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Performance analysis of thermal storage assisted cooling tower with night cooling

Termal depolama destekli soğutma kulesinin gece soğutmalı performans analizi

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Performance Analysis of Thermal Storage Assisted Cooling Tower with Night Cooling

Highlights

- *Integrating two water storage tanks to the new cooling systems enable to use low continuous flow rate.*
- *The new models reduce the mass flow rate to 1/3 and air mass flow rate to 0.69 of the regular models.*
- *The water storage tank capacity can be reduced by increasing the number of tanks.*
- *The new cooling systems provide lower condenser inlet water temperature and higher performance.*

Graphical Abstract

In this study, the low night temperature and thermal storage tanks effects on the cooling tower is studied using TRNSYS.

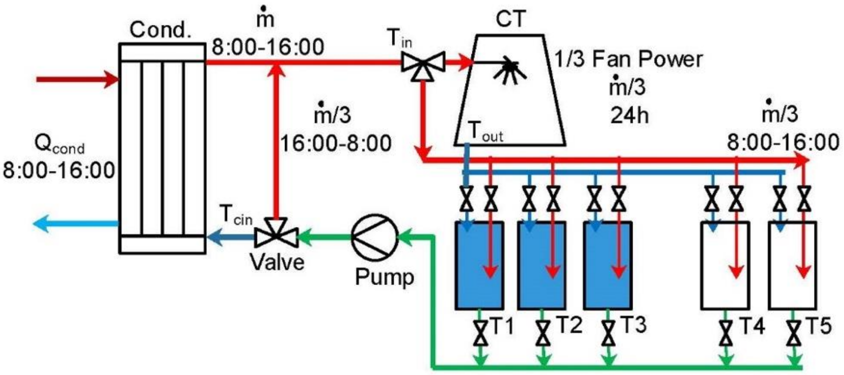

Figure. Proposed five thermal storage tanks cooling system

Aim

The aim of this study is exploring a feasible technique for a cooling system that lowers the cooling water temperature and enhances the performance.

Design & Methodology

Low continuous flow rate and contribution of the number of water storage tanks on the modeled cooling system is analyzed and optimized using TRNSYS.

Originality

This paper explores a feasible technique for a cooling system that lowers the cooling water temperature and enhances the performance.

Findings

The new modeled cooling systems enable to lower continuous flow rate. Their condenser inlet water temperature is reduced and the performance is increased compared to that regular model.

Conclusion

The required storage volume can be reduced up to 50% by using multiple smaller tanks and a control mechanism that alternately fills/drains tanks. For 12 tanks, storage capacity becomes 40% lower than the 2 tanks model. The new modeled cooling systems' condenser inlet water temperature is 20% lower and the performance are 36% higher than regular model.

Declaration of Ethical Standards

The author(s) of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

Performance Analysis of Thermal Storage Assisted Cooling Tower with Night Cooling

Araştırma Makalesi / Research Article

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ABSTRACT

As global warming and water scarcity issues continue to grow, it is essential to increase resources efficiency for air conditioners and power plants. In order to increase the efficiency, the systems need to be modified to take the advantages of the low night temperature and thermal storage tanks. In this study, the low night temperature and thermal storage tanks effects on the cooling tower is studied using TRNSYS. Using a chiller operating from 8:00 to 16:00 as a case study, hot water from the condenser is partially stored on daytime and cooled slowly during the night. The storage tank volume is optimized by considering two big tanks and five small tanks. The results show that night cooling reduces cooling water temperature by 5.8 °C or 21.8% while the cooling efficiency is increased by 36%. The thermal storage tanks enable to have the low continuous flow rate and help to reduce the fan power by 67.1%. On the storage side, compared to two tanks system, the tanks volume is reduced by 16.5% when 5 tanks are used. In theory this reduction can go up to 50% by increasing the number of tanks and reducing their individual size.

Keywords: Cooling tower, thermal storage, night cooling.

Termal Depolama Destekli Soğutma Kulesinin Gece Soğutmalı Performans Analizi

ÖZ

Küresel ısınma ve su kıtlığı sorunları artmaya devam ettikçe, klimalar ve enerji santralleri için kullanılan kaynakların verimliliğini artırmak esastır. Verimliliği artırmak için, düşük gece sıcaklığı ve termal depolama tanklarının avantajlarından yararlanmak için sistemlerde değişiklikler yapılması gerekir. Bu çalışmada, düşük gece sıcaklığı ve termal depolama tanklarının soğutma kulesi üzerindeki etkileri TRNSYS kullanılarak incelenmiştir. Bir vaka çalışması olarak sabah 8'den akşam 4'e kadar çalışan bir soğutucu kullanan kondansatörden gelen sıcak su kısmen gündüz depolanır ve gece boyunca yavaşça soğutulur. Depolama tankı hacmi iki büyük tank ve beş küçük tank dikkate alınarak optimize edilmiştir. Sonuçlar, gece soğutmanın soğutma suyu sıcaklığını 5,8 °C veya % 21,8 oranında azalttığını, soğutma verimliliğinin %36 oranında arttığını göstermektedir. Termal depolama tankları düşük sürekli akış hızına sahip olmayı ve fan gücünün %67,1 oranında azaltılmasını sağlar. Depolama tarafında, iki tank sistemine kıyasla, 5 tank kullanıldığında tank hacmi % 16,5 azalmıştır. Teoride bu azalma, tank sayısını artırarak her bir tank boyutunda ki azalma % 50'ye kadar çıkabilir.

