SIMPLIFIED MODELS FOR DIRECT AND INDIRECT CONTACT COOLING TOWERS AND EVAPORATIVE CONDENSERS

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ABSTRACT

This paper presents a simplified method of analyzing the combined heat and mass transfer phenomena in direct and indirect contact cooling towers and evaporative condensers.
The theoretical basis of the model is Merkel's theory. The cooling towers, evaporative condensers and fluid coolers are considered as members of a classical heat exchangers family working in wet regime. Here, the air side heat and mass transfer processes are governed by the same basic process. This paper shows that a "unified" theoretical treatment may be applied to all three evaporative exchangers. The key difference in the theory for each type relates to these exchangers unique characteristic, the global heat transfer coefficient UA, or its correspondent thermal resistances of the process fluid. Specific correlations for the calculation of UA or the equivalent thermal resistances, considering the influence of the water and air flow rates entering the exchangers, are discussed. An example for each case is shown illustrating the validation of the models with catalogue data.

INTRODUCTION

Direct contact cooling towers, indirect contact cooling towers and evaporative condensers are all considered evaporative cooling equipments, being used in many industrial applications to reject heat to the atmosphere. This is achieved partly by an exchange of latent heat from the water evaporation and partly by a transfer of sensible heat.
The fundamental heat transfer theory of evaporative cooling equipments operation was developed by Merkel (1925), and his method has been adopted as the basis of practically future authors studies. In Merkel's analysis, the enthalpy potential is used as the driving force for air-water exchange and is assumed a similarity between heat and mass convective transfer by stating the Lewis number equals the unity, and that the water loss due to evaporation can be neglected.
Webb (1984) presented a unified theoretical treatment of the equations applicable to cooling towers (DCCT), fluid coolers (ICCT) and evaporative condensers (EC), where the approximate Merkel's equation was employed for the air side heat and mass transfers. Braun (1988) developed another unified method to model the performance of both cooling towers and dehumidifying coils. Effectiveness relations have been developed, taking into account a saturated air-specific heat used for sensible heat exchangers. It was then proposed to generalize Braun's model replacing air enthalpies by wet-bulb temperatures as driving forces in cooling coils (Ding et al, 1991) and in cooling towers (Bourdouxhe et al, 1993 and Lebrun et al, 2002)

Also new mathematical models have been proposed for fluid coolers and in general for evaporating cooling devices (Zalewski et al., 1997 and Halasz, 1998). For evaporative condensers, specific studies have been carried out on sizing and control (Manske et al., 2001 and Hwang et al., 2001)

This paper describes a simplified general model for analyzing the combined heat and mass transfer in evaporative cooling equipments. The three evaporative cooling equipments are considered as classical heat exchangers, where it is just necessary to make the difference at the level of their unique characteristic, the global heat transfer coefficient UA, or its correspondent thermal resistances of the fluids.

UNIFIED THEORETICAL MODEL

The major assumptions used to derive the basic modeling equations may be summarized as follows:

- The humid air is replaced by a fictitious perfect gas which temperature is the wet bulb temperature.
- The interfacial air film is assumed to be saturated with water vapor.
- The supply and exhaust water mass flow rates are assumed to be equal (i.e. the effect of the water lost by evaporation is neglected).
- Lewis number is equal to one.
- The liquid conductance is assumed to be much greater than the gas conductance (i.e., the wet surface temperature is assumed to be equal to the water temperature).

MODEL

The energy balance on the air side gives:

$$\dot{Q} = \dot{M}_a \cdot (h_{a,ex} - h_{a,su}) \qquad (1)$$

In accordance to the assumptions, a humid air fictitious capacity flow rate is assumed, which can be express as:

$$\dot{Q} = \dot{C}_{af} \cdot (t_{wb,ex} - t_{wb,su}) \qquad (2)$$

$$\dot{C}_{af} = \dot{M}_a \cdot c_{p,af} \qquad (3)$$

where it can be observed that the fictitious fluid enters the heat exchanger at the temperature $t_{wb,su}$ and is characterized by a fictitious specific heat, $c_{p,af}$. By combining the above equations together, it results:

$$c_{p,af} = \frac{(h_{a,ex} - h_{a,su})}{(t_{wb,ex} - t_{wb,su})}$$

In accordance to the definition of an equivalent heat exchanger for the simulation of the three evaporative cooling equipments, the heat flow rate can be defined by:

$$\dot{Q} = \varepsilon_{fic} \cdot \dot{C}_{\min} \cdot (t_{r,su} - t_{wb,su}) \tag{4}$$

