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Sensitivity Studies of a Low Temperature Low Approach Direct Cooling Tower for Building Radiant Cooling Systems

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ABSTRACT

Recent interest in cooling towers as a mechanism for producing chilled water, together with the evolution of radiant cooling, have prompted a review of evaporative cooling in temperate maritime climates. The thermal efficiency of such systems is a key parameter, as a measure of the degree to which the system has succeeded in exploiting the cooling potential of the ambient air. The feasibility of this concept depends largely however, on achieving low approach water temperatures within an appropriate cooling tower, at acceptable levels of energy performance. Previous experimental work for a full scale evaporative cooling system has shown that it is possible to produce cooling water at low process approach conditions (1-3 K), at the higher temperatures required in radiant and displacement systems (14-18°C), with varying levels of annual availability in different temperate climate locations. For such conditions, evaporative cooling has the potential to offer an alternative approach for producing chilled water, particularly in temperate climates, where conventional mechanical air-conditioning systems can, for certain buildings, be considered to be an over engineered solution but where passive cooling is insufficient to offset cooling loads. The current paper describes the development of a mathematical model which analyses the behavior of a low approach open evaporative cooling tower. The model is used to carry out a series of sensitivity studies assessing the performance of the cooling tower subject to various weather and climatic boundary conditions.

1. INTRODUCTION

More than 40% of energy consumption in commercial buildings, especially in offices, can be attributed to HVAC systems. This figure varies for different countries and different types of commercial buildings, for instance energy consumption of HVAC systems in offices in the UK and the US accounting for 55% and 48%, respectively (Eicker 2009, Pérez-Lombard et al. 2008, Novoselac and Srebric 2002). Moreover, the rapidly growing use of air-conditioning systems in temperature climates as a major contributor to electrical energy consumption has lead to increased concerns in this matter. In the European Union (EU), for instance, it is reported the growth in central air-conditioning floor area has increased by 200% from 1990 to 2002 and this trend is predicted to continue by 60% to 2020 (Adnot 2002). Current predictions estimate that electricity consumption associated with the conditioning of buildings in the EU will increase to 50% by 2020 (Eicker 2009). On the other hand, the increased probability of higher ambient temperatures in the future due to global warming, leading towards hotter summers, is likely to lead to greater cooling demands in buildings in hitherto temperature climates (Poirrit et al. 2012, Smith et al. 2011). Consequently providing more effective cooling systems is crucial for reduction of energy consumption and life-cost of buildings.

Recent interest in radiant cooling systems and displacement ventilation, as a means of building cooling in temperate climates, has prompted renewed interest in evaporative cooling systems or cooling towers for generating cooling water (Dieckmann et al. 2009, Olesen 2008, Harvey 2006, Strand 2003). Evaporative cooling has the potential to offer an efficient approach for producing chilled water for such systems, particularly in temperate climates, where conventional mechanical air-conditioning systems are, for certain buildings, sometimes considered to be an over engineered solution and where passive cooling is insufficient to offset cooling loads (Alexander and O'Rourke 2008, Strand 2003). Water-side evaporative cooling systems are most effectively utilized when integrated with either a chilled floor or ceiling systems and in certain cases a displacement ventilation system, due to the higher cooling water temperatures (14-18˚C) associated with these systems (Costelloe and Finn 2003).

In the current paper, sensitivity studies utilizing a mathematical model of an open cooling tower are described. The model is based on the ε-NTU method for modeling tower heat and mass transfer, but also takes into account the water loss due to evaporation in the tower. The mathematical model is evaluated against experimental data taken from an open tower (Costelloe and Finn 2009, Simpson and Sherwood 1946) and is described in detail elsewhere (Nasrabadi et al. 2012). The sensitivity studies are aimed at assessing the performance of the cooling tower subject to different weather boundary, thereby examining its performance subject to different weather and climatic conditions.

2. LITERATURE REVIEW

There are various well-known methods for modeling the heat and mass transfer processes associated with wet cooling towers including: the Merkel method, the Poppe method and the effectiveness–NTU method (Jaber and Webb 1989, Kloppers and Kroger 2005). The earliest practical use of differential equations to describe the heat and mass transfer within cooling towers was presented by Merkel (Kloppers and Kroger 2005). The Merkel method makes use of some important assumptions (Kloppers and Kroger 2005, Khan and Zubair 2003) including: assuming that the Lewis factor is equal to unity, assuming that the air exiting from the tower is saturated, neglecting the reduction of water flow rate due to evaporation in the tower water mass balance and assuming that the specific heat capacity of air-stream mixture at constant pressure is same as that of the dry air.

Although the effectiveness-NTU relied on the same simplifying assumptions as outlined in the Merkel method, the Poppe model, developed by Poppe and Rogener (1991), does not use these assumptions (Kloppers and Kroger 2005). In Poppe method, the Lewis factor estimated according to the Bosnjakovic relation and then by using a fourth order Runge-Kutta approach water outlet temperature and heat transfer calculated (Kloppers and Kroger 2003). It is shown that the results of Poppe method are better than the Merkel or ε-NTU method for high temperature heat rejection applications (Kloppers and Kroger 2003 & 2005).