Anahtar Kelimeler: Soğutma kulesi, termal depolama, gece soğutma.

1. INTRODUCTION

Energy consumption of air-conditioning (AC) is increasing globally because of climate change and the growth of urbanization. According to Morna et al. [1], the global residential air-conditioning energy demand is forecasted to increase by 30% for heating and 70% for cooling during the period 2000-2100. In the agriculture sector as well, air conditioning is becoming more and more important as greenhouse farming are promoted worldwide [2]. Xu et al. [3] analyzed subtropical greenhouse cooling performance and realized that daily high temperature of 30 °C - 40 °C combined to a humidity level over 70% were the most challenging parameters for an effective cooling. Nidal et al. [4] lowered the humidity lever using a liquid desiccant to enhance the cooling efficiency of the greenhouse.

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A cooling water temperature above the standard 30 °C design inlet temperature of AC reduces the performance of the machines and drops of coefficient of performance (COP) 1% - 2% every per degree. Three types of cooling techniques are employed to chillers or thermal power plants to remove heat from their condensers: dry cooling (air cooled), liquid cooling (water cooled), and evaporative cooling [5]. Dry air cooling has the advantage of using free air but has poor performance when the ambient temperature is high. Fan energy consumption is also a drawback for dry cooling. Liquid cooling using a large body of water has a stable performance as cooling temperature is nearly constant. However, not all applications are located near a large water body such as a river or a sea. In the evaporative cooling technique, cooling effect in the cooling tower (CT) is achieved by evaporating a relatively small fraction (1% to 5%) of the water flowing through the system [6]. In the evaporation process, latent heat is removed by the water. Depending on the air-water temperature difference, sensible heat may also be removed by water. The water consumption as well as fan energy consumption are the drawbacks of evaporative CTs [7–9].

Ayoub et al. [10] studied the effect of long-term climate change on thermal power plant performance in France and concluded that increment in the air temperature will drop the efficiency of evaporative cooling tower that lead to a 50% loss in power generation. Following a system upgrade cost analysis, they found out that building additional cooling towers to meet the plant temperature requirement is not economical. The only viable solution is to redesign the cooling tower to operate at higher temperature/humidity. Condensers in cooling systems or thermal power plants got more attention since its mismanagement can drastically reduce the performance of the systems [11].

Night cooling has already been investigated by numerous researchers. Miguel et al. [12] showed that a good slab design for night ventilation is an efficient solution to lower indoor daytime temperature for an office building. In a similar study, Jiang et al. [13] found that night roof ventilation can reduce 10% - 24% of the AC energy consumption while reducing the indoor peak temperature by 3 °C - 5 °C. An experimental study of night sky cooling of water tank by natural convection and sky radiation by Ali [14] showed that water temperature can be lowered by 6.6 °C during the night. However, this result is obtained by keeping the deep of the water at 0.5 m, meaning a large area may be necessary to achieve cooling. Also, water consumption was as high as 4.5%. To avoid the use of water for cooling, Dyreson et al. [15] studied night sky cooling for a CSP plant and found that 80% - 90% of the cooling load can be satisfied with this technique.

In some places where the system is not running 24 hours, the cooling efficiency of a CT can be improved by changing the cooling schedule to take advantage of environmental and resource parameters such as low ambient temperature through the night, off-peak electricity rate, and by using variable speed fan to dynamically optimize fluids flow ratio. In a hybrid wet/dry CT operation, Dehaghani et al. [16] reduced the fan power to 64% and water consumption to 9.4% by using variable speed fans. Naik et al. [17] found that both air/water mass flow rates are proportional to water losses. Belmonte et al. [18] replaced CT of a chiller with a dry cooling device and a phase change material tank in an attempt to stabilize cooling water temperature. The result showed a 21% - 38% decrease in the COP of the chiller when the ambient temperature is greater than 30 ºC.

In the literature, night sky cooling principle whether by convection or sky radiation is limited to passive system operation. This paper attempt to apply night sky cooling principle in an active way through a mechanical draft cooling tower. In the paper, the effects of low continuous flow (LCF) rates on cooling system performance are studied by using TRNSYS. Low night sky temperature and low fan power effect on the cooling efficiency are investigated. System performance obtained by LCF versus intermittent fluid flow rates is analyzed. Also, the storage tanks that allow storing circulating hot and cool water is optimized in term of volume requirement.

2. SYSTEM MODELING & WORKING PRINCIPLE

In this section, three different cooling systems are created and modeled in TRNSYS. It is assumed that the working hours are from 08:00 to 16:00; in other words, the ON time period which space cooling is required. The first system is a regular cooling system having a condenser and a CT. Next two systems are newly modeled cooling systems using the LCF introduced earlier.

2.2. **Model 1** (Regular System)

Figure 1 A regular cooling system formed by a condenser, a CT, a pump and connecting pipes. Since the goal in this study is to lower cooling water temperature, the internal configuration of the condenser is not taken into account. It is assumed that the heat transfer area of the condenser as well as its material is designed to exchange the required heat load, Q_{cond}.