Where ε_{fic} depends on the type of heat exchanger considered. The minimum capacity flow rate is defined as:

$$\dot{C}_{\min} = \min(\dot{C}_r;\dot{C}_{af})$$

Cooling Tower

In the cooling tower model the cooling fluid (named here as refrigerant) flows in counter current arrangement with the air flow, thus the energy balance on the refrigerant side gives:

$$\dot{Q} = \dot{C}_r \cdot (t_{r,su} - t_{r,ex}) \tag{5}$$

where the refrigerant capacity flow rate is given by

$$\dot{C}_r = \dot{M}_r \cdot c_{p,r} \tag{6}$$

The effectiveness of this heat exchanger will be defined as:

$$\varepsilon_{fic} = \frac{1 - e^{(-NTU_{fic} \cdot (1-\omega))}}{1 - \omega \cdot e^{(-NTU_{fic} \cdot (1-\omega))}} \tag{7}$$

that is defined on the basis of the fictitious perfect gas. The fictitious NTU is considered as

$$NTU_{fic} = \frac{AU_{fic}}{\dot{C}_{\min}} \tag{8}$$

and the capacity flow rate ratio is given by the following relationships:

$$\omega = \frac{\dot{C}_{\min}}{\dot{C}_{\max}}$$

$$\dot{C}_{\max} = \max(\dot{C}_r;\dot{C}_{af})$$

The fictitious cooling tower heat transfer coefficient is related to the dry heat transfer coefficient by the following expression:

$$AU_{fic} = AU_{fic,dry} \cdot \frac{c_{p,af}}{c_{p,a}} \tag{9}$$

In the case of Direct Contact Cooling Tower:
In the DCCT the refrigerant considered is water which passes in counter flow arrangement with the air flow (as shown conceptually in figure 1). The influence of both water and air flow rates on the global heat transfer coefficient is:

$$AU_{fic,dry} = AU_{fic,dry,n} \cdot \left[\frac{\dot{M}_w}{\dot{M}_{w,n}}\right]^m \cdot \left[\frac{\dot{M}_a}{\dot{M}_{a,n}}\right]^n \tag{10}$$

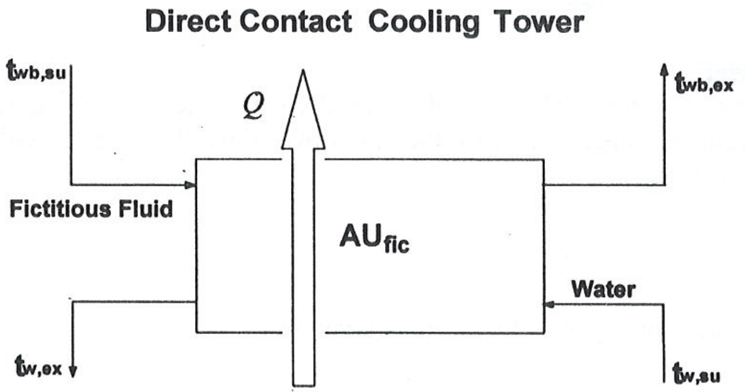

Figure 1: Conceptual scheme of the DCCT equivalent heat exchanger.

In the case of Indirect Contact Cooling Tower:
In the ICCT, the refrigerant temperature varies while it flows through the heat exchanger and the water mass flow rate is constant. A conceptual schema for the ICCT is shown in figure 2.
The global heat transfer coefficient is taken as the sum of two resistances, the refrigerant side resistance and the air side resistance.