Khan and Zubair (2003) investigated the effect of the ratio of mass flow rate of water to mass flow rate of moist air ($\dot{m}_w/\dot{m}_{moist\ air}$) on the performance of a wet cooling tower. The tower, which had a Lewis factor equal to 0.9 and where its NTU was estimated by means of an empirical equation based on the measurements of Simpson and Sherwood (1946). Boundary conditions included inlet air dry bulb and wet bulb temperatures of 29°C and 21.1°C, respectively and the inlet water temperature was 28.7°C (Khan and Zubair 2003). It is shown in different values of mass flow rate ratio, latent heat transfer or evaporation dominated the total heat rejection and this factor increased as $\dot{m}_w/\dot{m}_{moist\ air} \geq 1$ (Khan and Zubair 2003).

A full scale evaporative cooling system including: an open counter flow cooling tower, a primary and secondary circuit with an intermediate heat exchanger (Fig. 1) was designed by Costelloe and Finn (2003). It is shown this system can successfully deliver high temperature (14-18°C) chilled water for radiant cooling or chilled ceiling cooling systems which are used in temperate climate such as Dublin, Ireland (Costelloe and Finn 2003). The performance of this system on different water flow rate is investigated and it is indicated the COP of the cooling tower varied between 6 and 16, which are better than the average COP of standard air cooled machines (Costelloe and Finn 2007).

3. MATHEMATICAL MODEL

The mathematical model used to carry out the sensitivity studies described in the current paper is outlined in detail elsewhere (Nasrabadi et al. 2012). Briefly, the model is based on a mechanical draft counter-flow cooling tower, as shown in Fig. 1, using a *corrected effectiveness-NTU* approach. In this method, the Lewis factor is assumed to unity and the exiting air is considered saturated, but in contrast to standard effectiveness-NTU method (Kloppers and Kroger 2003) or the Merkel method (Kloppers and Kroger 2003), the change in tower water mass flow rate due to evaporation is taken into account and the specific heat of air-stream mixture is assumed not be constant and it is estimated in each step of the numerical calculation.

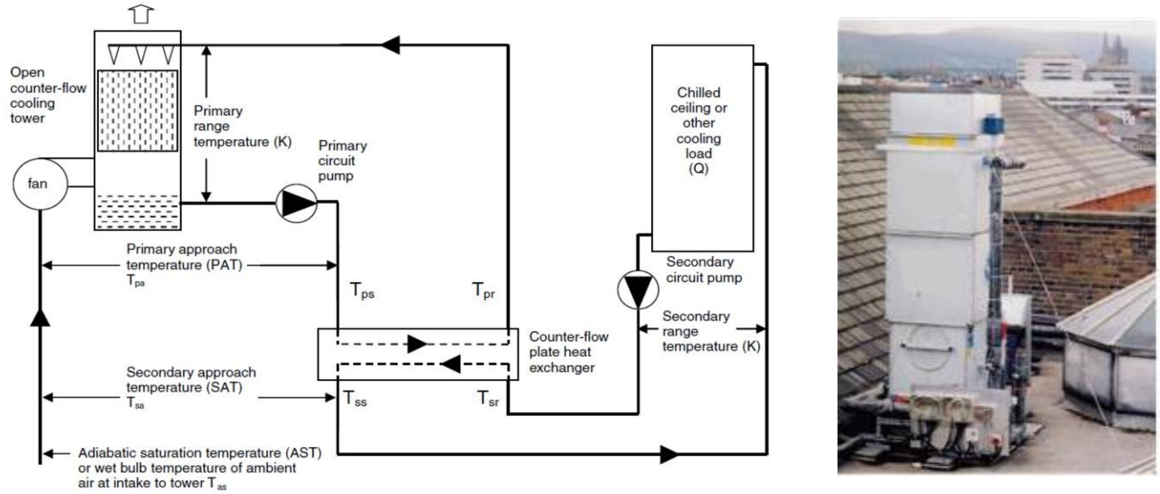

Figure 1: Schematic of an indirect evaporative cooling system and the associated cooling tower (Costelloe, 2009)

Equation 1, which is used to determine the heat transfer rate in the Merkel and ε-NTU methods, can be written as follows:

$$\dot{Q} = \dot{m}_w c_{pw}(T_{wi} - T_{wo}) \tag{1}$$

As can be seen in equation (1), the loss of water mass flow rate is ignored and thus in contrast to Poppe method, the Merkel and ε-NTU methods predict lower heat rejection rate (Kroger 2004). In order to correct for the Merkel assumption that ignores the evaporation of cooling tower water, the mass flow rate due to evaporation is calculated as follows:

$$\dot{m}_{evap} = \dot{m}_a \times (\omega_o - \omega_i) \tag{2}$$

Using Equation 2, Equation 1 can be modified to account for tower evaporation, as follows:

$$\dot{Q} = \dot{m}_{wi} c_{pw} T_{wi} - (\dot{m}_{wi} - \dot{m}_{evap}) C_{pw} T_{wo} \tag{3}$$

When Equation (3) is used for the calculation of heat transfer, estimates by the improved Merkel method and Poppe method are quite similar (Kloppers and Kroger 2003).