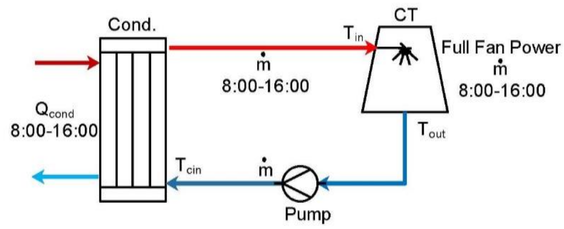

Figure 1. Cooling system of model 1

A mechanical draft CT is used in this study. Warm water entering the CT is spread over a fill to increase the contact area between water and the air being forced through the fill by the electric fan. In this regular cooling system, the CT and the condenser are switched ON/OFF simultaneously during the operation hours. This is a single loop intermittent flow system. From 08:00 to 16:00, the condenser is under heat load ($Q_{cond} = 22.5\,kW$). The pump maintains a mass flow rate ($\dot{m}$ =2000 kg/h) through the system while the CT fan is kept at its rated speed with a power of 3 kW. From 16:00 to 08:00, the condenser heat load is zero meaning the chiller is turned OFF. This simultaneous start-stop is the typical operation mode of commercial or industrial cooling system.

2.2. **Model 2** (Two Tanks System)

The system of model 1 has been modified by adding 2 identical tanks (T1 and T2) as illustrated in Figure 2. The rest of the components are the same as the previous model. The working principle is explained as follows: In the new system, tank T1 (cool tank) contains cool water with a mass M_i (volume V_i) and tank T2 (Hot tank) is empty at 08:00. The condenser is under a base heat load $Q_{cond} = 22.5\ kW$ from 08:00 to 16:00. Hot water from the condenser ($\dot{m}$ =2000 kg/h) is partially ($2\dot{m}/3$) stored in tank T2, the remaining ($\dot{m}/3$) being cooled in the CT then discharged in tank T1. The condenser doesn't operate from 16:00 to 08:00. During these hours, hot water in tank T2 is cooled at a rate of $\dot{m}/3$ and stored in tank T1 for the next cycle. In this system, LCF is achieved by reducing the mass flow rate through the CT to 1/3 of the flow rate of the regular system. Also, the air mass flow rate is reduced by setting the fan power to 1/3 of its rated value. In model 2, tank T1 initially stores water of mass M_i for only 6 working hours with a flow rate $\dot{m}$ for the cooling system. During this 6 hours period, CT provides cool water for the reaming 2 hours in order to complete the ON time period of 8 hours. This is done by cooling $\dot{m}/3$ of hot water out of the condenser. The total required cooling water for the condenser $\dot{m}\Delta t_{ON}$ can be expressed as:

$$\dot{m}\Delta t_{ON} = M_i + \tfrac{1}{3}\dot{m}\Delta t_{ON} \tag{1}$$

where Δt_{ON} is the working hours, M_i the initial water mass in tank T1. Since, it is assumed all cool water in T1 has been used by 16:00, the initial cool water M_i in the storage tank T1 can also be regarded as being produced

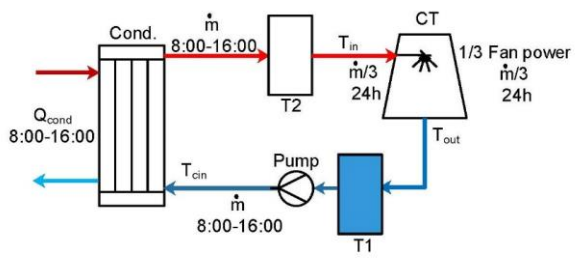

Figure 2. Cooling system of model 2

only during the OFF-time period (16:00-08:00) as given in the equation below.

$$M_i = \tfrac{1}{3}\dot{m}\Delta t_{OFF} \tag{2}$$

where Δt_{OFF} is the total OFF time hours. The volume of cool water at the end of the condenser working period must be as low as possible. The capacity of each tank is $M_i = 2\dot{m}\Delta t_{ON}/3$. For $\dot{m}$ =2000 kg/h, this corresponds to M_i = 10666.67 kg or V_i =10.67 m^3 of water.

2.3. Model 3

Model 2 enables to reduce the fan power and flow rate to 1/3 compared to model 1. However, it requires 2 identical storage tanks, each with a volume of V_i. Model 3 proposes a solution to reduce the total storage tank volume of model 2. Thus, the total mass capacity of tanks $2M_i$ is reduced in the new model. In terms of flow rates and fan capacity, model 3 works as model 2. The idea is to reduce the tanks volume by using multiple smaller size tanks. In this model, in addition to three water-filled tanks which total a capacity of M_i as in model 2, two identical empty tanks are included to store the discharged cool water from the CT ($\dot{m}/3$) as well as hot water from the condenser ($2\dot{m}/3$). Individual tank and total tanks capacities can be calculated as:

$$M_{1\text{tank}} = \frac{M_i}{n} \tag{3}$$

$$\text{Total tanks capacity} = (n+2)M_{1\text{tank}} \tag{4}$$

Since preliminary calculation showed that tanks number affect only the storage volume but not the cooled water average temperature, calculations of model 3 have been done for a total 5 storage tanks for simplicity of charge/discharge scheduling. Figure 3 shows the new proposed model and the added 5 storage tanks. First, three tanks (T1, T2, and T3) are filled with water and colored in blue at 08:00. Next, two tanks (T4 and T5) are empty and colored in white.

