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INFLUENCE OF AIR THERMOHYGROMETRIC PROPERTIES ON MECHANICAL DRAFT EVAPORATIVE TOWERS: A CALCULATION METHOD TO PREDICT EFFECTS IN POWER PLANTS AND REFRIGERATING ABSORPTION MACHINES

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ABSTRACT

The paper describes a simulation model for the evaluation of heat exchanges occurring to a flow of water falling down in counterflow with humid air, during its path through a mechanical draft cooling tower. The contemporary heat transfer by evaporation and forced convection are separately taken into account, assuming simplified hypotheses such as water flow divided into spherical drops, constant relative speed between air and water and constant thermodynamic conditions of water and air through the tower. The calculation method has been calibrated and validated through two existing evaporative towers and the comparison with the well-known Merkel and Poppe models showed only slight differences on thermodynamic parameters involved in the process. The simulation predicts water outlet conditions starting from the air inlet temperature and relative humidity, once the power to be drained is fixed. The knowledge of outlet air thermohygrometric conditions allows the definition of water outlet temperature minimum value. Since the outlet water temperature level affects the performances of plants connected with the evaporative towers, it is interesting to vary the air external conditions, investigating the repercussions on plants'characteristics. While the influence on power plants remains circumscribed to a light variation on the produced electric power, refrigerating absorption machines result more sensitive, showing considerable fluctuations of chilling power and Coefficient Of Performance (COP).

1. INTRODUCTION

The majority of industrial processes involving heat exchange needs devices to discharge the residual heat in the surrounding environment. An efficient and mature way of accomplishing this task is constituted by evaporative towers; their applications embrace power plants, industrial applications where heat removal is required and refrigerating devices such as vapour compression and absorption machines.
The success of evaporative towers depends on the easy concept of functioning: a flux of humid air is pushed against the water to be cooled; the latter loses heat by convection and, mainly, by evaporation: the air exits the tower more humid and the water temperature is decreased.
The design of evaporative towers starts from the knowledge of the heat amount to be removed and from meteorological conditions of the site; generally they are oversized at the aim of being sure that, for each external condition, the requirements are satisfied.
Nevertheless, it could happen that extreme weather conditions (high temperature and relative humidity) sometimes bring to full exploitation of a tower, causing the undesired effect of raising the temperature of the carrier fluid that enters the power plant condenser or the absorber-condenser couple of an absorption machine.
A deep understanding of thermal exchanges permits to manage water needs, defining at the same time how atmospheric conditions influence the thermodynamic parameters of the plants served by towers [1]. An overview of the most used methods for wet cooling towers analysis has been conducted, considering that mathematical models constitute a valid instrument to investigate these phenomena without arranging huge experimental campaigns by tower manufacturers or by users.
A new calculation method is then proposed at the aim of fulfilling the specific analysis of wet cooling towers serving power plants and refrigerating absorption machines.

2. WET COOLING TOWERS: SOME METHODS OF ANALYSIS

The investigation on heat transfer phenomena occurring in evaporative cooling started at the beginning of last century with the definition of the first simplified models and governing equations. The theory developed by Merkel [2] started from a series of assumptions that helped creating an easy-to-use calculation method:

- the air leaving the tower is saturated (100% relative humidity);
- the water flow rate reduction due to evaporation throughout the tower is negligible;
- the Lewis factor Le_{fa} that connects heat and mass transfer is considered constant and equal to 1. The Le_{fa} depends on

the heat transfer coefficient h and the mass transfer coefficient m by the following relation:

$$Le_{fa} = \frac{h}{mC_{p,a}} \tag{1}$$

where $C_{p,a}$ represents the air specific heat at constant pressure.

An evolution of this approach is represented by the Poppe method [3] that abandons Merkel hypotheses of outlet saturated air, negligible mass flow rate lost by evaporation, and invariability of the Lewis factor. The latter could be expressed [4] as a function of air specific humidity X and air specific humidity in saturated conditions at the local water temperature $X_{sat,w}$:

$$Le_{fa} = 0.865^{\frac{2}{3}}\left(\frac{X_{sat,w}+0.622}{X+0.622}-1\right)\Big/\ln\left(\frac{X_{sat,w}+0.622}{X+0.622}\right) \tag{2}$$

The system of differential equations deriving from the energy and mass balance of the Poppe theory could be solved by the Runge-Kutta method [5] that permits the evaluation of all the variables involved, both from the water and the air side.