$$AU_{fic} = \frac{1}{R_{fic}} \tag{11}$$

$$R_{fic} = R_{af} + R_r \tag{12}$$

being the fictitious air resistance given by

$$R_{af} = R_a \cdot \frac{c_{p,a}}{c_{p,af}} \tag{13}$$

and the air side resistance in this case will just consider variations due to the air flow rate, because the water flow is supposed to be constant, then:

$$R_a = R_{a,n} \cdot \left[\frac{\dot{M}_a}{\dot{M}_{a,n}}\right]^n \tag{14}$$

The refrigerant side resistance is calculate through

$$R_r = R_{r,n} \cdot \left[\frac{\dot{M}_r}{\dot{M}_{r,n}} \right]^m \tag{15}$$

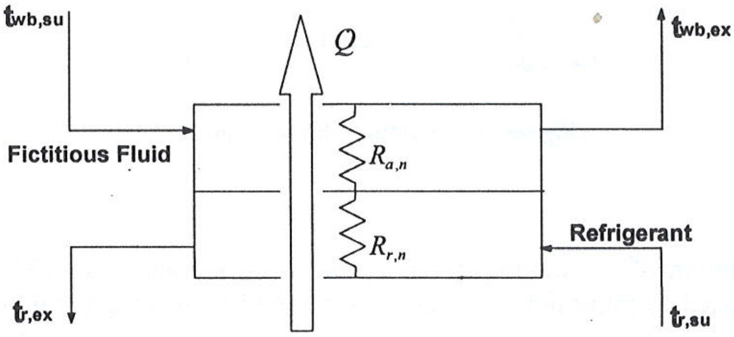

Figure 2: Conceptual scheme of the ICCT equivalent heat exchanger.

Evaporative Condenser

The EC model also is based on a classical heat exchanger model in which it is assumed a heat exchanger corresponding to the biphasic state of the refrigerant in the condensing process (see figure 3). The refrigerant and the water temperatures are considered to be constant (the circulating water mass flow rate is supposed to be high enough). Thus, the energy balance on the refrigerant side and the effectiveness are defined in a different way in comparison with the cooling tower models.

Heat flows on the refrigerant side are given by:

$$\dot{Q} = M_r \cdot (h_{r,su} - h_{r,ex}) \tag{16}$$

$$\dot{Q} = AU_r \cdot (t_{cd} - t_w) \tag{17}$$

The heat flow rate through the condenser can also be defined between the water and the air, in this case in equation (4) we have: $t_{r,su} = t_w$. As we considered that the water temperature remains essentially constant, the effectiveness for such situation is given by

$$\varepsilon_{fic} = 1 - e^{(-NTU_{af})} \tag{18}$$

and the fictitious NTU is

$$NTU_{af} = \frac{1}{R_{af} \cdot \dot{C}_{af}} \tag{19}$$

On the air side, the resistances will be considered as shown in equations (13) and (14). In the refrigerant side the resistance is given by

$$R_r = R_{r,n} \tag{20}$$

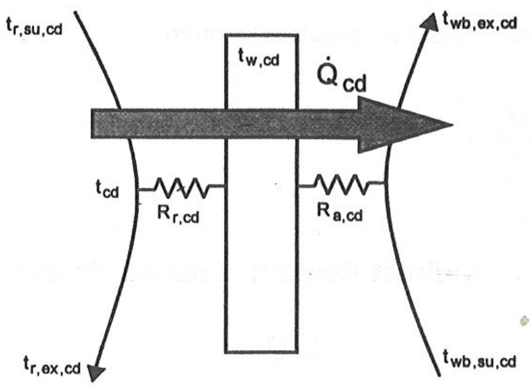

Figure 3: Conceptual scheme for the EC.

RESULTS

For the validation of the model, it was required a parameter identification process, which is focused in minimizing the error committed between the calculated cooling capacity, $\dot{Q}$, and the values taken from manufacturer catalogues. Wide ranges of operating conditions were considered, which ensures good simulation results of the evaporative cooling equipments.
In the attempt to visualize the equipments cooling capacity at the same operational condition, the following values will be defined:

$$t_{w,su} = 35°C$$
$$t_{wb,su} = 18°C$$
$$Range = 10°C$$

The cooling capacity for the DCCT and ICCT are 417 kW and 594 kW, respectively. If for the EC we considered $t_{wb,su} = 18°C$, $t_{cd} = 35°C$ and $t_{ev} = -5°C$ (superheating of 5 degrees) the cooling capacity will be 857 kW.

