematical model for the packed bed type desiccant system. They assumed that the concentration of outlet desiccant is constant, and solved the equations governing the system efficiency using an analytical solution. Also, the findings of the analytical solution were validated with the obtained results of previous studies. The results showed that using this method, it can compute the optimal flow rate of the flows. Yin et al. [37] used experimental model for packed-bed, and a lithium chloride solution-water was utilized as a liquid desiccant. The air-conditioning system had installation capability with heat sources which have the temperature between 60 and 80°C including industrial waste heat and solar energy. Laboratory findings indicated that Mass transfer coefficient in the regeneration procedure is 4.0 g/m² s. Liu et al. [38] theoretically studied heat and mass transfer procedures in dehumidifier and regeneration parts corresponding to the liquid desiccant system using a two-dimensional mathematical model. In this research, it was assumed that the flow pattern in the heat exchanger corresponding to the system is in the form of Cross-flow. Furthermore, the Lewis dimensionless numbers and NTU were utilized as input parameters, and the values of these parameters were obtained from experimental results. In this research, the variations in the amount of desiccant solution were ignored over the regenerating and dehumidifying processes. Alizadeh [39] experimentally studied the air-conditioning system integrated with the liquid desiccant system in Australia weather conditions in which solar energy was used in order to regenerate the desiccant solution. Lithium chloride solution as absorbent, a heat exchanger with polymer plates, warm water flow for regenerating and solar collectors as sources of heat, was used. The results indicated that in both conditions of warm and humid weather, the optimal flow rates for the liquid desiccant flow and air flow, respectively are equal to 3.0 L/min and 1000.0 L/s. Yin et al. [40] studied the liquid desiccant system equipped with internally-cooled and externally-cooled systems. Kumar et al. [41] mathematically studied air conditioning systems equipped with a liquid desiccant system. In this research, for solving nonlinear differential equations, the fourth order Runge-Kutta method was applied. To enhance the efficiency of the liquid desiccant system, two dehumidifier towers were used. The results indicated that using the proposed model compared to previous models, the efficiency of a liquid desiccant system improves up to 67%. Liu et al. [42] studied the mass transfer procedure in the liquid desiccant system with different solutions of lithium chloride (LiCl) and lithium bromide (LiBr). The results indicated that, under the same operating conditions, the amount of dehumidification of air flow is larger when the lithium chloride solution (LiCl) is used as adsorbent comparing to the application of the lithium bromide solution (LiBr). Woods and Kozubal [43] introduced an air conditioning system equipped with a liquid desiccant system and indirect evaporative coolers. They investigated the efficiency of the suggested system numerically and experimentally. In the first stage, a liquid desiccant solution absorbed humidity from the air, and in the second stage, the cooled air provided in an evaporative with countercurrent flow pattern was used. Gao et al. [44] studied liquid desiccant system combined with an indirect evaporative cooling system that benefits from Maisotesenko Cycle (M-Cycle) experimentally. They showed that when the flow rate and the humidity of air increase at the inlet of the system. Furthermore, by increasing flow rate and liquid desiccant concentration in the inlet of the system, the efficiency of both parts (moisture-absorbing system and evaporative air cooler) improves. Yin et al. [45] studied compressed air drying using a liquid desiccant system in which the dehumidifier and regenerator worked under high operating and atmospheric pressures.

**Fig. 1.** Schematic diagram of the hybrid system, including a shower cooling tower with countercurrent flow pattern and a liquid desiccant system equipped with the internal cooling section.
Under operating pressure of 0.3 MPa, energy consumption for absorption per gram of water in the dehumidifier section is 6.17 kJ/g, and this is 10.1% much smaller than air cooling dehumidification systems.

1.3. Objectives and innovations of the present research

Since the wet cooling towers do not have a good performance in warm-humid and moderate-humid climates, the towers are not used in these areas, and dry cooling towers are used instead of it. In dry towers, because of indirect contact, the heat transfer is only made due to the temperature difference. Also, electrical energy consumption is high in these cooling towers. In this study, the use of a hybrid system consisting of a countercurrent flow pattern and a liquid desiccant absorbing system equipped with the internal cooling part is feasible. Therefore, the efficiency of the combined system and the effect of various parameters (geometric, physical and environmental) on its performance are investigated. In summary, the innovations of this study are:

1. A hybrid system is modeled with the counter current flow pattern, considering the alteration in the water droplets diameter during the evaporation process along the tower.
2. The use of a shower cooling tower integrated with a liquid desiccant absorbing system with the internal cooling part as a new configuration is introduced, and its efficiency is studied in various operating conditions.

2. Introducing the proposed hybrid system

A hybrid system including a shower cooling tower with a counter-current flow pattern and a liquid desiccant absorbing system equipped with an internal cooling section are shown in Fig. 1. The warm and humid air stream is directly contacted with the stream of falling film of the desiccant solution, and since the vapor partial pressure on the amount of desiccant is less than the partial pressure corresponding to water vapor of the air stream, moisture is transferred from the air towards the solution. In other words, the humidity of air is absorbed with the desiccant solution, and the ratio of air humidity decreases. During this time, the solution temperature increases because of transferring the heat and mass from the air stream to desiccant solution. By increasing the desiccant solution temperature, the vapor partial pressure increases at the desiccant surface, which decreases the driving force of the mass transfer between the air stream and the desiccant solution. To reduce rising desiccant stream temperature, the solution must be cooled down during the dehumidification process. In this regard, cold water is applied as a cooling fluid in the heat exchanger corresponding to the desiccant liquid system to keep the temperature corresponding to the desiccant solution low during the dehumidification process. The diluted desiccant solution then comes back to the regeneration part for re-use in a dehumidifier and concentrates on reaching the required concentration with losing water. In the reduction section, by enhancing the desiccant solution temperature, the vapor partial pressure in the solution surface becomes larger comparing to the vapor partial pressure in the air stream. So the water is evaporated from the desiccant solution and enters the air stream. In this study, water was utilized as a coolant, and a solution of lithium chloride was utilized as a desiccant solution. The outlet air from this system enters the SCT after the process air humidity ratio decreases by the liquid desiccant system (Fig. 1). The Inlet dry air stream to the SCT is in direct contact with the water droplets stream which is sprayed by the nozzle from the top of the tower. By spraying water as small droplets, the water contact surface with air stream increases. The heat of water droplets is transferred to the air stream through a sensible and latent heat transfer. Finally, water droplets are collected at cooling tower outputs with lower-temperature for reuse in other process equipment.