The Merkel method strength consists of its simple hand calculation while it shows weaknesses at low air ambient temperatures simulations and in the prediction of the plumes exiting the cooling tower. The Poppe theory results more accurate in the entire range of ambient air thermohygrometric properties as well as in the definition of the air outlet conditions, even if it refers to an indirect method of evaluating the relation between heat and mass transfer processes (Eq. 2). In the next paragraph a new calculation method is proposed: the two mechanisms of heat exchange are evaluated separately, taking into account that they occur with the same values of the thermodynamic variables.

3. THE PROPOSED CALCULATION METHOD

The proposed calculation method hypothesizes that the water flow through the mechanical draft cooling tower is constituted by spherical drops, subjected to a double way of heat exchange in their whole surface: forced convection and surface evaporation. Drops are assumed to maintain the spherical shape and the diameter length (the amount of water evaporated for each drop is considered negligible respect to its volume); actually, the water coming out of the nozzles is not necessarily divided into spherical drops and, during the passages through the water packing, is continuously split into smaller parts and slowed by impacts. Besides, the entire surface is not completely in contact with the air because of the presence of the packing; therefore the calculation method does not intend to represent the actual evolution of the phenomenon but it describes an equivalent flow of water that is supposed to exchange the same heat by convection and evaporation transferred by the real flow.

Equations describing the heat transfer of water drops by forced convection and evaporation are firstly treated separately, than the model implementation couples the two heat exchange modalities and links them to the tower geometrical features.

3.1 CONVECTION HEAT TRANSFER

The heat transfer between a water sphere (diameter D) immersed in air, is represented by the following relation [6]:

$$\frac{hD}{k_f} = 2.0 + 0.60\left(\frac{Dv_r\rho_f}{\mu_f}\right)^{1/2}\left(\frac{C_p\mu}{k}\right)_f^{1/3} \tag{3}$$

where the convection heat transfer coefficient h is relative to the entire sphere surface and the value of the relative velocity v_r between air and droplet is the sum of the air velocity v_∞ imposed by the fan and the drop velocity v_g due to the gravitational force, slowed by the packing impacts (fig. 1).

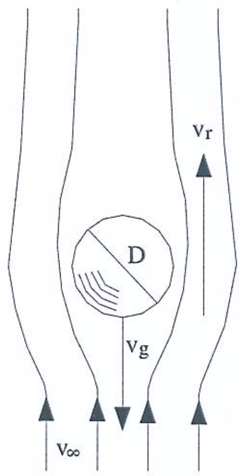

Figure 1: velocity composition between water drop and air.

The thermophysical properties such as k, ρ, μ and C_p have to be evaluated at the film temperature: it is considered as the average between the drop mean temperature (hypothesizing the drop temperature homogenous as it moves along the evaporative tower) and the air mean temperature, during their paths through the tower.

Once the heat transfer coefficient is obtained, the heat exchanged between one drop and surrounding air (per time unit) can be written considering the entire drop surface A:

$$Q_{conv} = hA(T_{m,w} - T_{m,a}) \tag{4}$$

3.2 EVAPORATION HEAT TRANSFER

The molar water evaporation rate from the drop surface towards the surrounding humid air, in case of slow mass transfer, can be expressed in terms of the mass transfer coefficient m_w [7]:

$$W_w = m_w A(x_{w0} - x_{w\infty}) + x_{w0}(W_w + W_a) \tag{5}$$

where x_{w0} is the water concentration in proximity of the drop surface and $x_{w\infty}$ is the water concentration in the undisturbed air.