The wet cooling tower coefficient or the Merkel number (Me) is an important factor for calculation of the number of transfer units in cooling towers. Various researchers have examined this issue, including Khan and Zubair (2001). Khan and Zubair showed that the cooling tower coefficient can be calculated by:

$$Me = KaV/L = c \times \left(\frac{m_w}{m_a}\right)^n \tag{4}$$

where c and n are empirical constants. Costelloe and Finn (2009) developed an empirical relation for the wet cooling tower coefficient or the Merkel number and for the open cooling tower described in Fig. 1. It was found to be:

$$Me = 1.3 \times \left(\frac{m_w}{m_a}\right)^{-0.77} \tag{5}$$

When the inlet conditions, including the mass flow rate of water and air, the dry bulb and wet bulb temperatures of inlet air and the inlet water temperature are known, the outlet water temperature and the outlet air temperature can be calculated, thereby allowing the effectiveness and heat transfer for the cooling tower to be determined.

Tables 1 and 2 show the results using the current model. The experimental data in Table 1 are taken from the mechanical draft tower, as shown in Figure 1 and is described elsewhere by Costelloe and Finn (2009). Using the data from Table 1, a comparison between the calculated outlet water temperatures and the experimental data is presented in Figure 2. The largest difference in calculation of water temperatures can be seen to be 0.28 K (Table 1), equivalent to an error of less than ±2.5% (as per Figure 2), referenced with respect to 0^oC.

Table 1: Comparison between the results of model and experimental data (Costelloe and Finn, 2009)

Measured data by Costello & Finn (2009)					Results from current model	
Test No	**DBT (^oC)**	**RH (%)**	**Water Inlet Temp (^oC)**	**Water Outlet Temp. (^oC)**	**Water Outlet Temp. (^oC)**	**Water Outlet Temp. Difference**
1	14.09	57.69	15.06	12.89	12.82	-0.07
2	15.36	51.61	16.21	13.37	13.24	-0.13
3	14.99	60.18	14.61	12.69	12.89	0.20
4	14.98	60.95	14.93	12.70	12.86	0.16
5	15.86	57.0	15.30	13.00	13.28	0.28
6	13.95	66.67	15.11	12.35	12.23	-0.12
7	14.26	64.31	15.16	12.36	12.42	0.06
8	14.76	62.93	15.55	12.35	12.37	0.02
9	14.91	59.80	15.97	11.91	11.92	0.01

Table 2 presents measured data for alternative high temperature heat rejection applications in a cooling tower analysed by Simpson and Sherwood (1946). In this table, a comparison between air and water outlet temperature are shown. The largest discrepancy for air and water are 0.83 and 0.57 K respectively, which is equivalent to less than ±2.4% (Figure 2).

Table 2: Comparison between current model and measured data (Simpson and Sherwood, 1946)

Measured data by Simpson and Sherwood (1946)								Results from current model			
Case	**$m_{a,in}$ (kg/s)**	**$m_{w,in}$ (kg/s)**	**$T_{db,in}$ (^oC)**	**$T_{wb,in}$ (^oC)**	**$T_{w,in}$ (^oC)**	**$T_{w,out}$ (^oC)**	**$T_{wb,out}$ (^oC)**	**$T_{w,out}$ (^oC)**	**Temp. Dif.**	**$T_{wb,out}$ (^oC)**	**Temp. Dif.**
1	1.158	0.754	34.11	21.11	41.44	26	31.16	26.29	0.29	30.49	-0.67
2	1.187	1.259	29	21.11	28.72	24.22	26.17	24.78	0.56	25.59	-0.58
3	1.187	1.259	30.50	21.11	34.50	26.22	29.90	26.61	0.39	29.32	-0.58
4	1.2653	1.008	35	26.67	38.78	29.33	32.89	29.87	0.54	32.45	-0.44
5	1.250	1.008	35	26.67	38.78	29.33	32.89	29.90	0.57	32.49	-0.40
6	1.1871	1.0088	28.83	21.11	33.22	25.50	28.44	25.26	-0.24	27.97	-0.47
7	1.1653	1.0088	31.78	26.67	34.39	29	31.22	28.95	-0.05	30.69	-0.53
8	1.1584	0.7548	35	23.89	43.61	27.89	32.78	28.24	0.35	32.48	-0.3
9	1.2653	1.0088	35	26.67	38.78	29.33	33.28	29.87	0.54	32.45	-0.83
10	1.1566	0.7548	35.72	26.67	43.06	29.72	33.89	29.93	0.21	33.46	-0.43

4. SENSITIVITY STUDIES

The mathematical model described in Section 3 was used for a series of sensitivity studies aimed at examining the performance of the cooling tower for different climatic design conditions that could be expected for temperate climates. In order to study the performance of such a low temperature cooling tower subject to different ambient conditions, the necessary boundary conditions should first be determined. A control volume of cooling tower with boundary conditions is shown in Figure 3. Applying the first law of thermodynamics to the tower under conditions of steady state steady flow (SSSF), the following expression can be written:

