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Thermal behaviour of closed wet cooling towers for use with chilled ceilings

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Abstract

A new closed wet cooling tower, adapted for use with chilled ceilings in buildings, was tested. Experimental correlations were obtained for mass and heat transfer coefficients. Existing thermal models for this type of cooling tower were found to predict well thermal performance, if the above correlations are used.

Keywords: Cooling tower; Chilled ceiling; Experimental testing; Thermal models

1. Introduction

There is an increasing demand for cooling in buildings. Better insulated buildings and the increased use of working equipment (computers, etc.) led to higher cooling requirements, particularly in office and commercial buildings. The need for reducing energy consumption and CO_2 emissions, together with the need for using environmental-friendly refrigerants, justify a strong demand for CFC free, efficient and cheap cooling systems.

Chilled ceilings are a relatively new approach to cooling, with about 100,000 m^2 installed in central European countries in the last five years. Chilled ceilings offer several advantages over conventional alternatives. The use of water instead of air reduces energy requirements for energy transportation. It also allows a reduction of ventilation rate to a minimum level, for hygiene purposes. Heat transfer from indoor space to chilled ceilings is made by combined

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convection and radiation. Radiative heat transfer allows chilled ceilings to remove considerable heat loads at a relatively small temperature difference between room air and ceiling. This makes it possible to run the system with ceiling supply temperatures of about 18–20°C. Heat transfer rates between 25 and 75 W/m^2 are possible, with lower air velocity in the rooms and a resulting comfortable indoor environment.

Due to the above mentioned moderately high water temperatures used in chilled ceilings, it is possible to deliver cold water with a closed wet cooling tower during most of the cooling period. The cooling tower can be combined with a refrigeration machine or it can be used alone, if some overheating is allowed during short periods, or if energy storage or nightcooling techniques are used.

Closed wet cooling towers were conventionally used to remove excess heat from various industrial processes, with a usual range of 32–46°C hot water temperatures and typical cooling capacities above 40 kW. A recent research work performed in Switzerland, [1], showed that these towers are greatly overpowered in airflow and spray water rate, when applied to the range of 22–25°C hot water temperatures, as needed for chilled ceilings. Tower design for lower cooling loads — ≤ 10 kW — leads to smaller tower dimensions.

Existing models to predict thermal performance of closed wet cooling towers were developed for conventional units and operating conditions: high water temperatures and cooling capacities. It is the objective of this work to verify the applicability of existing models to smaller towers, adapted for use with chilled ceilings. For that purpose, a new cooling tower built specifically for use with chilled ceilings was tested. Experimental results were used to introduce corrections to existing model correlations.

2. Thermal models for a closed wet cooling tower

Thermal models to predict cooling tower performance can be classified as detailed models or correlation based models. Detailed models are based on a CFD-type approach, involving the numerical solution of differential equations for air/spray water flow, energy and water vapour concentration. After velocity, temperature and humidity fields are calculated, transfer coefficients can be calculated as a result. Examples of such models, which need numerical codes and high performance computers, can be found in Ref. [2]. Although a detailed analysis of air and spray water distribution in the tower is possible, they have some limitations, namely regarding boundary conditions in the tubes. Water flow inside the tubes has to be treated separately, which poses some practical problems: for instance, heat flux at tube surface is not uniform. These models and codes also consume considerable computing time and require a certain degree of specialization.

Correlation based models may also need the solution of differential equations, although not necessarily. However these equations result from local energy and mass balances, after some simplifications are introduced and assuming mass and heat transfer coefficients can be calculated first. These coefficients are calculated through experimental correlations. This approach will be used in this work, since it is more practical and may also lead to improved accuracy.

Fig. 1 represents schematically a closed wet cooling tower, with the main variables involved.

All correlation based models assume that the tube surface is completely wet, through a uniform distribution of spray water over the tube bundle. Therefore, mass and heat transfer occurs between an water film and air flow. For simultaneous heat and mass transfer, heat flux can be calculated through enthalpy potential [3]:

$$q = \alpha_m (h_i - h_{air}) \tag{1}$$

where α_m is the mass transfer coefficient for water vapour and h_i is the enthalpy at water film/air interface.