To better understand the system of liquid desiccant, Fig. 2 shows the pattern of streams in a unit of the dehumidifying part of the system of liquid desiccant equipped with the internal cooling section. As observed in Fig. 2, a plate heat exchanger is utilized for cooling down the dehumidifying section. In this heat exchanger, the air stream and desiccant solution from the top of the heat exchanger enters the number channel (II). Desiccant solution (lithium chloride-water) is sprayed into the walls of this channel by nozzles and flows to the bottom part of the plate in the form of thin film due to the gravitational force. Desiccant solution and air stream inside the channel number (II) are directly in contact and exchange mass and heat. When dehumidification and regeneration process is performed, the cooling liquid and the heating fluid enter the channel number (I) respectively. The cooling or heating fluid enters from the top part of the channel and goes out from the bottom part of the channel after cooling or heating operations. The cooling or heating fluids and the desiccant stream are separated by a wall, where the heat transfer between the channel number (I) and (II) is performed across it.

3. Mathematical modeling

In this section, the and the liquid desiccant system are presented separately, and the needed equations are extracted to study their behavior and performance.

3.1. Mathematical modeling of SCT with the countercurrent flow pattern

In the evaporative cooling tower, hot water is sprayed via some nozzles as droplets from the top part of the tower. The water droplets move downwards because of the gravitational force, and they are contacted with the air stream that comes from the bottom part of the tower (water droplets are directly in contact with the air stream). The thermal energy corresponding to the droplets is transferred to the air, and their temperature decreases. Finally, water droplets will be cooled and collected at the bottom part of the tower. The subsequent assumptions have been utilized here to model the SCT with counter-current flow pattern:

1. Water droplets move vertically from nozzles to the reservoir.
2. The droplets are spherical, and their shapes do not change the process; however, their size variations during movement will be considered over the motion along the tower.
3. Radiation heat transfer between the air stream and the water droplets is neglected since the difference in their temperatures is low.

Based on Fig. 3, the direction of the droplets’ velocity and the gravity is downward, and the direction of the Drag and buoyancy forces corresponding to the air is upward. Therefore, the droplets movement is
divided into two stages:

1. The droplet movement from the nozzle until it reaches a constant speed.
2. When the droplet reaches a constant speed and continues to move at a constant speed.

Based on Fig. 3, the forces applied are gravity, buoyancy, and drag, respectively:

\[ G = m_d g = \frac{1}{6} \pi p_m d_d^3 \]  
\[ B = \frac{1}{6} \pi p_m d_d^3 \]  
\[ D_T = \frac{1}{8} C_d \pi p_m u_d^2 d_d^2 \]  
\[ (1) \]  
\[ (2) \]  
\[ (3) \]

The drag coefficient of the water droplet is computed using the following equation [19]:

\[ C_d = \begin{cases} 18.5 \frac{Re}{Re_{cr}} & (1.917 \leq Re \leq 508.4) \\ 0.44 \frac{Re}{Re_{cr}} & Re \geq 508.4 \end{cases} \]  
\[ (4) \]

The Reynolds number is computed using the equation of

\[ Re = u_d d_d / \nu \]

According to Fig. 3, the equilibrium of forces for a droplet is taken into account as follows:

\[ m_d u_d \frac{dz}{dt} = (B + D_T - G) \]  
\[ (5-a) \]

By substituting Eqs. (1)–(3) in Eq. (5-a), the final equation is obtained for calculating the motion of a water droplet as follows:

\[ \rho_w u_d \frac{dz}{dt} = (\rho_w - \rho_d) g - 0.75 C_d p_d \frac{(u_d + u_a)^2}{d_d} \]  
\[ (5-b) \]

Since the evaporation is done from the water droplets surface that is in contact with the air, the droplets diameter is changed. The required equation for estimating the change in the droplet diameter will be obtained further. When the water vapor partial pressure at the interface of droplet and air is more than its value in the air stream, mass transfer occurs from the surface of the droplet to the air stream. The mass transfer process is along with a latent heat transfer and water evaporation. Simultaneously, the sensible heat transfer becomes active because of the difference between the temperature of the water droplets and the air stream. To calculate the change in air temperature and water droplets during the process, the subsequent assumptions are considered:

1. The tower is taken into account as an adiabatic system, and the heat exchange between the air stream, water droplets, and the surrounding air is ignored.
2. Radiation heat transfer is ignored due to the small difference in temperatures.
3. The temperature distribution inside the water droplet is ignored because of its small size.
4. The temperature at the interface of the air stream and droplets of water is assumed to be the water temperature.