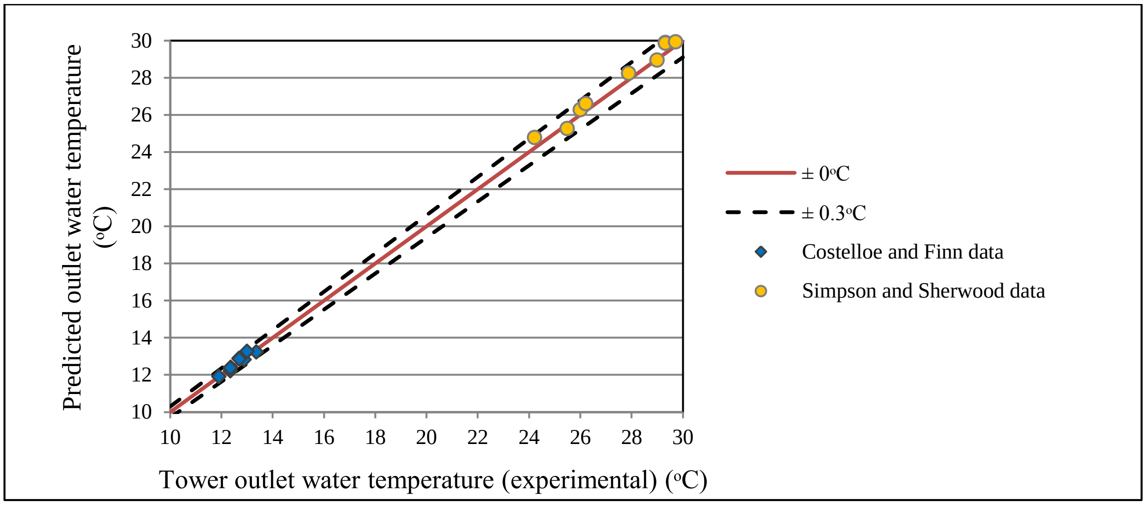

Figure 2: Comparison of tower water outlet temperature against experimental data (Costelloe and Finn 2009, Simpson and Sherwood 1946)

$$\dot{m}_w \times i_{water\_in} + \dot{m}_{Air\_in} \times i_{Air\_in} = (\dot{m}_{Air\_in} + \dot{m}_{evap}) \times i_{Air\_out} + (\dot{m}_w - \dot{m}_{evap}) \times i_{water\_out} \quad (6)$$

where $\dot{m}_{Air}$ and i_{Air} are the mass flow rate and enthalpy of air-water vapor mixture respectively.

Equation (6) can be rewritten as:

$$\dot{m}_w \times i_{water\_in} - (\dot{m}_w - \dot{m}_{evap}) \times i_{water\_out} = (\dot{m}_{Air\_in} + \dot{m}_{evap}) \times i_{Air\_out} - \dot{m}_{Air\_in} \times i_{Air\_in} \quad (7)$$

where $$\dot{m}_{Air} = (1 + \omega) \times \dot{m}_a \quad (8)$$

and $\dot{m}_a$ is mass flow rate of dry air and ω is humidity ratio.

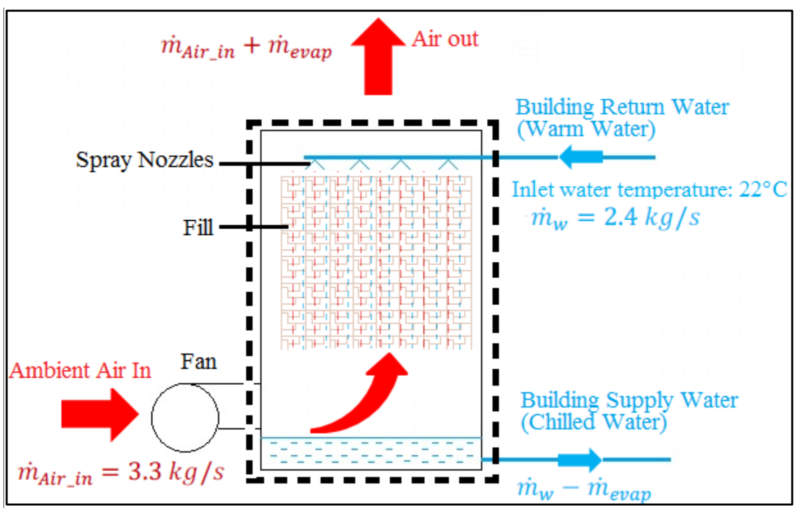

Figure 3: Cooling tower control volume with water and air flow boundary conditions.

The enthalpy of the air-water vapor mixture is defined as follows:

$$i_{Air} = \{C_{pa} \times T + \omega[i_{fgwo} + C_{pv} \times T]\}/(1 + \omega) \quad (9)$$

where C_{pa} and C_{pv} are the specific heat capacities of dry air and saturated water vapour respectively and i_{fgwo} is the latent heat of water at 0°C.