Figure 4 shows the flow schedule of tanks and whether they are filled with hot or cool water. Each column represents the state of the tank at different time of the day. Flow schedule for the tanks helps to use LCF and low fan power while keeping the required tanks capacity at a

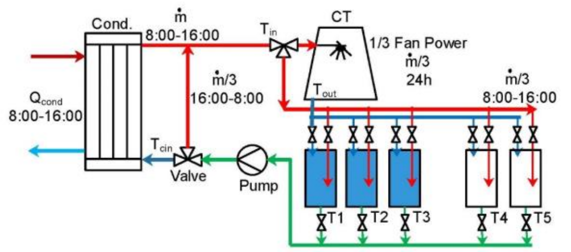

Figure 3. Cooling system of model 3

minimum. The beginning of the day at 08:00, the first three tanks are filled with cool water and the remaining two tanks are empty. During the first two hours, cool water at first tank is circulated through the cooling system at $\dot{m}$ mass flow rate. Hot water from the condenser is cooled and stored in tank number 4 with the mass flow rate $\dot{m}/3$. The remaining $2\dot{m}/3$ hot water is stored in tank 5. This process is continued for all tanks until 16:00. After 16:00, the condenser operation is stopped, and CT cools all the stored hot water until 08:00 of the next day.

3. CT SIMULATION TOOL AND PARAMETERS

TRNSYS software is used to model the proposed cooling systems. The software is well suited for thermal system simulation as its library has all the necessary thermal components. The weather data for the first week of July

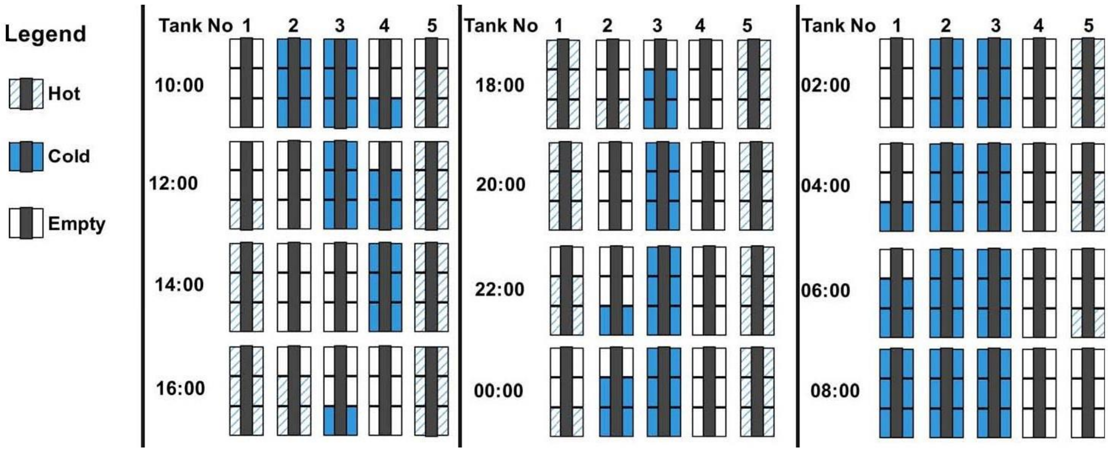

Figure 4. Flow schedule schema of cooling system of model 3

for Izmir, Turkey has been used for the simulation because it is within the hottest period of the year for the selected location. Figure 5 shows the weather data obtained from TRNSYS for the given location through the month of July.

The modeled CT has been calibrated to cool water from

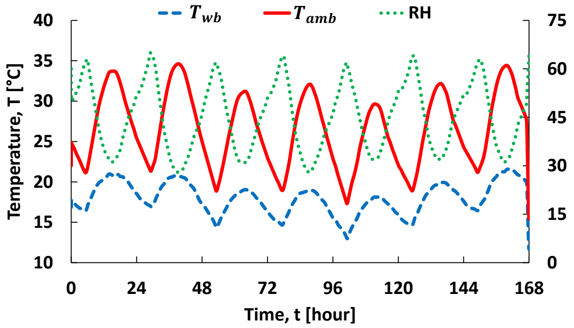

Figure 5. Meteorological ınput data

40 °C to 30 °C at 25 °C wet bulb temperature. Heat load is taken as the sum of two components; a base load of 22.5 kW and 0.5 kW per degree above 25 °C as given in the following equation.

$$Q_{cond} = Q_{base} + 0.5(T_{amb} - 25) \quad \text{for } T_{amb} > 25\ °C$$
$$Q_{cond} = Q_{base} \quad \text{for } T_{amb} <= 25\ °C \tag{5}$$

This heat rejection rate represents the condenser heat load of a 10 kW cooling capacity water-lithium bromide absorption chiller. Using equation 5 and ambient temperature data from figure 5, the heat load profile of the condenser for the given location is shown in figure 6.

An energy balance and predefined parameters are the basis of condenser analysis [19]. A shell-and-tube heat exchanger is modeled as a condenser. A simplified model for shell-and–tubes heat exchanger developed by Vera-Garcia et al. [20] is adopted. The maximum heat transfer Q_{max} in the heat exchanger is

$$Q_{max} = m_w c_{pw}(T_{vi} - T_{wi}) \tag{6}$$

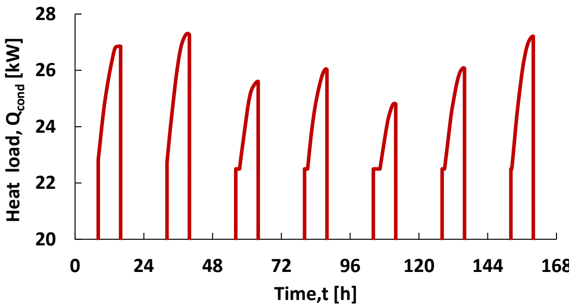

Figure 6. Condenser daily heat load

where m_w, c_{pw}, T_{vi}, and T_{wi} are water mass flow rate, water specific heat, the water vapor and water inlet temperatures, respectively. The heat transfer for the condenser, Q_{cond} is related to Q_{max} and calculated as