Fig. 4 shows the control surface around a droplet. The energy balance equation on the control surface is written as follows:

\[ \frac{dQ_t}{dt} = -(Q_s + Q_e) \]  
\[ (6) \]

where \( Q_s \) represents a sensible heat transfer between the airflow and water droplets, and \( Q_e \) represents the latent heat transfer between saturated vapor surrounding water droplets and air stream. \( Q_s, Q_e, \) and \( Q_d \) are calculated by the subsequent equations:

\[ Q_s = m_d C_p T_w \]  
\[ (7) \]
\[ Q_e = h_{ev} A_d (T_w - T_i) \]  
\[ (8) \]
\[ Q_d = h_{md} A_d (\omega_{sv} - \omega_d) l_v \]  
\[ (9) \]

where,

\[ h_{md} = \frac{N u_k k_d}{d_d} \]  
\[ (10) \]

Moreover, \( N u_k \) is computed as following [19]:

\[ N u_k = 2 + 0.6 R e^{0.5} P r^{0.33} \]  
\[ (11) \]

By substituting the Eqs. (7)–(9) in Eq. (6), Eq. (12) is achieved as follows:

\[ \frac{dT_w}{dt} = \frac{[h_s (T_w - T_i) + h_{md} (\omega_{sv} - \omega_d) l_v] A_d}{C_{pwa} m_d} \]  
\[ (12) \]

\[ (12) \]

where \( \omega_d \) is the mass transfer coefficient and is achieved as following [19]:

\[ \omega_d = \frac{h_s}{C_f \alpha} \]  
\[ (13) \]

\( I_{maw}, I_{ma}, \) and \( I_v \) are enthalpies of saturated water vapor, humid air, and water vapor, respectively, and are calculated as follows [46]:

\[ I_{maw} = C_{pwa} T_w + \omega_{aw} (I_{f0vo} + C_{pwa} T_w) \]  
\[ (14) \]
\[ i_{ma} = C_{pwa} T_w + \omega_d (I_{f0vo} + C_{pwa} T_w) \]  
\[ (15) \]
\[ i_v = I_{f0vo} + C_{pwa} T_w \]  
\[ (16) \]

By subtracting Eq. (15) from Eq. (14) and substituting the result in Eq. (16), the following equation is obtained:

\[ T_w - T_i = \frac{[I_{mah} - i_{ma} - l_v (\omega_{sv} - \omega_d)]}{C_{pma}} \]  
\[ (17) \]

where \( C_{pma} \) is a specific heat for the wet air and is obtained as follows [46]:

\[ C_{pma} = C_{pwa} + \omega_d C_{pv} \]  
\[ (18) \]

Considering the water droplets as a sphere \( A_d = \pi d_d^2 \) and \( m_d = \rho_d \frac{\pi d_d^3}{6} \) substituting Eqs. (13) and (17) in Eq. (12), the first-order differential equation for calculating water droplet temperature variations over the evaporation process is obtained as follows:

\[ \frac{dT_w}{dz} = \frac{-6 h_{ma} [\alpha (I_{maw} - i_{ma}) + (1 - \alpha) (\omega_{sv} - \omega_d) l_v]}{C_{pwa} \rho_d m_d} \]  
\[ (19) \]

Assuming that Lewis number is 1.0 and substituting \( dt = \frac{dz}{u_d} \) in Eq. (19), the governing differential equation to estimate the changes in the temperature of the water droplet is obtained as follows:

\[ \frac{dT_w}{dz} = \frac{-6 h_{ma}}{C_{pwa} \rho_d u_d d_d} (I_{maw} - i_{ma}) \]  
\[ (20) \]
The mass transfer rate from the water droplets surface is achieved as follows:

\[
\frac{dm_d}{dt} = h_{\text{m}} A_d (\omega_{\text{in}} - \omega_a)
\]  \hspace{1cm} (21)

\[
dm_w = n_d dm_d
\]  \hspace{1cm} (22)

\[
dm_m = N_j dm_d
\]  \hspace{1cm} (23)

where \(N_j\) is the number of water droplets passing through the tower cross section at any given time:

\[
N_j = \frac{m_c}{m_d} = \frac{6m_w}{\rho_v \pi d_d^2}
\]  \hspace{1cm} (24)

Due to the evaporation of water droplets, their mass and size decrease along the tower. To achieve an equation to calculate the altered in the water droplets diameter, the mass change of each droplet is calculated by the combination of the equations \(dt = \frac{dz}{u_d}\) and (21):

\[
\frac{dm_d}{dz} = h_{\text{m}} A_d (\omega_{\text{in}} - \omega_a)
\]  \hspace{1cm} (25)

By differentiating from the equation \(m_d = \rho_v \pi d_d^2/6\) versus \(dz\), the mass change of each droplet is calculated as:

\[
\frac{dm_d}{dz} = \frac{\rho_v \pi d_d^2}{2} \frac{d(\omega_a)}{dz}
\]  \hspace{1cm} (26)

By combining Eqs. (25) and (26), the following equation is obtained:

\[
\frac{d(\omega_a)}{dz} = \frac{2h_{\text{m}} (\omega_{\text{in}} - \omega_a)}{\rho_v u_d}
\]  \hspace{1cm} (27)

By combining Eqs. (21)–(23) and using Eq. (24), the following equation is obtained:

\[
\frac{dm_w}{m_a} = N_j \frac{dm_d}{m_\text{a}} = \left( \frac{m_c}{m_\text{d}} \right) \frac{h_{\text{m}} A_d (\omega_{\text{in}} - \omega_a)}{m_\text{a} u_d} dz
\]  \hspace{1cm} (28-a)

By rewriting the Eq. (28-a), the differential equation for calculating the humidity ratio corresponding to water droplet along the cooling tower is obtained as follows:

\[
\frac{d\omega_a}{dz} = \left( \frac{m_a}{m_\text{d}} \right) \frac{6h_{\text{m}} (\omega_{\text{in}} - \omega_a)}{\rho_v u_d d_d}
\]  \hspace{1cm} (28-b)

To calculate air temperature changes, the energy balance equation between water droplets and air flow in a differential element with the height of \(dz\), which is presented in Fig. 5, is written as follows:

\[
dQ = m_a dm_{\text{in}} = m_w dm_w + i_w dm_w
\]  \hspace{1cm} (29)

According to the Figs. 4 and 5, the energy balance between water droplets and air flow can be presented as follows:

\[
dQ = dQ_a + dQ_i = h_{\text{i}} (T_a - T_i) dA + i_w h_{\text{m}} (\omega_{\text{in}} - \omega_a) dA
\]  \hspace{1cm} (30)