The enthalpy of the air-water vapor mixture per unit mass of dry air is given by:

$$i'_{Air} = C_{pa} \times T + \omega[i_{fgwo} + C_{pv} \times T] \tag{10}$$

The sensible and latent heat transfer associated with the air in the tower can estimated as follows:

$$\dot{Q}_{air\_sensible} = \dot{m}_a \times (C_{pa} + \omega \times C_{pv})_{out} \times T_{Air\_out} - \dot{m}_a \times (C_{pa} + \omega \times C_{pv})_{in} \times T_{Air\_in} \tag{11}$$

$$\dot{Q}_{air\_latent} = \dot{m}_a \times i_{fgwo} \times \omega_{out} - \dot{m}_a \times i_{fgwo} \times \omega_{in} \tag{12}$$

Using the above equations, the performance of the cooling tower as a function of different weather design ambient conditions is considered in the following sections.

4.1 Constant Humidity Ratio

Figures 4 and 5 show cooling tower performance as a function of ambient dry bulb temperature for three different humidity ratios. The humidity ratios can be considered to be representative of the range of design conditions that could be expected for different temperate climates, from dry conditions (low humidity ratio, ω = 4 g/kg) to more humid conditions (high humidity ratio, ω = 12 g/kg). Variation of the dry bulb temperature at constant humidity ratio can be considered to approximate a 24 hour diurnal variation between day and night conditions. Examining Figure 4, it can be seen that the tower chilled water outlet temperature tracks the wet bulb temperature closely, thereby ensuring that the primary approach temperature is almost constant, although it can be observed to decrease slightly with increasing wet bulb temperature. For drier conditions, a larger approach is evident (5K), with a small approach (1K) evident for conditions where higher humidity ratios are present. It can also be observed that for a humidity ratio of ω = 12 g/kg and associated high wet bulb temperature conditions, the primary approach reaches zero, as the wet bulb temperature converges with the tower entry water temperature, under which conditions the tower cooling capacity declines to zero.

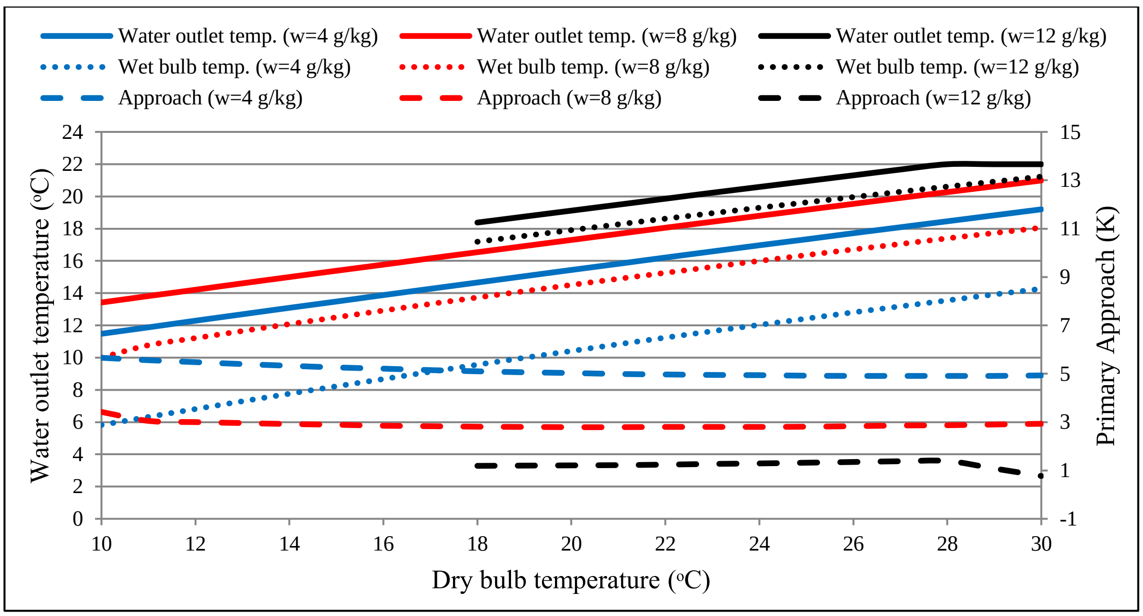

Figure 4: Predicted tower water outlet temperature as a function of ambient air temperature at different humidity ratios ($\dot{m}_{air}$ = 3.3 kg·s^{-1}, $\dot{m}_{water}$ = 2.4 kg·s^{-1}, tower water inlet temperature = 22°C).

Referring to Figure 5, it can be seen that for most conditions, the majority of cooling tower heat transfer can be attributed to latent heat transfer. Latent heat transfer is observed to decrease significantly with increasing humidity ratio as well as increasing dry and wet bulb temperature. Sensible heat transfer plays a less significant role relative to latent heat transfer, but as ambient air temperatures approaches and exceeds the tower water entry temperature, sensible heat transfer can be observed to be negative, thereby acting to decrease the overall capacity of the tower. Finally, it can be observed at warm, humid conditions that the tower overall capacity tends to zero.