a) Direct Contact Cooling Tower

The model of the DCCT requires the identification of 3 parameters, the $AU_{dry,n}$, and the exponents m and n. The information supplied by the catalogues correspond to supply wet-bulb temperature and the water temperatures, as well as its corresponding mass flow rate, by combining the latest is possible to obtain the tower cooling capacity. This catalogue cooling capacity is then compared with that one calculated by the model for the same conditions.
The identified parameters for the DCCT are shown hereafter:

$$AU_{dry,n} = 16.43$$
$$m = -0.025$$
$$n = 0.6$$

The comparison between the catalogue cooling capacity and the calculated values is shown in figure 4 in means of the percentage of error committed. In the simulation, a range of wet-bulb temperatures from 15 to 27°C was considered, as well as an extended range for the water flow rate. No variations of the air flow rate was considered.

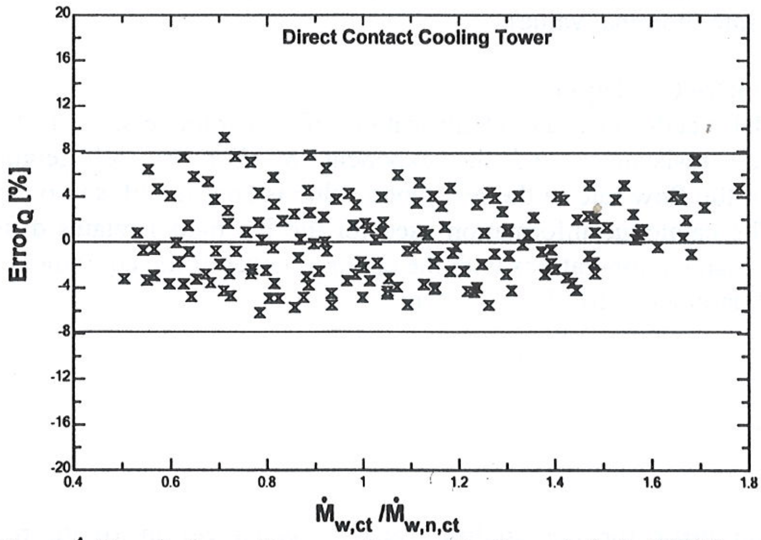

Figure 4: Error in the cooling capacity vs. water flow rate in the DCCT model.

b) Indirect Contact Cooling Tower

The ICCT model works with 4 parameters, the resistances on the air and refrigerant side, and the exponents m and n. The same analysis done for the DCCT was considered for the ICCT.

The parameter identification gave the following values:

$$R_{a,n} = 0.025$$
$$R_{r,n} = 0.00529$$
$$m = -0.8$$
$$n = -0.6$$

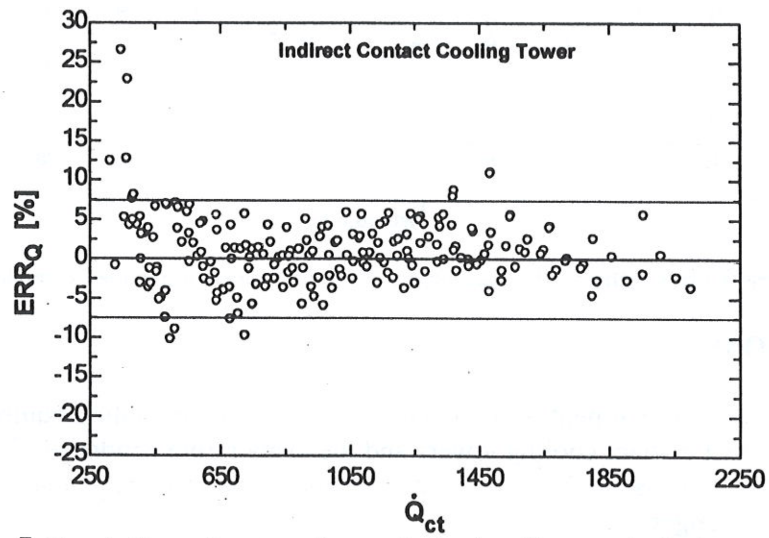

Figure 5: Error in the cooling capacity vs. calculated cooling capacity in the ICCT model.