By substituting Eq. (17) in Eq. (30):

\[
dQ = h_{\text{i}} [Le(i_{\text{in}} - i_m) + (1 - Le)(\omega_{\text{in}} - \omega_a)i_e] dA
\]  \hspace{1cm} (31)

“\(A\)” in Eq. (31) represents the heat transfer surface between the air stream and water droplets located in a cooling tower element with a height of \(dz\), and the differential of which is as follows:

\[
dA = A_d N_j dt
\]  \hspace{1cm} (32)

Given the Lewis number equals one and with the combination of Eqs. (29) and (31) and the use of Eq. (32) in them, the governing differential equation for the enthalpy change of humid air along the tower is obtained as follows:

\[
\frac{d\text{in}}{dz} = \frac{dQ}{m_\text{a}} = \left( \frac{m_w}{m_\text{a}} \right) \frac{6h_{\text{m}} (i_{\text{in}} - i_m)}{\rho_v u_d d_d}
\]  \hspace{1cm} (33)

3.2. Mathematical modeling of the liquid desiccant system with internally cooling part

According to Fig. 2, the air flow in the channel (II) and the solution of liquid desiccant falling in the form of a film are directly in contact with each other. Thus, heat and mass transfers are done between them. To obtain the governing equations for heat and mass transfer processes in this system, the subsequent assumptions have been made:

1. Temperature and concentration gradient across are neglected through the flows’ depth [48].
2. Lewis number is considered to be a constant value [49].
3. Variations in the thermophysical properties with temperature are intended in the model [49].
4. Radiation heat transfer is neglected because of small temperature differences [49].
5. The flow in the channels is fully developed hydro-dynamically and thermally [49].
6. Operating condition of the system is considered as steady-state [49].

Governing equation for predicting the variations in temperature of the water is as follows [48]:

\[
\frac{dT_a}{dz} = \frac{2h_{\text{i}} W_{\beta}}{m_\text{a} C_{\text{pa}} L} (T_a - T_i) + \frac{2h_{\text{i}} W_{\beta}}{m_\text{a} C_{\text{pa}} L} (T_a - T_i)
\]  \hspace{1cm} (34)

The changes in air humidity ratio along the desiccant tower are expressed by [48]:

\[
\frac{d\omega_a}{dz} = -\frac{2h_{\text{i}} W_{\beta}}{m_\text{a}} (\omega_a - \omega_{\text{in}})
\]  \hspace{1cm} (35)

Air temperature variation along the air stream is presented with the subsequent equation [48]:

\[
\frac{dT_a}{dz} = \left( \frac{1}{C_{\text{pa}} + C_{\text{pu}} \omega_a} \right) \times \left[ \frac{2h_{\text{i}} W_{\beta}}{m_\text{a}} (T_i - T_a) + \frac{2h_{\text{i}} W_{\beta}}{L m_\text{a}} (T_a - T_i) \right]
\]  \hspace{1cm} (36)

The governing differential equation for computing the desiccant
solution concentration is as follows \[48\):
\[
dX_i = \left(\frac{X_i^0 - X_i}{m_i}\right) dm_i
\]  
(37)

Temperature variations of the desiccant solution are gained by the following equation \[48\]:
\[
\frac{dT_s}{dz} = \frac{h_w L}{m_v c_p}(T_s - T_v) + \frac{h_w L}{m_v c_p}(T_v - T_a) + \ldots
\]  
(38)

Mass transfer coefficient \(h_D\) in the dehumidifier sector is calculated by the following formula \[47\]:
\[
Sh = 4.513 \times 10^{-5} Re^{0.5} Sc^{0.33}
\]  
(39-a)

\[
f = 76.456 T_s^{2.991}
\]  
(39-b)

After calculating the Sherwood number, and using Eq. (40), the mass transfer coefficient is obtained \[47\]:
\[
h_D = \frac{D Sh}{d_i}
\]  
(40)

Heat transfer coefficient between airflow and liquid desiccant solution, after determining the mass transfer coefficient is calculated using the subsequent equation \[47\]:
\[
Le = \frac{h_a}{h_D C_p w}
\]  
(41)

4. The numerical solution of the governing differential equations and validation of the results of the mathematical models

In this section, the numerical solution method of the governing differential equations for the shower cooling tower and the liquid desiccant system and the flowcharts for solving their equations are expressed separately. Also, to ensure the accuracy of the results corresponding to the numerical solution for differential equations achieved from the modeling for the SCT and the liquid desiccant system, the comparison was performed between these results and the experimental results reported in previous studies.

4.1. Numerical solution of the differential equations obtained for the SCT

Differential equations (5-b), (20), (27), (28-b) and (33) fully describe the behavior and performance of the SCT with countercurrent flow pattern. Proper boundary conditions are needed to solve the governing equations. The water droplets and the air flow velocity and the initial diameter of the droplets must be specified at the entrance of the tower. The humidity ratio for inlet air, water and air flow temperatures at the entrance of the tower is equal to its value in ambient air. Fig. 6 illustrates the flowchart of solving the governing equations. In this research, the finite difference method has been applied for solving the equations. In this method, a repetitive numerical evaluation was applied for reducing the relative error of the outlet air from the tower to lower than \(10^{-6}\). Uniform meshes with fine size were utilized to attain acceptable precision. A careful study was carried to obtain the results which are not depended to the mesh size for ensuring the accuracy and validity of the numerical results. The mesh size used were between 100 and 3500. It was observed that if the mesh size increased from 3400 to 3500, the air stream temperature changed at the exit of the SCT was in the order of \(10^{-3}\). Therefore, the mesh size of 3500 was utilized to achieve the results.

To assess the accuracy of the mathematical model presented for the SCT, calculations were performed in the similar conditions of Ref. [26]. The comparison conditions are listed in Table 1. The analysis corresponding to the reported errors in Table 1 displays that the results of the current research are closer to the experimental results than the numerical results of Ref. [26]. The highest error was about 1.2%. Therefore, the proposed mathematical model can predict the temperatures of the outlet water and air from the tower well. There is a slight difference between the numerical results of the present study and the results reported in Ref. [26], because of considering the alteration in the water droplets diameter during the evaporation process in this research and ignoring it in Ref. [26].