The air solubility in water could be neglected and, considering the drop a sphere with diameter D, Eq.(5) becomes:

$$W_w = m_w \pi D^2 \frac{x_{w0} - x_{w\infty}}{1 - x_{w\infty}} \quad (6)$$

The mass transfer coefficient m_w is derived from the relation of the convection heat transfer, adapted for the mass transfer [7]:

$$\frac{m_w D}{c_{a,f} \Xi_{awf}} = 2.0 + 0.60 \left(\frac{D v_r \rho_f}{\mu_f} \right)^{1/2} \left(\frac{\mu}{\rho \Xi_{awf}} \right)_f^{1/3} \quad (7)$$

As already stated for the convection heat transfer, the fluid properties have to be calculated at the film conditions.

The diffusivity of the gaseous mix between air and water could be expressed in terms of their critical pressures, critical temperatures and molecular weights, at the pressure and temperature where the phenomenon occurs [8]:

$$\frac{p \Xi_{aw}}{(p_{cw} p_{ca})^{1/3} (T_{cw} T_{ca})^{5/12} \left(M_w^{-1} + M_a^{-1}\right)^{1/2}} = 3.64 \cdot 10^{-4} \left(\frac{T_f}{\sqrt{T_{cw} T_{ca}}} \right)^{2.334} \quad (8)$$

From Eq.(6) is therefore obtained the evaporation velocity that brings to the evaporative heat exchange rate for each drop, simply multiplying by the latent heat; the latter (in MJ/kg) is a function of the drop temperature (in Celsius degrees), according to the following equation [9] for water:

$$r_w = 2501 - \left(2361 \cdot 10^{-3}\right) T_{m,w} \quad (9)$$

Finally, the heat lost by the drop through the evaporation process could be evaluated:

$$Q_{ev} = W_w M_w r_w \quad (10)$$

3.3 CALCULATION METHOD STRUCTURE

Some of the variables contained in previous equations are fixed once the fluids involved are chosen (air and water): p_{cw}, p_{ca}, T_{cw}, T_{ca}, M_w, M_a. All the other parameters depend from inlet and outlet conditions of water and air.

The temperature and relative humidity of external air (inlet to the tower) are easily measurable, while their values at the tower outlet are in most cases unknown.

The plants served by the evaporative tower ask for steady power to drain, normally with fixed water flow rate and temperature differences; the only parameter that varies with external air conditions is represented by the outlet (and inlet) temperature of water.

The purpose of the calculation method consists of determining a relation between environmental air conditions and the lower level of the inlet water temperature sent to users: this value, in fact, affects the performances of the plants served by the tower.

In thermoelectric plants, the outlet water from the tower corresponds approximately to the condenser inlet; that means that its variation modifies the condensing temperature (and pressure) of the vapour cycle, with direct consequences on the cycle efficiency.

In absorption refrigerating machines, the cooling inlet to the condenser and absorber has a direct influence on the machine performance, once the other parameters are fixed. In particular, it is easy to demonstrate that the Coefficient Of Performance decreases when the cooling water inlet temperature increases, keeping constant the generator inlet temperature [10].

The calculation procedure of the calculation method starts from the evaluation of the environmental air specific humidity X, as a function of temperature and relative humidity Φ:

$$X = \frac{622\Phi}{p - \Phi p_{sat}(T)} p_{sat}(T) \quad (11)$$

where p is the atmospheric pressure; the saturation vapour pressure $p_{sat} = p_{sat}(T)$ may be expressed as follow [11]:

$$p_{sat} = 6.11 \cdot 10^{\left(\frac{7.5T}{237.7+T}\right)} \quad (12)$$

where p_{sat} is expressed in hPa and T in °C.

Similarly, it is possible to evaluate the enthalpy of the inlet air:

$$i_{a,i} = C_{p,a} T_{a,i} + X[r_w(T_{a,i}) + C_{p,w,sh} T_{a,i}] \quad (13)$$

The amount of power to be drained and the air flow rate allow the evaluation of the air enthalpy jump and the value of outlet air enthalpy.