An enthalpy balance, for an elementary transfer surface dA, can be expressed as

$$m_{air}\,dh_{air} = \alpha_m (h_i - h_{air})\,dA \tag{2}$$

which is known as the Merkel equation [4] and which integration for the whole tower gives

$$\frac{\alpha_m A}{m_{air}} = \int_{in}^{out} \frac{dh_{air}}{(h_i - h_{air})} \tag{3}$$

The local energy balance, for the elementary surface dA in Fig. 2, can be written as

$$k(T_w - T_i)\,dA = \alpha_m (h_i - h_{air})\,dA - m_{spray} c_{p,\,spray}\,dT_{spray} \tag{4}$$

neglecting spray rate variation and with k being the heat transfer coefficient between water inside tube and film/air interface. It can be calculated by adding all thermal resistances:

$$\frac{1}{k} = \frac{1}{\alpha_w}\frac{d_{ext}}{d_{int}} + \frac{d_{ext}}{2k_{tube}}\ln\frac{d_{ext}}{d_{int}} + \frac{1}{\alpha_{spray}} \tag{5}$$

using external tube surface as reference (d_{ext}).

Several models can be found in the literature, differing in simplifications assumed. Four different models will be considered. When calculating tower performance, inlet conditions

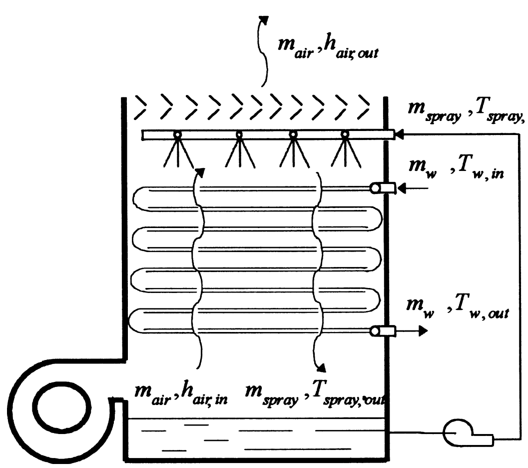

Fig. 1. Closed wet cooling tower and main variables.

(water and air) are known and outlet conditions are a model output. An iterative procedure is usually applied: water outlet temperature is guessed, which allows to calculate outlet air properties and the enthalpy integral in Eq. (3); since the integral value will not be equal to the first member of Eq. (3) — available number of transfer units, NTU — a new outlet temperature is used until equality is obtained. Models differ in the method used to calculate the integral in Eq. (3).

The first model to be considered, developed by Mizushina, [5], neglects spray temperature variation in Eq. (4) and considers interface conditions (h_i, T_i) constant throughout the cooling tower. Integration of Eq. (3) then gives

$$\frac{\alpha_m A}{m_{air}} = \ln\frac{h_i - h_{air,\,in}}{h_i - h_{air,\,out}} \tag{6}$$

The integration of Eq. (4) and substitution in Eq. (6) leads to

$$\frac{T_{w,\,in} - T_i}{T_{w,\,out} - T_i} = \frac{h_i - h_{air,\,in}}{h_i - h_{air,\,out}} \exp\left(\frac{k}{\alpha_m c_{pw}} \frac{m_{air}}{m_w}\right) \tag{7}$$

which together with the function $h_i = h_i(T_i)$ forms a non-linear set of equations for calculating (T_i, h_i).

This is an overall model, since temperature/enthalpy variation inside the tower is not calculated.

The second model, from Kals [6] uses a discretisation of the tube transfer surface. The domain is divided in several nodes to allow the calculation of water temperature and air enthalpy. Spray water temperature variation is also neglected. To evaluate local air enthalpy the following equation is used

$$\frac{dh_{air}}{dT_w} = \frac{\Delta h_{air}(j)}{\Delta T_w(k)} = \exp\left(\frac{m_w\ c_{pw}}{m_{air}}\right) \tag{8}$$

where j and k are subscripts representing different nodes. Interface properties (T_i, h_i) are