4.2. Numerical solution of the differential equations obtained for the liquid desiccant system

To assess the efficiency of the desiccant system under diverse conditions, Eqs. (34), (35), (36), (37) and (38) were solved numerically. Fig. 7 shows the flowchart of the governing equations solution process. Mass and heat transfer coefficients are obtained by Eqs. (39)–(41). Air humidity, air temperature, water temperature, the temperature of the desiccant solution and solution concentration in each node are calculated by Eqs. (34), (35), (36), (37) and (38), respectively. This process continues until all parameters at the outlet system are calculated.

Fig. 6. Flowchart for solving differential equations governing the shower cooling tower with countercurrent flow pattern.
The temperature of air at the desiccant system outlet is obtained by varying the number of computing nodes from 100 to 1000 to certify the results independence to computing nodes number, and the results were shown in Fig. 8. The results show that by using 1000 calculating nodes comparing with 800 calculating nodes, variations in the air temperature at the outlet will be in the order of $10^{-3}$. This is the reason for using calculating 1000 nodes for the computation of all results in a numerical solution. The results corresponding to the numerical solution were compared with experimental and theoretical results of Ref.[47], on the same conditions showed in Table 2. It should be mentioned that for the computation of this part, the wettability coefficient for the surface is 0.7 and the Lewis number is considered equal to 0.3–0.5.

The results presented in Table 2 display that there is no considerable difference between the results of this research and the experimental results. The average error in this research comparing with the experimental results is 0.67%, whereas the numerical results average error of the presented work in Ref.[47] is 1.75%. The results of the study are closer to the experimental ones; unlike the Ref.[47], air humidity changes are herein considered in the equation of air temperature changes.

5. Results and discussion

In this part, the efficiency of the combined system is evaluated and analyzed parametrically. To better understand the efficiency of this system, the impact of important factors on the efficiency of the combined system included geometric, physical and environmental parameters is investigated in three sections. For this purpose, the initial conditions considered for the application of the dehumidifier tower and the cooling tower are shown in Table 3. The data in this table has been utilized in the subsequent three sections. The output water stream temperature of the combined system is the key parameter which the efficiency of the hybrid system is studied based on its amount.

5.1. Investigating the effect of geometric factors on the performance of the hybrid system

The performance of the hybrid system depends on the geometric parameters, including the initial water droplets diameter, the height, and width of the dehumidifier tower, and the SCT height. In this part, the impact of the above factors on the temperature of the output water of the SCT is investigated.

5.1.1. Investigating the impact of the initial diameter of the water droplets on the performance of the hybrid system

Fig. 9 shows variations in the temperature of output water temperature at the outlet of the SCT in the combined system in the various diameters of the water droplets entering the cooling tower. By increasing the initial diameter of the inlet droplets into the cooling tower, the temperature of output water from the hybrid system increases. Whatever, initial water droplets diameter is larger, the overall contact surface of the water droplets with the air flow decreases. In addition, because of the rise of the droplets weight, their contact time with air flow reduces. Therefore, the overall surface and the required time for mass and heat transfer between the droplets of water and the air flow decreases, which results in increasing the water temperature at the outlet of the cooling tower. It should be noted that reducing the initial diameter of the water droplets will reduce their weight, and they are performed with the airflow outside the tower. Therefore, the water droplets diameter should not be less than a certain size.

5.1.2. Investigating the impact of cooling tower height on the performance of the hybrid system

One of the key geometric parameters affecting the output water temperature is the height of the cooling tower. Fig. 10 shows variations in the temperature corresponding to the outlet water in two modes, with and without application of a dehumidifier (liquid desiccant system) in a condition where a cooling tower with different heights is
used. By increasing the cooling tower height, the temperature of output water of the hybrid system and the single cooling tower decreases. As the cooling tower height increases, the contact time between the air flow and the water droplets increases, which increases the mass and the heat transfer rate. According to Fig. 10, the reduction of the temperature of output water of cooling tower is higher in the conditions of the hybrid system. Because the air entering the cooling tower from the system of liquid desiccant has lower moisture content than the environment air.

Fig. 11 shows the variations in the percentage of process water reduction applied in the cooling tower in two conditions with and without application of the dehumidifier system. The results showed that with increasing the cooling tower height, the amount of water loss increased. The mass ratio of the water loss to the inlet water into the cooling tower is calculated from Eq. (42):

\[
P_{wtr} = \frac{m_{w\text{-in}}} {m_{w\text{-feed}}} \times 100
\]  (42)

The amount of water loss when applying a hybrid system is greater than the situations in which a single cooling tower is used. Because when a hybrid system is applied, due to less humidity of the air stream enters the cooling tower, the driving force to do the mass transfer process in that tower is higher, and more humidity is absorbed by the air and moved to the outside.

Fig. 12, is presented for better understanding of the effect of changing the cooling tower height on the efficiency of the combined system. This Fig shows the percentage varies in the reduction of the temperature of output water at the outlet of a cooling tower in two states with and without application of the desiccant system. The percentage of reduction of the outlet water temperature can be computed with Eq. (43):

\[
P_{wtr}(\%) = \frac{T_{w\text{-in}} - T_{w\text{-out}}}{T_{w\text{-in}}} \times 100
\]  (43)

With a rise in the cooling tower height, the amount of the percentage of process water reduction increases, and the temperature of output water of the system decreases further. In other words, increasing the cooling tower height will develop the efficiency of the combined system. However, if only the cooling tower is used, the development in the

---

**Table 2**

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<th>(m_a) (m³/s)</th>
<th>(T_a) (°C)</th>
<th>(\omega_a) (g/kg)</th>
<th>(T_w) (°C)</th>
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<th>(Le)</th>
<th>(\omega_s) (g/kg)</th>
<th>Error (%)</th>
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<td>Num. (present study)</td>
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**Table 3**

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</table>

(B) Shower cooling tower

| \(H\) (m) | \(d_{in}\) (m) | \(m_{r\text{-in}}\) (kg/kg) | \(T_{w\text{-in}}\) (°C) |
|-----------------------------|
| 4.0 | 0.0013 | 0.88 | 60.0 |

---

**Fig. 8.** Air temperature changes at the outlet of dehumidifier tower with the changes of mesh size.

**Fig. 9.** Changes in the temperature of output water of cooling tower by altering the initial diameter of the water droplets entering it.