$$Q_{cond} = \varepsilon Q_{max} \tag{7}$$

where ε is the heat exchanger effectiveness assumed to be $\varepsilon = 0.8$. The latent heat removed by the condenser is

$$Q_{cond} = m_v h_{fg} \tag{8}$$

where m_v is the vapor flow rate and hfg water latent heat of condensation. Finally, the cooling water outlet temperature T_{wo} is calculated by

$$T_{wo} = T_{wi} + \frac{Q_{cond}}{m_w c_{pw}} \tag{9}$$

Following the condenser analysis, CT is the next component of interest in the cooling system. According to the CT report of SPX cooling technologies company [21], the CT fan is sized at 0.85 hp/ton of cooling capacity of a vapor compression cooling system. Taking into account the COP of these devices (assumed COP = 4), the heat rejection rate is 1.25 ton. By assuming a COP of 0.5 for an absorption chiller and using the sizing scheme of 0.85 hp/1.25 ton of heat rejected, a 3 kW fan is used for the CT under study. In the results, the fan power of the proposed models is expressed as a fraction of the regular system fan power by using the fan law given in Fantech datasheet, rewritten as [22]

$$P_2 = P_1 \left(\frac{\dot{V}_2}{\dot{V}_1} \right)^3 \tag{10}$$

where P and V are fan power and air volume flow rate at the specific state of operation. This makes the sizing accuracy to visualize the exact effect of fan speed reduction on system performance. Table 1 summarizes the CT parameters. The following assumptions have been made to simplify the study of the systems:

-Water temperature in the tanks is uniform

-Tanks heat loss/gain is negligible

-Cooling tower parameters change negligibly within the operation range

-Makeup water temperature is equal to the ambient temperature

Table 1. Cooling tower parameters

	Model 1	Model 2 & 3
Flow geometry	Counterflow	Counterflow
Water flow rate	2000 kg/h	666.67 kg/h
Cell rated air flow	1280 m^3/h	1280 m^3/h
Mass transfer coefficient [4]	1.96	1.96
Mass transfer exponent	-0.69	-0.69
Air velocity ratio	1	0.69
Power	3 kW	3 kW

4. COOLING SYSTEM PERFORMANCE PARAMETERS

The performance of CT is commonly measured through its cooling efficiency. The cooling efficiency, η defined in equation 11 is a measurement of how much the CT outlet temperature is close to the theoretical possible temperature which is the wet bulb temperature. In the literature, the CT inlet-outlet temperature difference is known as its range and the temperature difference between the outlet and the wet bulb temperature is known as the approach. With these definitions, the efficiency of CT is the ratio of the range over the sum of the range and the approach as given as

$$\eta = \frac{T_{in} - T_{out}}{T_{in} - T_{wb}} \tag{11}$$

On the fan side, the ratio of the heat rejection by the condenser, Q_{reject} to per unit work input of the fan, W_{fan} gives the information about the electric energy performance of the cooling system, Φ.

The water loss rate from the CT, wl is considered as a performance indicator as well. For applications that cannot rely on abundant seawater, the water loss rate can increase the water expenditure.

5. RESULTS AND DISCUSSION

5.1. Storage Volume Size Optimization

Figure 7 shows the variation of single tank capacity and the total storage capacity along with the number of tanks for the system of model 3. Model 2 uses only two tanks and the total required storage capacity is $2V_i$. Even though, the control scheme of model 3 suggests to reduce the storage capacity, it is increased to $3V_i$ when only 3 tanks are used. This happens because of the change in the operation mode; two empty tanks being added. Effective volume reduction is obtained by using at least 5 tanks. With 5 tanks, the capacity becomes $1.67V_i$ in model 3 against $2V_i$ in model 2. As the number of tanks further increases, storage capacity is getting reduced toward V_i which is the theoretical minimum capacity. However, splitting water into more tanks reduces drastically the size of the single tank which may become expensive or impractical to use. The single tank size is only 10% of V_i with 12 tanks. Given the mass flow rate in this study, this represents 12 tanks of 1.067 m^3 each.