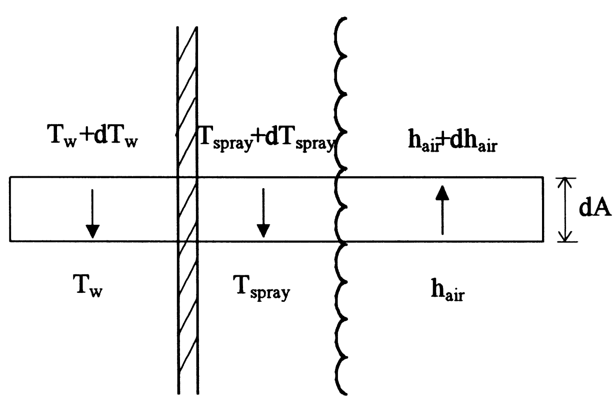

Fig. 2. Elementary transfer surface.

calculated locally — in each air node. After calculation of T_w and h_{air} in each node, the enthalpy integral in Eq. (3) is calculated by numerical integration.

The third model to be considered was also formulated by Mizushina [5]. Spray water temperature variation is not neglected. It is assumed that spray water film thermal resistance is negligible, which implies $T_i = T_{spray}$. Three differential equations for water temperature, air enthalpy and spray temperature are solved by numerical integration — finite differences [5].

The fourth model was presented by Peterson [7] and uses the concept of cooling efficiency. It also uses a global approach. Spray temperature variation is neglected and a linearization is adopted for saturation enthalpy as a function of wet bulb temperature.

All these models need experimental correlations for mass and heat transfer coefficients. Among others, Mizushina [8] found a relationship between mass transfer coefficient, air and spray flow rates:

$$\alpha_m a = 5.028 \times 10^{-8} Re_{air}^{0.9} Re_{spray}^{0.15} d_{ext}^{-2.6} \tag{9}$$

where a is the ratio between heat transfer surface and heat exchanger volume. It is valid for Re_{spray} between 50 and 240, Re_{air} between 1200 and 14,000 and d_{ext} between 12 and 40 mm. In this correlation, mass transfer coefficient depends on spray flow rate. Other authors have obtained correlations that just use air flow rate as variable [9,10]. In fact, Niitsu [10] concluded that mass transfer coefficient is independent of spray flow rate for $\Gamma/d_{ext} > 0.7$, where Γ is the spray water load, equal to spray mass flow rate divided by tower section.

Heat transfer coefficient between tube surface and water film, α_{spray}, can be correlated with spray load and tube diameter. Mizushina [8] presented the following relationship:

$$\alpha_{spray} = 2100\,(\Gamma/d_{ext})^{1/3} \tag{10}$$

valid for Γ/d_{ext} between 0.2 and 5.5 kg/m^2/s.

3. New closed wet cooling tower and experimental results

A new closed wet cooling tower was designed in order to be used with chilled ceilings. Design conditions were a cooling capacity of 10 kW, for an inlet water temperature of 21°C, a water flow rate of 0.8 kg/s and an air wet bulb temperature of 16°C. The tower has a section of 0.6 × 1.2 m and a height of 1.55 m. The tube bundle has 228 staggered tubes of 10 mm outside diameter, with a total transfer area of 8.6 m^2. This corresponds to a much smaller size than usual towers. The load/volume ratio of this tower is equal to 29 kW/m^3 (per heat exchanger volume), with a ratio between heat transfer surface and heat exchanger volume of 25 m^2/m^3.

The tower was manufactured by Sulzer Escher Wyss (Lindau, Germany). It is much smaller than cooling tower models usually built by this manufacturer. A forced draft configuration was chosen with a crossflow fan located at air entrance. This arrangement has a lower noise level, and also leads to a lower pressure drop. It was also chosen to facilitate air flow measurements.

A test facility was assembled at Porto to test this cooling tower. It is shown schematically in Fig. 3, including main instrumentation used. The thermal load was modeled with an electrical heater located in a water tank. Tower inlet water temperature was controlled by varying heating power — PID controller. Fan speed was also controlled by means of a frequency controller, which allowed changing air flow rate. Spray water flow rate could be changed manually (valve with seven fixed positions). Cooling water flow rate could also be changed, by changing pump speed and by using regulating valves.