Since the outlet air temperature is not known, an iterative method has been developed hypothesizing a starting value $T_{a,0}$, hence, it is possible to define the specific humidity of outlet air by Eq. (13) rearranged:

$$X_{a,o} = \frac{i_{a,o} \cdot C_{p,a} \cdot T_{a,o}}{r_w\left(T_{a,o}\right) + C_{p,w,sh} T_{a,o}} \quad (14)$$

The corresponding value of relative humidity derives from Eq.(11) solved by relative humidity Φ:

$$\Phi_{a,o} = X_{a,o} \frac{p}{(622 + X) p_{sat}(T_{a,o})} \quad (15)$$

It is now possible to obtain the mean value of the air $T_{a,m}$ during its path through the tower as the mean value between the inlet and outlet.

From the water point of view, it is necessary to hypothesize a value for the inlet so that the outlet is immediately known, allowing the calculation of the water temperature mean value $T_{w,m}$ inside the tower.

It is supposed that the average between the mean values of air and water temperatures represents the film temperature:

$$T_f = \frac{T_{a,m} + T_{w,m}}{2} \quad (16)$$

The drops velocity and their diameter was fixed in the validation process, therefore, it is possible to extract the convection heat transfer coefficient by Eq.(3) and the heat exchanged between one drop and the surrounding air by Eq.(4).

Equation (8) gives the diffusivity of the gaseous mix composed by water and air; consequently it is possible to determine the mass transfer coefficient m from Eq.(7).

Since the concentration $x_{w,\infty}$ of water in the undisturbed air (far from the drop) could be considered as a function of the mean temperature and mean relative humidity, the only unknown term to solve Eq.(6) for the evaporation velocity is the concentration $x_{w,0}$ of water near the drop surface. This term could be expressed as follow:

$$x_{w,0} = x_{w,\infty} + \xi [x_{sat}(T_d) - x_{w,\infty}] \tag{17}$$

where $x_{sat}(T_d)$ is the water vapour tension at the drop temperature and the "resistance factor" ξ (between 0 and 1) represents the resistance to mass transfer at the water-air interface (fig. 2) and it is derived from the validation process.

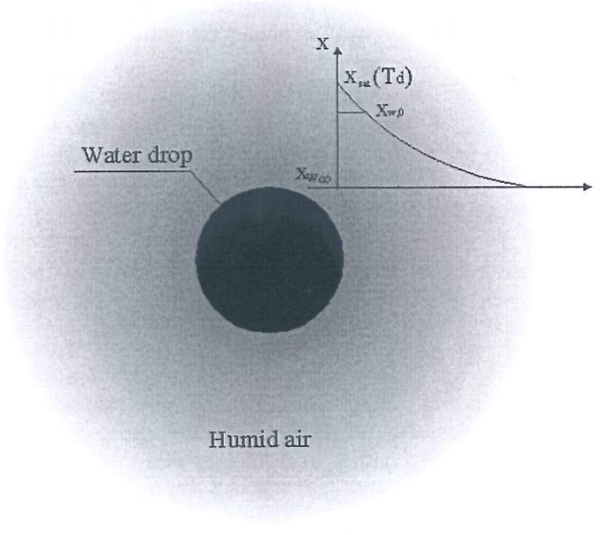

Figure 2: model for the water concentration distribution outside the drop surface.

The outlet air temperature and inlet water temperature constitute the two starting values for the iteration.
The global thermal exchange is obtained by summing the two contributions to the heat exchanged between the drop and the air; besides, the number of drops entering the tower each second is calculated, dividing the total water flow by the single drop volume, while the time the drop stays in the tower derives from the division of the mean drop velocity by the tower height. This means that, multiplying the permanence time by the drops entering the tower per second, the number of drops present at the same time inside the tower is evaluated. Finally, multiplying the single drop global thermal exchange by the number of drops simultaneously present inside the tower, the new value for the power drained could be found and compared with the nominal one; if the difference is not acceptable, the inlet water temperature is modified until the error results lower than the fixed value (for example 1%).

Since the heat exchanged by evaporation for each drop is known, dividing by the latent evaporation heat, the mass evaporation rate is obtained. Dividing again this value by the air flow rate, the increase of the air flow specific humidity is evaluated and, adding the initial specific humidity value, the outlet air specific humidity is found, together with the corresponding value of the outlet air temperature: the latter is compared with the initial value of the iteration.