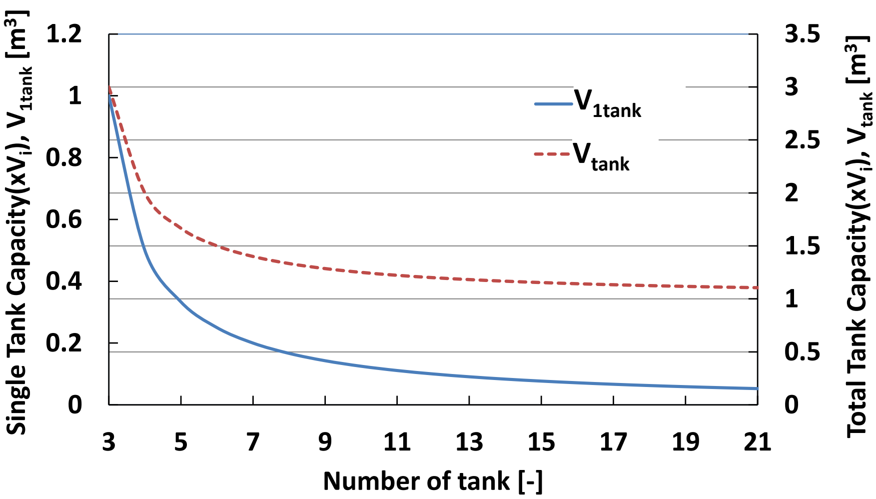

Figure 7. Storage capacity as a function of number of tanks

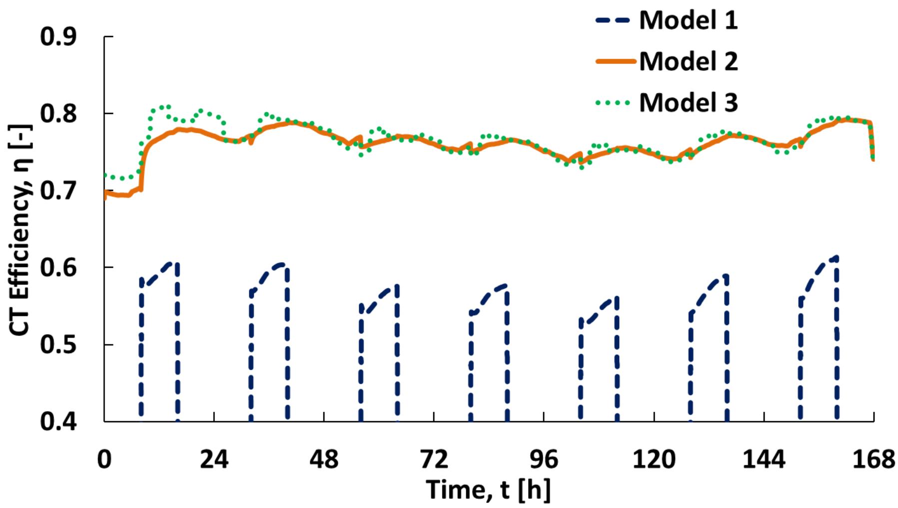

Figure 8. Cooling tower efficiency

5.2. Thermal Response

Figure 8 shows the cooling efficiencies of CT for the three systems. The results show that the cooling efficiencies of model 2 and 3 are approximately 36% higher than that of model 1. Furthermore, unlike model 1, the cooling efficiencies curves for model 2 and 3 are nearly flat during the month since they are working 24 hours/day. Model 1 cooling efficiency is rising steadily from 08:00 to 16:00.

Figure 9 shows the changes in the heat rejection per unit fan power of the systems with time. The heat rejected per unit fan input power for model 2 and 3 is greater than that of model 1. The results show that more heat is rejected per unit fan power during the OFF time. In fact, the wet bulb temperature is lower during the OFF time as shown in the weather data in figure 5, which allows more heat rejection. Still, from the weather data, the low night ambient temperature contributes to the lower CT outlet temperature since water temperature is lowered by the action of both latent heat loss and sensible heat loss. Ultimately the CTs outlets temperatures must be lower as well.

In table 2, both models 2 and 3 have a condenser inlet water temperature around 20% lower than model 1. Model 2 and 3 give nearly the same average temperature results which means that using multiple tanks only reduce the total storage volume. Using more tanks gives less uniform curves since each tank has a distinct discrete temperature profile. Additionally, the water mass flow

Table 2: Condenser inlet temperature and fan power

	Model 1	Model 2	Model3
Condenser inlet, T_{wi} [C]	26.9	21.03	21.6
T decrease [%]	-	21.82	19.70
Fan power [kW]	3	1	1
Fan power decrease %	-	67.16	67.16

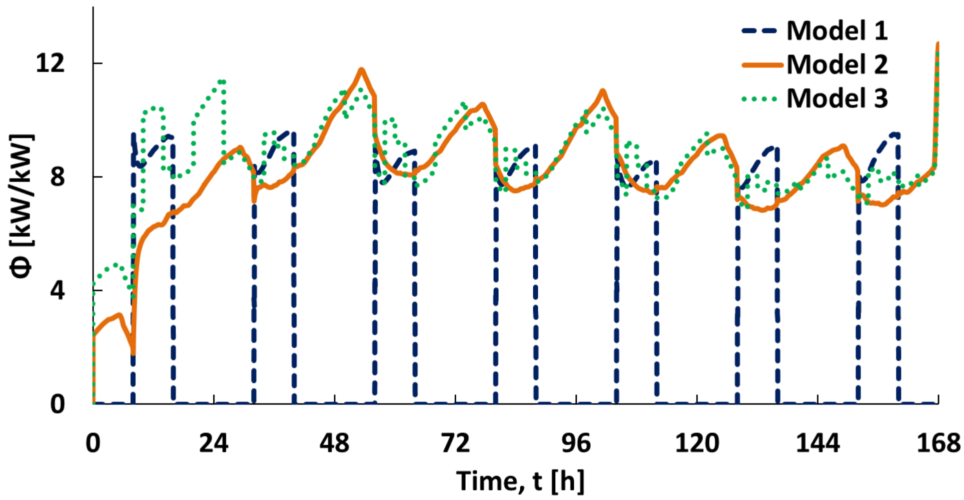

Figure 9. Heat rejection to fan power ratio as a function of time

rate of model 2 and 3 is reduced to 1/3 and the air mass flow rate is reduced to 0.69 of the value of model 1. The NTU being a function of the ratio $\dot{m}_{water}/\dot{m}_{air}$ in CT, reducing this ratio in turn increases the NTU of the CT which increases the cooling efficiency. The ratios of $\dot{m}_{water}/\dot{m}_{air}$ for model 2 and 3 are reduced to 0.47 of its reference value in model 1. Physically, the higher relative air velocity in model 2 and model 3 removes the humid air at the water-air interface faster, creating a higher potential for mass transfer by evaporation.