Air flow rate was measured with one vane anemometer, located in different points of tower outlet section. In order to measure cooling water temperature evolution, several thermocouples

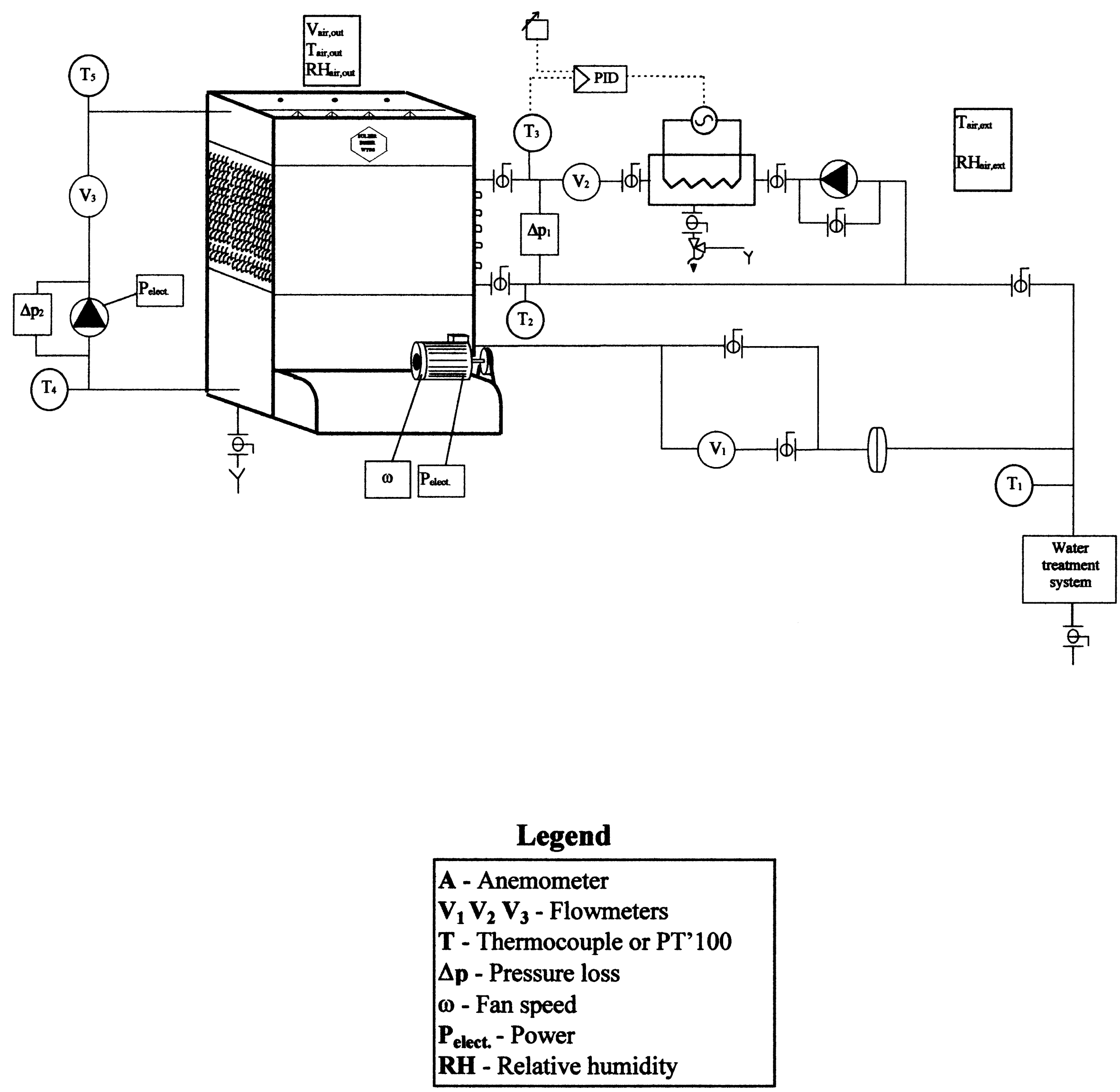

Fig. 3. Experimental facility for cooling tower testing.

were connected to the tubes. The data acquisition system used a data logger — HP 34970A — and HP VEE as software.