3.4 CALCULATION METHOD APPROXIMATIONS

The calculation method fits to the various counterflow mechanical draft towers building techniques. With few modifications it could be adapted to cocurrent flow mechanical draft cooling towers, while the passage to natural draft results more complex because of the need of combining the energy equations with the dynamic phenomena linked to the natural convection.

Focusing on calculation method approximations, it is assumed that heat exchange modalities have been evaluated considering an average of the thermophysical properties of water and air, instead of considering their instantaneous value and following them through their evolution inside the tower. Furthermore, a lack in precision is due to the spherical, constant and one-diameter assumption for the water drops: the presence of the filling has actually the task of increasing the exchange surface breaking the drops in smaller ones, so that in the real situation a distribution of diameters would have to be taken into account and the drops should be considered of different shapes. The packing has also the function of slowing the water downward falling: the speed of each drop is characterised by continuous accelerations and decelerations under the gravity force and impacts with the filling material; the top velocity is not very high, so the error of considering a mean constant value all over the tower is not far from the real behaviour, having said that this speed could not be higher than the value derived from the free falling equations. The same observations remain valid for the air velocity, assumed constant in each tower section and inside the section itself; while it seems nearly true that the air speed remains more or less unchanged in different sections because of the continuity equation, some variations occur through the section area, being higher in the centre and decreasing towards the tower edges; nevertheless, the average speed assumption does not affect considerably the final result, because of the counterbalance between higher velocities in the inner part and lower velocities in the external one.

4. CALIBRATION AND COMPARISON WITH MERKEL AND POPPE MODELS

The calculation method proposed has been validated through an experimental campaign on mechanical draft cooling towers. For this purpose, a 22 680 kW tower that cools, with nine other similar towers, the condenser of a coal-fired power station has been investigated [12].

The manufacturer indicates the nominal data in terms of maximum power drainable with fixed inlet and outlet water temperatures, water flow rate, air flow rate and inlet air conditions (temperature equal to 23° C and 60% of relative humidity). Transferring these data to the calculation method, it has been possible to define the mean diameter of the drops (0.005 m) and their mean falling velocity (0.44 m/s, lower than the value obtained by free falling equations, as expected); the "resistance factor" ξ is obtained considering that, in nominal conditions, the performance of the tower is at its best, that is, the outlet air is hypothesized to be in saturation conditions.
Once the calibration is executed, a comparison with the Merkel and Poppe theories was conducted: the psychrometric chart of fig. 3 shows the path of air inside the above mentioned cooling tower for the three models, hypothesizing

a temperature of 15°C and a relative humidity of 60% for the inlet air.

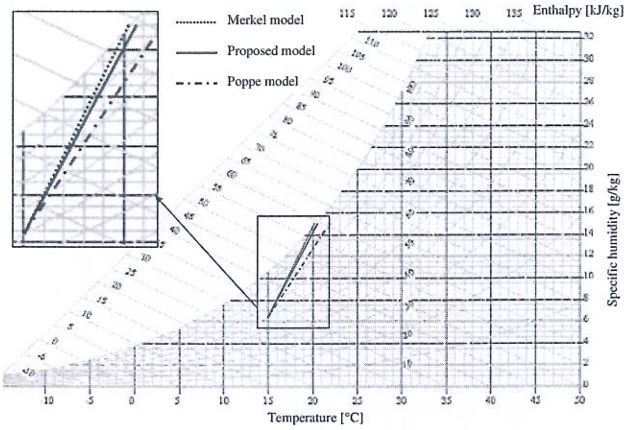

Figure 3: path of humid air in the cooling tower on the psychrometric chart for the three models analysed.

The diagram shows that the air outlet conditions foreseen by the three models present slight differences: the Merkel model considers saturated conditions as an hypothesis, the Poppe model moves the final point towards higher temperatures and lower relative humidity; the proposed calculation method stands between the others.