In figure 5 is shown the error committed when comparing the catalogue cooling capacity and the calculated values. In the simulation, wet-bulb temperatures from 16 to 26°C were considered. Variations of the refrigerant flow rate were analyzed at a fixed air flow rate (nominal value).

c) Evaporative Condenser

The EC model needs also the identification of 3 parameters, both the air and refrigerant side resistances and the exponent n. For the refrigerant side no dependency on the flow rate in the resistance value is considered, so no exponent m is required. The catalogue information used for the EC model relates different wet-bulb and condensing temperatures with the cooling power of the equipment.
The identified parameters for the EC where:

$$R_{a,n} = 0.0191$$
$$R_{r,n} = 0.00545$$
$$n = -0.5$$

The simulation error for the cooling capacity is shown in figure 6. Wet-bulb temperatures from 10 to 28°C where considered, as well as condensing temperatures starting from 29 up to 45°C.

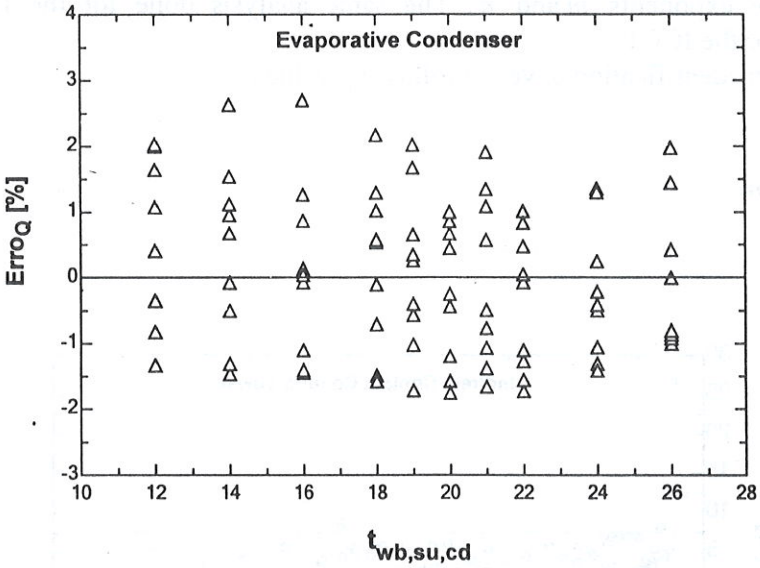

Figure 6: Error in the cooling capacity vs. wet-bulb temperature in the EC model.

CONCLUSIONS

A unified theoretical treatment is applied to three evaporative cooling equipments, the direct and indirect contact cooling towers and the evaporative condenser. The method deals with the analysis of the combined heat and mass transfer phenomena in the evaporative exchangers.
A general and simple model have been presented which is represented by a fictitious heat exchanger with wet-bulb temperature as driving force on the air side. This approach is validated with manufacturers catalogue data information, for which is

required a "manual" parameter identification process. The results of the model compare well with the manufacturers information, when analyzing the cooling capacity of the three evaporative exchangers.
Simple relationships are propose to correlate the UA values, or their equivalent thermal resistances, with the air and refrigerant mass flow rates.

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NOMENCLATURE

A = exchange surface area, m^2
AU = heat transfer coefficient, W/K
$\dot{C}$ = capacity flow rate, W/K
c_p = specific heat, J/kg K
ERR = error on the heat flow rate, %
h = enthalpy, J/kg K
m = refrigerant side mass flow rate ratio exponent
$\dot{M}$ = mass flow rate, kg/s
n = air side mass flow rate ratio exponent
NTU = number of transfer units
p = pressure, Pa
$\dot{Q}$ = heat flow rate, W
t = temperature, K
v = specific volume, m^3/kg

Greek symbols
ε = effectiveness
ρ = density, kg/m^3
ω = capacity flow rate ratio

Subscripts
a = air
af = fictitious air
cd = condenser
dry = dry regime
ex = exhaust
fic = fictitious
min = minimum
max = maximum
n = nominal conditions
r = refrigerant
su = supply
w = water
wb = wet-bulb

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