Tower thermal performance was expressed by means of tower thermal efficiency. This is defined as

$$\varepsilon = \frac{T_{w,\,in} - T_{w,\,out}}{T_{w,\,in} - T_{wb}} \tag{11}$$

where $T_{w,\,in}$ is inlet water temperature, $T_{w,\,out}$ outlet water temperature and T_{wb} the inlet wet bulb air temperature.

Experiments were carried out to analyse the influence of inlet water temperature. It was found that this parameter has a very little influence in tower efficiency. The effect of spray water flow rate can be seen in Fig. 4. An increase in spray rate increases efficiency up to a certain level. Above a rate of about 1 kg/s, an increase in spray rate does not improve significantly tower performance, because tube surface is almost completely wet. This means that an optimum spray rate can be found for this tower, regarding cooling capacity and water/energy (pumping) consumption. This analysis can only be done through experimental measures, since all thermal models assume tube surface is completely wet.

Then, if spray rate is kept close to the maximum or optimum value, tower efficiency is a function of three parameters: air flow rate, cooling water flow rate and wet bulb temperature. Fig. 5 shows experimental efficiency values for different air and water rates, for the same wet bulb temperature. As expected, efficiency increases with air flow rate and decreases with an increase in water flow rate (due to a decrease in water temperature difference).

Air wet bulb temperature has an influence in tower efficiency which is shown in Fig. 6.

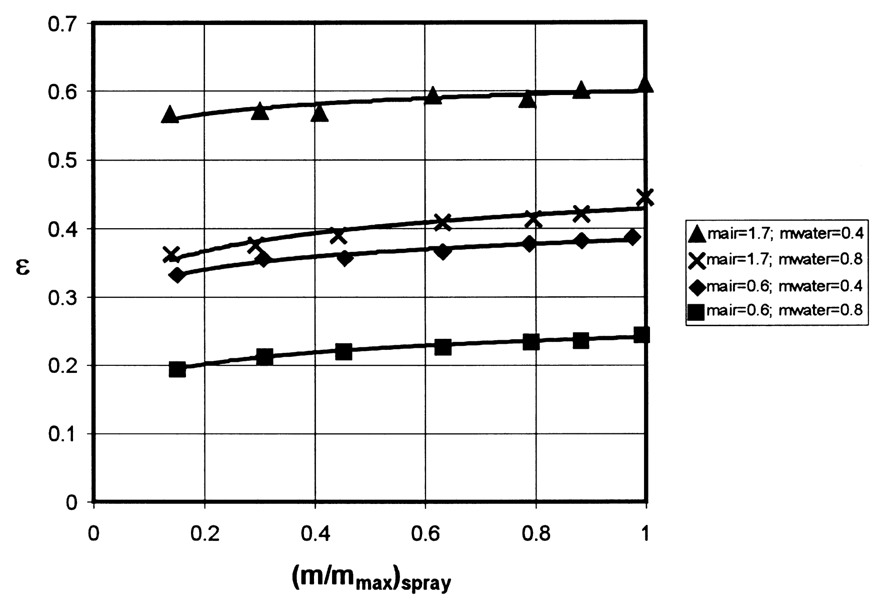

Fig. 4. Tower efficiency as a function of spray water flow rate, for different air and water rates: $(m_{max})_{spray} = 1.39$ kg/s, wet bulb temperature = 12.6°C.