Focusing on the comparison with the more accurate model (Poppe), it emerges that the use of the Lewis factor to link the heat and mass transfer instead of calculating separately the convection and the evaporation brings to overestimate the contribution of sensible heat exchanges. As a consequence, air exits warmer and water leaves the tower at lower temperatures. This consideration is confirmed by the graph in fig. 4 where the minimum water temperature at the tower outlet is sketched against the inlet air temperature, at three fixed value of relative humidity (40%, 60% and 80%). The curves are presented in conjunction with the data provided by the cooling tower manufacturer [13].

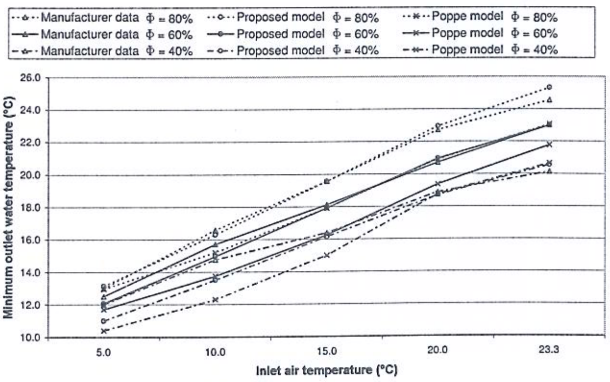

Figure 4: minimum outlet water temperature versus external air conditions: comparison among the proposed calculation method, the Poppe model and the manufacturer data.

The values do not differ significantly, except for a light underestimation of the outlet water temperature for the Poppe model; therefore, the calculation model here developed seems more adherent to the manufacturer results, and it has been selected to evaluate the effect of inlet air conditions on the power plant and absorption machine performance through the temperature of the cooling water. The choice has been dictated by a compromise between easiness of calculation and results accuracy.

5. APPLICATIONS IN ABSORPTION MACHINES AND POWER PLANTS

The calculation method becomes reliable after the calibration for each specific evaporative tower, a process that imposes to take into account the geometry, the power to be drained and the flow rate of water to be cooled.

Starting from absorption machines, simulations have been conducted considering a plant equipped with a 48 kW wet cooling tower. Once the relations sketched in fig. 4 are known and once the curves of chilling power and Coefficient Of Performance (COP) against the cooling water inlet temperature are calculated, it is possible to evaluate the performance of the absorption machine (chilling power and Coefficient of Performance), varying only the inlet air temperature (fig. 5).

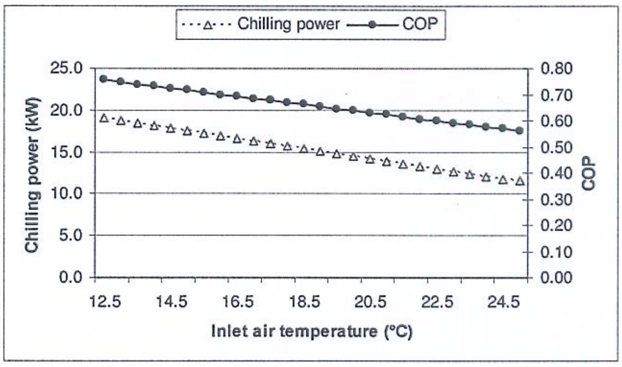

Figure 5: absorption machine chilling power and Coefficient Of Performance versus tower inlet air temperature.

In the tower serving the power plant, considering a fixed temperature jump and a fixed power to be drained by the condenser, the variation of the inlet air temperature reflects on a variation of condenser inlet water temperature that, on its turn, generates the variation of the saturation temperature in the condenser and the energy produced in the power cycle (fig. 6), supposing constant the inlet turbine vapour temperature.

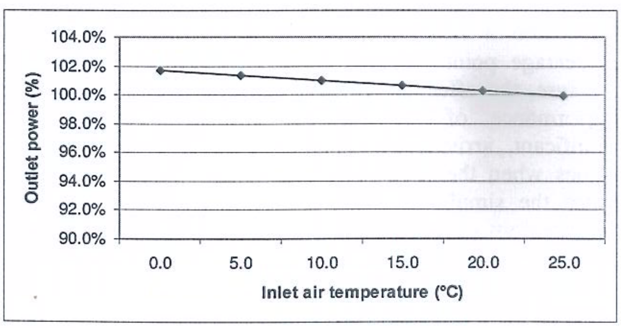

Figure 6: energy outlet versus tower inlet air temperature (ratio by nominal conditions) in the power plant.