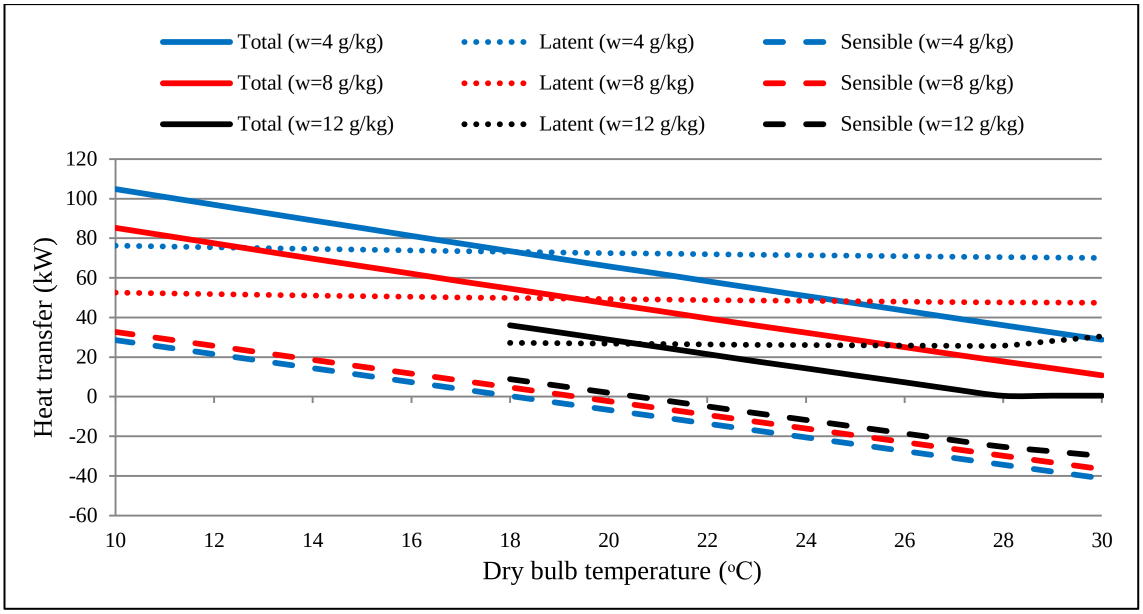

Figure 5: Predicted tower heat transfer as a function of ambient air temperature at different humidity ratios ($\dot{m}_{air}$ = 3.3 kg·s^{-1}, $\dot{m}_{water}$ = 2.4 kg·s^{-1}, tower water inlet temperature = 22°C).

4.2 Constant Ambient Temperature

Using three different dry bulb conditions (10°C, 20°C and 30°C) as a function of relative humidity, Figures 6 and 7 present tower performance data, which could be considered akin to considering three geographical locations, where conditions can vary from relatively dry to relatively humid. Examining Figure 6, it can be observed, in all cases, that the water outlet temperature increases and the approach decreases as the relative humidity increases.

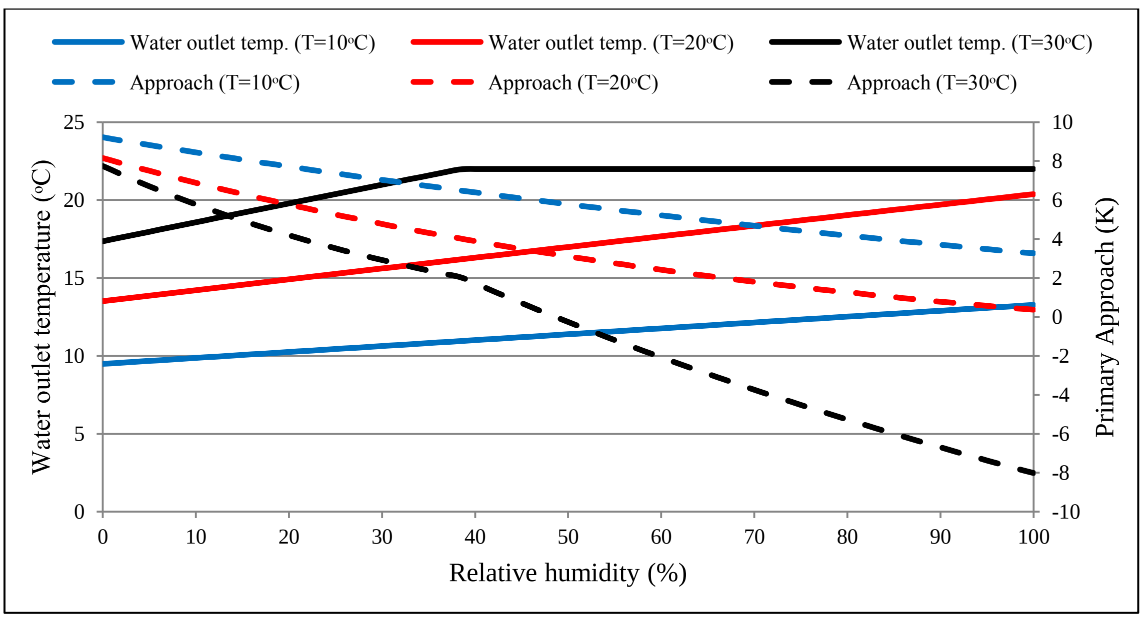

Figure 6: Predicted tower water outlet temperature as a function of relative humidity at different ambient temperatures ($\dot{m}_{air}$ = 3.3 kg·s^{-1}, $\dot{m}_{water}$ = 2.4 kg·s^{-1}, tower water inlet temperature = 22°C).