**Fig. 10.** Changes in the temperature of output water of cooling tower by changing its height in two modes with and without application of the dehumidifier system.
tower’s performance for the taller tower is not as good as the application of a combined hybrid system. According to Fig. 12, with a rise in the tower height, the impact of applying the hybrid system in decreasing the temperature of the outlet water increases, so that in the case of application of a cooling tower with a height of 10.0 m, the temperature of output water of the hybrid system compared to the temperature corresponding to the water outlet from the single cooling tower, has reduced 11.6% further.

5.1.3. Investigation of the influence of dehumidifier tower height on the performance of the hybrid system

Fig. 13 shows the variations in the temperature of output water of the cooling tower of the combined system by varying the dehumidifier tower height. With increasing the height of the dehumidifier tower, the temperature of output water reduces further. With an increase in the heat exchanger height corresponding to the dehumidifier, the required time for heat and mass transfer between the air flow and the desiccant solution surface increases. Therefore, the humidity ratio for the air at the dehumidifier outlet decreases more, and the air enters a cooling tower with a lower humidity ratio. This also raises the amount of mass transfer from water droplets to the air stream in the evaporative cooling tower, which increases the amount of latent heat transfer in this section of the system. By enhancing the amount of latent heat transfer, the water droplets temperature decreases further, and subsequently, the amount of heat transfer is also increased. In the case of not using the liquid desiccant system, ambient air enters the tower without the loss of humidity. Under the mentioned condition, the temperature of the water exiting from the cooling tower will be 39.9 °C (for the height of 0.0 from the dehumidifier tower) as presented in the Fig. 13. Though, if a hybrid system with the dehumidifier height of 3.0 m is utilized, the temperature of output water of cooling tower is 34.5 °C, and it is 5.4 °C lower.

5.1.4. Investigation of the effect of dehumidifier tower width on the performance of the hybrid system

Fig. 14 shows the changes in the temperature of output water of the cooling tower at various widths of the dehumidifier system. Assuming the mass flow rate for the air entering the dehumidifier system is constant, the temperature of output water of cooling tower reduces with enhancing the heat exchanger’s width. By increasing the width of the heat exchanger corresponding to the dehumidifier, the air stream contact surface enhances with the desiccant solution stream. Therefore, the heat and mass transfer between these two flows are increased, and so, the humidity ratio and the air stream temperature at the dehumidifier outlet are reduced. As a result, the air entering the cooling tower has lower temperature and humidity ratio, and the temperature of output water of cooling tower decreases more. However, if only the cooling tower is used, the temperature of output water is 39.9 °C.

5.2. Investigating the impact of physical parameters on the performance of the hybrid system

In this section, the impact of the physical parameter of the cooling tower which is the ratio of rates of mass flow of water to air at the inlet of the cooling tower on the temperature of output water of the cooling tower and the water consumption is investigated. Fig. 15 shows the variations of the temperature of output water of the cooling tower in different mass ratios in both conditions with and without the application of a dehumidifier system. According to Fig. 15, as this ratio increases, the temperature of output water of the cooling tower increases, and the efficiency of the combined system becomes weaker. This process is also found in a single cooling tower. As this ratio increases, mass and heat transfer between air and more volume of water is occurred, and accordingly the process water temperature increases.

Fig. 16 shows the impact of the ratio of the inlet water mass flow rate to the inlet air mass flow rate of the cooling tower on the percentage of process water reduction (It has been computed using Eq. (42)) in the combined system and the single cooling tower. By increasing the ratio of water flow to air flow rate, the percentage of process water reduction in the cooling tower decreases. As the ratio increases, water temperature along the cooling tower enhances. Accordingly, the saturated humidity ratio (ωsat) (it is computed at water temperature.) at the interface of water droplets and air stream decreases. Under this condition, the difference between air humidity ratio and saturated humidity ratio reduces; therefore, the fewer amount of water vaporizes into the environment. Generally, the amount of water loss in a single cooling tower is less comparing to the amount of water loss in the hybrid system.

Fig. 17 has been presented to better understand the impact of the ratio of water mass flow rate to air mass flow rate at the entrance of the cooling tower on the hybrid system performance and single cooling tower. This Fig shows the variations in the percentage of the water temperature reduction (Pwtr) at the cooling tower outlet by changing the mass flow rate ratio. By increasing the mass flow rate ratio, the Pwtr is reduced. Whatever the mass flow rate to the tower ratio decreases, the influence of using a hybrid system on water temperature reduction at the cooling tower outlet increases. As an example, in a mass ratio of 0.3, the temperature of output water compared to the temperature of output water of the single cooling tower has been further reduced by 7.6% more.

5.3. The effect of environmental factors on the performance of the hybrid system

The humidity and temperature ratio of the input air to the dehumidifier tower is one of the key parameters, which affect the efficiency of the hybrid system. Environmental factors are different depending on geographic and time conditions and cannot be managed and regulated
Fig. 13. Changes in the temperature of output water of the cooling tower by changing the dehumidifier tower height.

Fig. 14. Changes in the temperature of output water of the cooling tower by altering the dehumidifier tower width.

Fig. 15. Changes of the temperature of output water of the cooling tower by changing the ratio corresponding to the mass flow rate of water to air at the entrance of the cooling tower in two modes of with and without application of the dehumidifier system.