5.3. Water Loss And Fan Power

There is no significant difference in water loss for all 3 models as illustrated in figure 10. The average water losses are 1.64%, 1.66% and 1.74% for the model 1, 2, and 3 respectively. However, a small amount of water loss has a significant effect on temperature drop since the loss is accompanied by latent heat removal. Also, from the weather data in figure 5, the lowest relative humidity coincides with the peak ambient temperature which gives the dry air more water absorption potential. As a result of the increasing ambient temperature, water will be absorbed easily. For model 2 and 3, fan power has been reduced by 67% as stated in table 2. This result was expected since fan power drops from 3 kW to 1 kW when air velocity is reduced from 100% to 69% according to fan law.

6. CONSLUSION

The cooling performance of regular cooling tower that switches ON/OFF simultaneously with a condenser has been compared with a LCF cooling system. From the two tanks operation, the required storage volume can be reduced up to 50% by using multiple smaller tanks and a control mechanism that alternately fills/drains tanks. As the number of tank increases, they become smaller and may represent handling or economical drawbacks. For 12 tanks, storage capacity becomes 40% lower than the initial 2 tanks model. This can be a good arbitrary cut off number in this study if a practical implementation is sought. The number of tanks appears not to have an impact on the thermal response of the system.

When the working hours considered, the heat rejection to fan power ration for three systems shows similar values; 8.46, 8.11, and 8.57. Even though the fan power is 3 kW in model 1 and 1 kW in model 2 and 3, since model 1 working hours is 8 hours and model 2 and 3 working hours is 24 hours, the total electric power consumption of fans for all three systems is 24 kWh. The amount of rejected heat from CT is 203.04 kWh for model 1, 196.64 kWh for model 2, and 205.68 kWh for model 3. The power consumption of the pump has not been recorded since the power being function of flow rate and reducing the flow rate to 1/3 will reduce pump power consumption.

The most significant outcome of the study is the reduction of condenser inlet temperature by 5.8 ºC or 21.8%. For a heat driven engine, this translates into a system overall efficiency improvement as the heat source and the heat sink temperature gap gets wider. Therefore, using low night temperature is an efficient way to boost the performance of thermal devices. This is done with no penalty of water loss which is nearly the same for both systems. Both sensible heat and latent heat can be removed from the water at night while only latent heat can be significantly removed on daytime because of the high ambient temperature. From the power side, delayed cooling has the advantage of running at 33% of the regular CT fan power. For a big cooling tower, this can represent a saving in power demand charge. From an energy point of view, there is no saving since fans in model 2 and 3 are running 24 h/day. However, fan in the proposed models can be replaced by a smaller one to save money and improve electrical performance.

The disadvantage of delayed cooling is the requirement for storage tanks. The working principle can be used for others applications where storage is freely available or where the condenser heat is valuable. Cogeneration system which outputs cool water for the condenser and thermal energy for domestic hot water production for

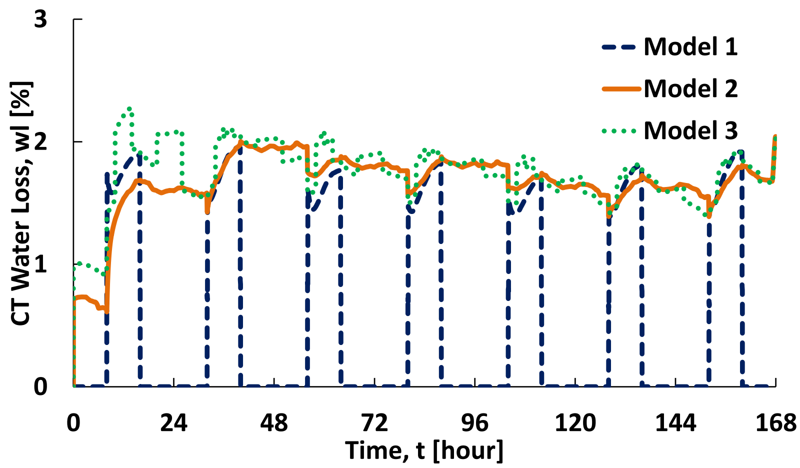

Figure 10. Cooling tower water loss rate

Nomenclature

COP	Coefficient of performance [-]	T_{vi}	Water vapor inlet temperature [ºC]
$\dot{m}_{air}$	Air mass flow rate [kg/h]	T_{wb}	Wet-bulb temperature [ºC]
$\dot{m}_{w}$	Water mass flow rate [kg/h]	T_{wi}	Water inlet temperature [ºC]
$\dot{m}_{v}$	Water vapor mass flow rate [kg/h]	W_{fan}	Fan input power [kW]
M_i	Initial stored water capacity [kg]	$\dot{V}_i$	Air volumetric flow rate [m^3/h]
M_{1tank}	Single tank capacity in [kg]	T_{in}	Cooling tower inlet temperature [ºC]
NTU	Number of transfer unit [-]	Δt_{ON}	Condenser ON time [hours]
P_i	Fan power [kW]	Δt_{OFF}	Condenser OFF time [hours]
Q_{base}	Condenser base heat load [kW]	ε	Heat exchanger effectiveness [-]
Q_{cond}	Condenser heat load [kW]	n	Number of tanks [-]
Q_{reject}	Cooling tower rejected heat [kW]	η	Cooling tower efficiency [-]
T_{amb}	Ambient temperature [ºC]	Φ	Heat rejected to fan power ratio [-]
T_{out}	Cooling tower outlet temperature [ºC]		

example can fit such application. In overall, the suggested system reduces significantly the condenser working temperature and fan power at once while keeping water loss negligible.