As stated before, since the calculation method gives the possibility to separate the contribution of the evaporation from the sensible heat exchange, it could be useful to evaluate the water evaporated, varying inlet air conditions (fig. 7), at the aim of predicting the amount of the water to be reintegrated.

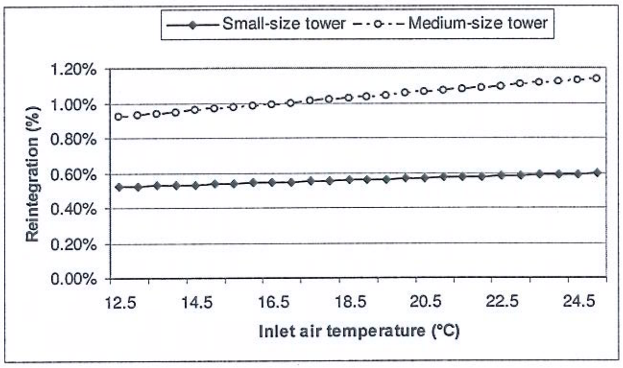

Figure 7: water lost by evaporation (as a percentage of the total flow) versus inlet air temperature for both towers.

It is just the case to remind that to know the global water needed to run the tower properly, two other contributions have to be taken into account: the water pushed away by the air flow (about 0.1 % of the total water flow rate) and the water necessary for resetting the salts concentration.

6. CONCLUSIONS

A calculation method for predicting the behaviour of forced draft wet cooling towers has been developed, with a new method that separately analyses the two mechanisms of heat exchange (convection and evaporation).
The comparison with different size evaporative towers allowed the calibration of the calculation method and validated the procedure, showing good agreement with manufacturer data. Besides, the calculation method output resulted close with the Merkel method of analysis and also with the more accurate Poppe theory, showing minor differences in the outlet air temperature.
The complete knowledge of water and air parameters permits the analysis of the connection between external air conditions and performance of the power plant and the refrigerating absorption machine, served respectively by a medium-size and a small-size cooling tower, when they work in nominal conditions.
While the effects on thermoelectric plants is limited to a short fluctuation of the power produced on the vapour cycle (a few percentage points) caused by the change on condensation pressure, the effect on chilling capacity and Coefficient Of Performance of the absorption machine results more significant, arriving to cut almost an half of the nominal values when the temperature in Celsius degrees is halved. Since the simulation code offers the possibility to divide contributions of each single form of heat exchange, the trend of water evaporated (to be reintegrated) with external air conditions has been defined, finding a linear growth in both towers.
The calculation method, regardless the approximations introduced, reveals easy to manage and flexible to different towers'geometries, needing only few data for its calibration. The encouraging results indicate the possibility of extending the modelling to natural draft cooling towers.

7. LIST OF SYMBOLS

A	drop surface [m^2]
C	specific heat [J/kg/K]
c	molar concentration
COP	Coefficient Of Performance
D	diameter [m]
h	convection heat transfer coefficient [W/m^2/K]
i	enthalpy [J/kg]
k	thermal conductivity [W/m/K]
Le_{fa}	Lewis factor [-]
M	molecular weight
m	mass transfer coefficient [kg/m^2/s]
p	pressure [Pa]
Q	heat per time unit [Watt]
r	latent heat [J/kg]
T	temperature [K]
v	velocity [m/s]
W	molar flow rate [mol/s]
X	specific humidity
x	molar fraction

Greek symbols

Φ	relative humidity [%]
μ	viscosity [kg/m/s]
ρ	density [kg/m^3]
ξ	resistance factor
Ξ	diffusivity [m^2/s]

Subscripts

a	air
c	critical
d	drop
conv	convection
ev	evaporation
f	conditions of the film
g	gravitational
i	inlet
m	mean
o	outlet
p	constant pressure
r	relative
sat	saturation
sh	superheated
w	water
0	conditions near the drop surface
∞	conditions of undisturbed air

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