Considering Figure 7, the overall cooling performance of the tower can be seen to heavily influenced by its latent heat transfer capabilities. Of particular interest is the case for an ambient temperature of 30°C, where for RH values in excess of 40% RH, it can be seen that the tower water exit temperature remains the same as the tower inlet temperature. This is also evident in Figure 7, where total heat transfer is zero for RH values above 40%.

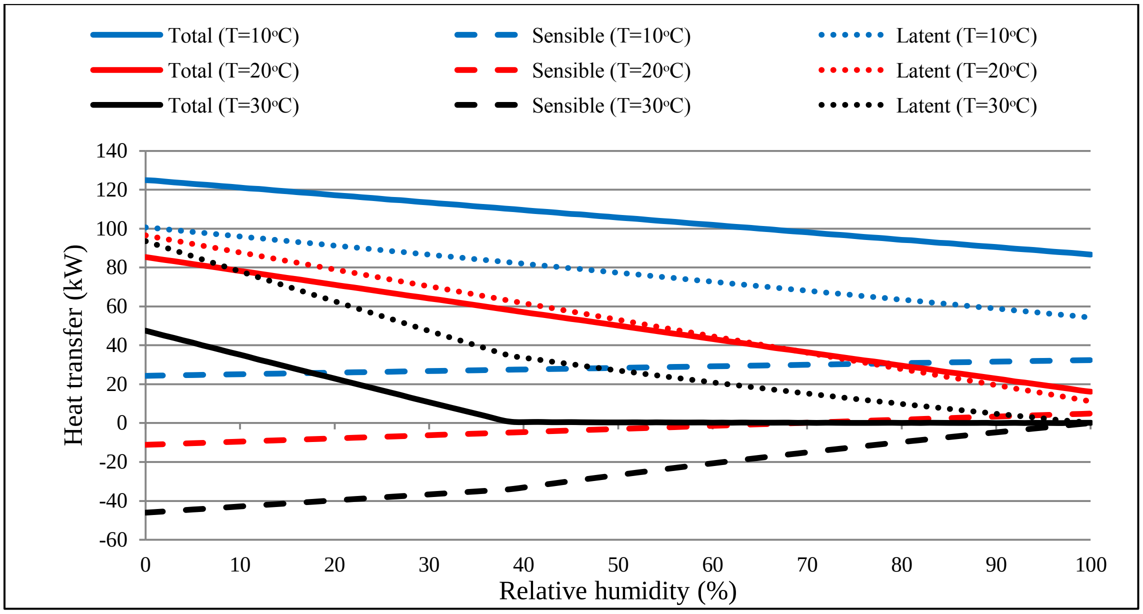

Figure 7: Predicted tower heat transfer as a function of relative humidity at different ambient temperatures ($\dot{m}_{air}$ = 3.3 kg·s^{-1}, $\dot{m}_{water}$ = 2.4 kg·s^{-1}, tower water inlet temperature = 22°C).

4.3 Constant relative humidity

To examine the effect of relative humidity, the performance of the cooling tower at different relative humidity values (low, medium and high) for a range of ambient temperatures from 10°C to 30°C is shown in Figures 8 and 9. It can be seen that the ability to produce chilled water decreases as relative humidity increases. It can also be seen that when the relative humidity is approximately 50%, the tower can only produce chilled water at 20°C for ambient temperatures less than 25°C.