DECLARATION OF ETHICAL STANDARDS

The author(s) of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission..

REFERENCES

[1] Isaac M., and van Vuuren D. P., "Modeling Global Residential Sector Energy Demand for Heating and Air Conditioning in the Context of Climate Change", ***Energy Policy***, 37(2): 507–521, (2009).

[2] Ghani S., Bakochristou F., ElBialy E. M. A. A., Gamaledin S. M. A., Rashwan M. M., Abdelhalim A. M., and Ismail S. M., "Design Challenges of Agricultural Greenhouses in Hot and Arid Environments – A Review", ***Eng. Agric. Environ. Food***, 12(1): 48–70, (2018).

[3] Xu J., and Li Y., "Experimental Performance of Evaporative Cooling Pad Systems in Greenhouses in Humid Subtropical Climates", ***Appl. Energy***, 138: 291–301, (2015).

[4] Abu-Hamdeh N. H. and Almitani K. H., "Solar Liquid Desiccant Regeneration and Nanofluids in Evaporative Cooling for Greenhouse Food Production in Saudi Arabia", ***Sol. Energy***, 134: 202–210, (2016).

[5] M. R. Susta, "Condensers and Coolings Systems", (2004).

[6] TRNSYS, "Mathematical Reference Volume 5", (2006).

[7] Kumar P., "Cooling Tower", Energy Efficiency in Electrical Utilities Guide Book, (2005).

[8] Liu N., Zhang L., and Jia X., "The Effect of the Air Water Ratio on Counter Flow Cooling Tower", ***Procedia Eng.***, 205: 3550–3556, (2017).

[9] Abdel-Ghaffar Y. E., "Effect of Operating Parameters on the Performance of Counter Flow Type Cooling Towers", ***Industry Education, DepL, Mansoura Univ, New Damietta***, Egypt, (2004).

[10] Ayoub A., and Gjorgiev B., "Cooling Towers Performance in a Changing Climate: Techno-Economic Modeling and Design Optimization", ***Energy***, 160: 1133–1143, (2018).

[11] Cengel Y., and Boles M., Thermodynamics: An Engineering Approach, (2006).

[12] Lança M., Coelho P. J., and Viegas J., "Enhancement of Heat Transfer in Office Buildings during Night Cooling − Reduced Scale Experimentation", ***Build. Environ.***, 148: 653–667, (2019).

[13] Jiang L., and Tang M., "Thermal Analysis of Extensive Green Roofs Combined with Night Ventilation for Space Cooling", ***Energy Build.***, 156: 238–249, (2017).

[14] Ali A. H. H., "Passive Cooling of Water at Night in Uninsulated Open Tank in Hot Arid Areas", ***Energy Convers. Manag.***, 48(1): 93–100, (2007).

[15] Dyreson A., and Miller F., "Night Sky Cooling for Concentrating Solar Power Plants", ***Appl. Energy***, 180: 276–286, (2016).

[16] Taghian Dehaghani S., and Ahmadikia H., "Retrofit of a Wet Cooling Tower in Order to Reduce Water and Fan Power Consumption Using a Wet/Dry Approach", ***Appl. Therm. Eng.***, 125: 1002–1014, (2017).

[17] Naik B. K., Choudhary V., Muthukumar P., and Somayaji C., "Performance Assessment of a Counter Flow Cooling Tower – Unique Approach", ***Energy Procedia***, 109: 243–252, (2017).

[18] Belmonte J. F., Izquierdo-Barrientos M. A., Eguía P., Molina A. E., and Almendros-Ibáñez J. A., "PCM in the Heat Rejection Loops of Absorption Chillers. A Feasibility Study for the Residential Sector in Spain", ***Energy Build.***, 80: 331–351, (2014).

[19] Kasumu A. S., Nassar N. N., and Mehrotra A. K., "A Heat-Transfer Laboratory Experiment with Shell-and-Tube Condenser", ***Educ. Chem. Eng.***, 19: 38–47, (2017).

[20] Vera-García F., García-Cascales J. R., Gonzálvez-Maciá J., Cabello R., Llopis R., Sanchez D., and Torrella E., "A Simplified Model for Shell-and-Tubes Heat Exchangers: Practical Application", ***Appl. Therm. Eng.***, 30(10): 1231–1241, (2010).

[21] Spxcooling.com. Application of Cooling Towers for Free Cooling. [online] Available at: https://spxcooling.com/library/detail/the-application-of-cooling-towers-for-free-cooling [Accessed 24 Jan. 2019].

[22] Fantech.com.au. Fan Laws, (2016). [online] Available at: https://www.fantech.com.au/images/PDF/Catalogue/fanlaws.pdf [Accessed 24 Jan. 2019]