Fig. 16. The percentage of process water reduction in the cooling tower by changing the ratio of the water-to-air mass flow rate ratio at the entrance of the cooling tower in two modes with and without using the dehumidifier system.

Fig. 17. The percentage of the water temperature reduction at the outlet of the cooling tower by changing the water-to-air mass flow rate ratio at the entrance of the cooling tower for two modes of with and without using the dehumidifier system.

in any way. Consequently, it is vital to investigate the efficiency of the combined system with their change.

5.3.1. Investigating the impact of relative humidity of ambient air on the performance of the hybrid system

Fig. 18 shows the impact of relative humidity of ambient air on the temperature of output water of the cooling tower with using the combined system as well as the single cooling tower. The temperature of output water of the system increases by increasing the relative humidity ambient air. When the relative humidity of ambient air increases, because of the reduction of the driving force necessary for the mass transfer process on the dehumidifier, the air humidity absorption decreases with the desiccant solution, and accordingly the moisture level of the output air decreases less. Further, in the cooling tower, the driving force decreases to carry out the mass transfer process between water droplets and air stream. Therefore, less latent heat transfer occurs, and the temperature of output water of the cooling tower is reduced to a lesser extent. This process is also observed when a single cooling tower is used. The results of Fig. 18 show if the relative humidity of ambient air is high, the influence of the combined system comparing with the single cooling tower in reducing the temperature of output water is more. As an example, in relative humidity of 100%, the difference in output water temperature decrease of a cooling tower in two conditions with and without the use of a dehumidifier is 3.24°C, and in relative humidity of 10%, this difference is 0.83°C.

Fig. 19 shows the effect of the relative humidity corresponding to the ambient air on the percentage of water reduction in the cooling tower under conditions of use of the hybrid system and the single cooling tower. Increasing the relative humidity corresponding to ambient air reduces the water loss in the system, which is because of the decrease in the mass transfer rate in the cooling tower. Besides, Fig. 19
shows that, in the case of using a single cooling tower, the amount of water loss in the cooling tower is lower compared to the hybrid system.

Fig. 20 is showed to better understand the influence of the relative humidity ambient of air on the efficiency of the system. This Fig shows the Ppwr at the outlet of the cooling tower in the case of utilizing the hybrid system and single cooling tower by altering the relative humidity of ambient air. According to Fig. 20, in both the combined system and the single cooling tower, with the rise in the relative humidity ambient of air, the Ppwr decreases. By increasing the relative humidity of the air, the evaporation rate in the cooling tower reduces. Though, it should be mentioned that the greater the relative humidity of the ambient air, the more specific the superiority of the combined system comparing to the cooling tower will be. Surging in the relative humidity of ambient air, the capability of the hybrid system to reduce the process water temperature will be more. In high humidity region, a single cooling tower is not very efficient, since the rate of mass transfer and latent heat transfer in higher humidity are lower. However, the dehumidifier system reduces air humidity and increases the rate of evaporation of water droplets and latent heat transfer in the cooling tower.

5.3.2. Effect of ambient air temperature on the performance of the hybrid system

The ambient air temperature is another parameter that influences the hybrid system performance. Fig. 21 shows the variations of the temperature of output water of the cooling tower at different ambient temperatures in two modes with and without the use of a dehumidifier system. The influence of increasing the ambient temperature on the system efficiency is similar to the influence of increasing the relative humidity of environment air on it. As presented in Fig. 21, with an increase in the temperature of environment air from 25 to 45 °C, when the air relative humidity is 70%, temperature of output water of the single cooling tower is increased by only 6 °C. However, with the increase in the temperature of the air, the influence of the hybrid system on reducing the temperature of output water of the cooling tower is more than the single cooling tower. For example, when the environment air temperature is 45 °C, the difference between the temperature of output water of the cooling tower in two conditions with and without using the humidity system is 3.4 °C, and this difference is 0.25 °C when the temperature of environment air is 25 °C.

Fig. 22 shows the amount of water loss in a SCT for increasing ambient temperature. With increasing ambient temperature, the rate of water lost in the system decreases. As the environment air temperature increases, the temperature of the output air stream of the dehumidifier increases. In addition, the vapor partial pressure difference between the air stream and droplets of water in the cooling tower decreases, which reduces the rate of mass transfer.

Consequently, less water evaporates from the water droplets surface. This procedure is also observed in a single cooling tower. But as air humidity in a single cooling tower is more than air humidity in a hybrid system, the driving force for the mass transfer is less and less evaporation occurs. Consequently, the rate of water loss in a single cooling tower is less.

Fig. 23 is presented for better understanding of the influence of ambient air temperature on the combined system performance. This Fig shows the variations in the Pwtr at the outlet of the cooling tower for two modes of the hybrid system and single cooling tower at different ambient temperatures. As the environment air temperature increases, the Pwtr at the outlet of the single cooling tower and the hybrid system decreases. According to Fig. 23, with enhancing the ambient air temperature, the influence of the hybrid system on reducing water temperature is higher than the single cooling tower. So that, at 25 °C, the difference in outlet water temperature in two modes of applying the hybrid system and single cooling tower is 1.0 °C, while, this difference at 45 °C is 5.0 °C.

In general, based on Figs. 20 and 23, it can be stated that in hot and humid regions, a hybrid system containing a cooling tower and a liquid desiccant system comparing with a single cooling tower, has more performance.

5.3.3. Investigation of the impact of the inlet desiccant solution temperature to the dehumidifier tower on the performance of the hybrid system

Fig. 24 indicates the influence of the inlet desiccant solution temperature to the dehumidifier on the temperature of output water of the cooling tower of the hybrid system. The outcomes indicate that by increasing desiccant solution temperature, the temperature of output water increases. However, as can be observed, the effect is not significant, and when the solution temperature is enhanced by 10 °C, the temperature of output water of the cooling tower has increased by only 1.0 °C. Increasing the desiccant solution temperature increases the vapor partial pressure at the surface the desiccant film and reduces the driving force for the mass transfer, which leads to a reduction in the absorption of humidity from the air stream. As a result, the air enters the cooling tower with a higher humidity ratio, and the temperature of output water of the cooling tower slightly increases.