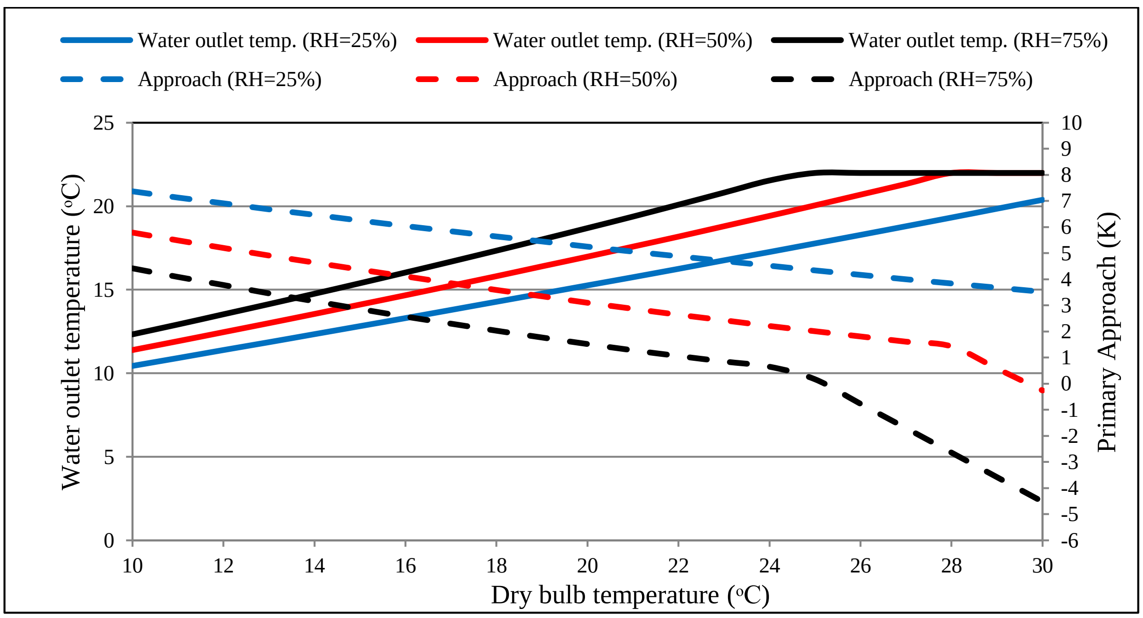

Figure 8: Predicted tower water outlet temperature as a function of ambient air temperature at different relative humidity ($\dot{m}_{air}$ = 3.3 kg·s^{-1}, $\dot{m}_{water}$ = 2.4 kg·s^{-1}, tower water inlet temperature = 22°C).

Examining Figure 9, the potential of cooling tower is depleted at high RH values (RH > 75%), where the ambient temperature is greater than 25°C. Referring to Figure 9, it can be seen that total heat transfer tends to zero for high level of relative humidity, where the ambient temperature is more than 25°C.

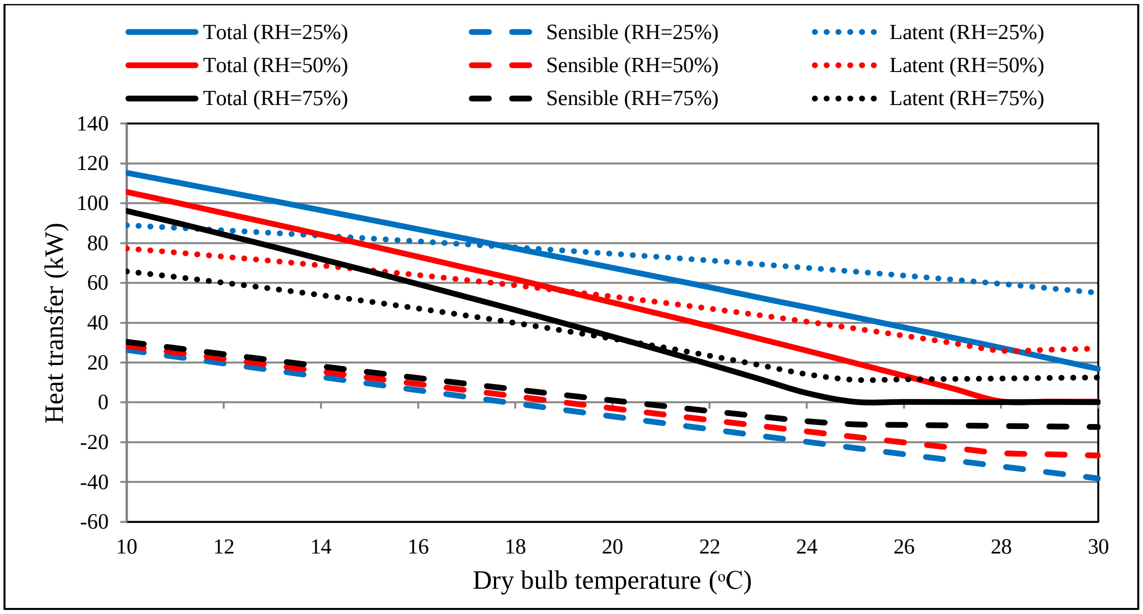

Figure 9: Predicted tower heat transfer as a function of ambient air temperature at different relative humidity ($\dot{m}_{air}$ = 3.3 kg·s^{-1}, $\dot{m}_{water}$ = 2.4 kg·s^{-1}, tower water inlet temperature = 22°C).

6. CONCLUSIONS

Sensitivity studies using a mathematical model of a low approach open evaporative cooling tower for the production of high temperature indirect cooling water (14-16°C) are described. The performance of the tower is examined in terms of a number of its operational parameters including: tower water outlet temperature, tower approach temperature, tower total heat transfer, sensible heat transfer and latent heat transfer. Overall tower heat transfer is primarily by latent cooling, with cooling capabilities constrained by increasing humidity of the air. Likewise, the ability of the cooling tower to produce chilled water is also strongly dictated by the state of the ambient air, being constrained by increasing moisture content of the air.

NOMENCLATURE

A	area (m^2)	a	surface area per unit volume, m^{-1}
c_p	specific heat capacity (J·kg^{-1}K^{-1})	C	heat capacity rate, mC_p(W·K^{-1})
i	specific or latent heat of enthalpy (J·kg$_{a+v}^{-1}$)	i'	enthalpy of dry air (J·kg$_a^{-1}$)
K	mass transfer coefficient (kg·m^{-2}s^{-1})	L	water mass flow rate (kg·s^{-1})
Me	Merkel number	$\dot{m}$	mass flow rate, (kg·s^{-1})
$\dot{Q}$	heat transfer rate (W)	T	temperature (°C)
V	volume of tower (m^3)	ω	humidity ratio (kg$_v$·kg$_a^{-1}$)

<u>Subscripts</u>

a	dry air or ambient conditions	$evap$	evaporated
fg	latent heat of vapourisation	i	inlet
o	outlet	v	vapour
w	water		

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