5.3.4. Investigating the impact of water temperature at the inlet of the dehumidifier tower on the performance of the hybrid system

Fig. 25 indicates the influence of increasing the cold water temperature at the inlet of the dehumidifier on the temperature of output water of the cooling tower. The results indicate that with increasing this temperature, the temperature of output water of cooling tower increases. The reason is the rise in temperature and humidity for the air stream at the dehumidifier outlet. Under these conditions, the amount of heat and mass transfer between the droplets of water and the air stream along the cooling tower decreases, which causes a lower reduction of the process water temperature at the cooling tower outlet.

5.3.5. Investigation of the impact of water temperature at the inlet of the cooling tower on the performance of the hybrid system

Fig. 26 shows the influence of the temperature of input water of the cooling tower on the temperature of output water of the cooling tower for two modes of applying the hybrid system and the single cooling tower. Based on Fig. 26, at the constant relative humidity, with increasing the input water temperature of the cooling tower, the temperature of output water enhances. Though, based on Fig. 26, the rise in the temperature of output water of the cooling tower is not high when the temperature of inlet water increases. For an increase of 20 °C in the input water temperature of the cooling tower, the temperature of output water of the cooling tower is enhanced by only 2.6 °C, when the air relative humidity is 70 percent.

The variations in the input water temperature of the cooling tower in the conditions of applying the hybrid system does affect its performance in reducing the temperature of output water of the cooling tower. Also, according to Fig. 26, it can be concluded that if the air
Fig. 20. Percentage of process water reduction at the outlet of the cooling tower by altering the relative humidity of ambient air in two conditions with and without application of the dehumidifier system.

Fig. 21. Changes of the temperature of output water of the cooling tower with the temperature change of the ambient air at a relative humidity of 70%, in both cases with and without the use of the dehumidifier system.

Fig. 22. Percentage of process water reduction in a cooling tower with environment air temperature change at a relative humidity of 70%, in two modes with and without application of the dehumidifier system.

Fig. 23. Percentage of the water temperature reduction in the cooling tower by changing ambient air temperature in two conditions with and without the use of the dehumidifier system.

Fig. 24. Changes of the temperature of output water of the cooling tower by altering the temperature of input desiccant solution of dehumidifier system.

Fig. 25. Variation of the temperature of output water of the cooling tower as the temperature of the cooling water at the inlet of the dehumidifier is changed.

Fig. 26. Changes in the temperature of output water of the cooling tower by varying the temperature corresponding to the input water in two modes with and without application of the dehumidifier system.

The increase of the input water temperature affects the water mass lost in the cooling tower. Fig. 27 shows the changes in the rate of water loss in the cooling tower at different temperatures of the inlet water, in two modes of using and not using a liquid desiccant system. By enhancing the input water temperature of the cooling tower, the amount of water lost in the tower increases in both cases. As the input water temperature of cooling tower increases, the vapor partial pressure increases in the droplets surface; consequently, the driving force for mass transfer goes up. The rate of water lost in the hybrid system is more comparing with a single cooling tower. Because the inlet air of the cooling tower from the liquid desiccant system has less humidity ratio. Consequently, the rate of evaporation from the droplets surface increases to the air stream.
Fig. 27. Percentage of water temperature reduction at the outlet of the cooling tower by changing the water temperature into the cooling tower in two modes with and without using the dehumidifier system.

6. Conclusion

In this research, a SCT with countercurrent flow pattern and a liquid desiccant system equipped with the internal cooling system used lithium chloride as an adsorbent were modeled separately and mathematically. The governing relations were solved numerically using a finite difference scheme. Considering the influence of air humidity on the air temperature changes during the moisture absorbing system, the precision of numerical modeling was increased, and the results became closer to experimental results. Also, by considering the change in the diameter of the water droplets along the SCT, the precision of the numerical modeling was improved compared to the literature. Then, a new configuration was suggested by the combination of an SCT and a liquid desiccant system as a combined system, and the influence of geometric, physical and environmental parameters on the efficiency of the systems was studied. The outcomes indicated that:

1. By reducing the diameter of the input water droplets of the cooling tower, the temperature of output water of the cooling tower decreases.
2. Increasing the cooling tower height in the hybrid system compared to the single cooling tower has a greater effect on the temperature reduction of the process water at the outlet of SCT.
3. By increasing the height and width of the heat exchanger corresponding to the desiccant system, the temperature of output water of the cooling tower decreases more. It was found that if the environment temperature is 40°C and the relative humidity is 70%, and dehumidifier tower's height increases from 0.5 m to 3.0 m, the temperature of output water of cooling tower decreases about 2.7°C.
4. When the ratio of the water-to-air mass flow rate at the inlet of the cooling tower rises, the temperature of output water of SCT enhances, and the efficiency of the hybrid system reduces. It was also observed that the rate of water loss in a single cooling tower is less compared to the condition at which it is utilized in the hybrid system.
5. When the temperature and relative humidity of the ambient air enhances, the efficiency of the hybrid system for reducing the process water temperature becomes higher than the single cooling tower.
6. The ambient air temperature under conditions of application of the hybrid system has a little impact on the percentage of process water reduction in the cooling tower. Though, in a single cooling tower, changes in ambient air temperature have a larger effect on the percentage of the process water reduction.
7. By increasing the temperature of the desiccant solution and the cooling water entering the liquid desiccant system, the temperature of output water of cooling tower increases in the hybrid system.
8. The temperature of water at the outlet of SCT decreases when it is utilized in the hybrid system.
9. The temperature of water at the inlet of the cooling tower in terms of applying the hybrid system does not affect the efficiency of this tower in reducing the outlet water temperature.

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