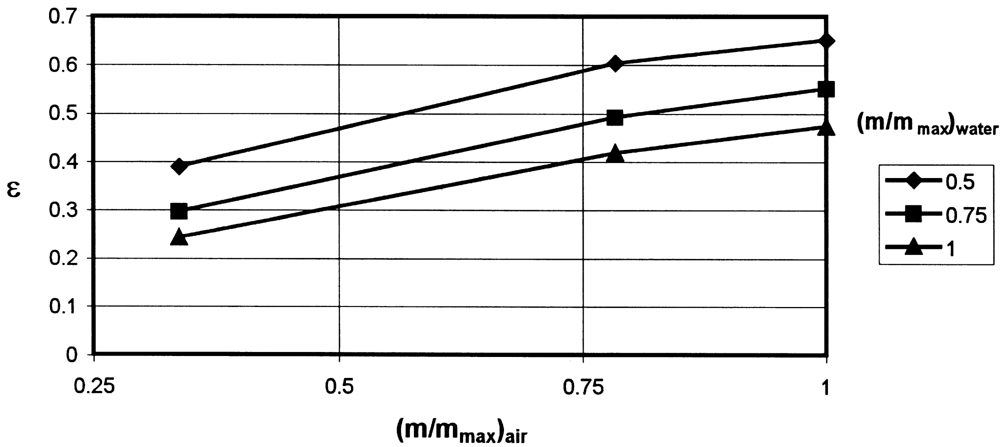

Fig. 5. Tower efficiency as a function of air and water mass flow ratios, for a wet bulb temperature of 15.8°C; $(m_{max})_{water} = 0.8$ kg/s, $(m_{max})_{air} = 1.7$ kg/s.

Efficiency increases slightly with wet bulb temperature: about 8% (absolute value) for temperatures between 10 and 20°C. The increase is linear and the influence is similar for different air and water flow rates.

A global coefficient of performance (COP) can be defined for the cooling tower. It is equal to cooling capacity divided by energy input (pumping and fan consumption). By measuring electricity input in the fan motor and spray flow rates, with $T_{w,\,in} = 21°C$ and $T_{wb} = 16°C$.

Mass and heat transfer coefficients (α_m and α_{spray}) were calculated for different operating conditions. These coefficients were correlated to, respectively, air flow rate and spray flow rate, following the same approach from other authors [8,9,10]. The following correlations were obtained:

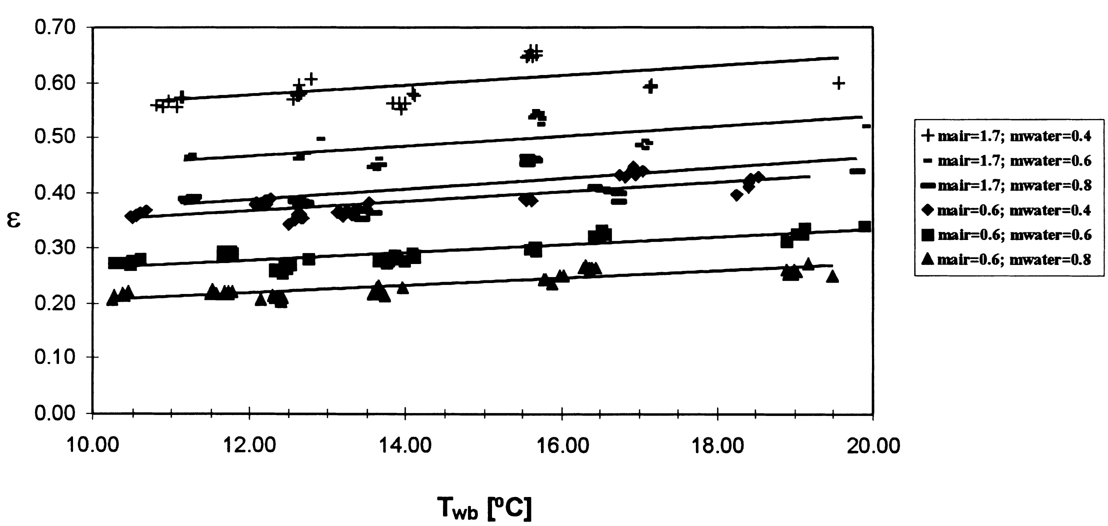

Fig. 6. Influence of air wet bulb temperature in tower efficiency for different air and water mass flow rates.

$$\alpha_{\mathrm{m}} = 0.1703(m/m_{\mathrm{max}})_{\mathrm{air}}^{0.8099} \tag{12}$$

$$\alpha_{\mathrm{spray}} = 700.3(m/m_{\mathrm{max}})_{\mathrm{spray}}^{0.6584} \tag{13}$$

valid for $0.4 < m_{\mathrm{w}} < 0.8$ kg/s, $10 < T_{\mathrm{wb}} < 20°\mathrm{C}$, $15 < T_{\mathrm{w,\,in}} < 28°\mathrm{C}$, with $(m_{\mathrm{max}})_{\mathrm{air}} = 1.7$ kg/s and $(m_{\mathrm{max}})_{\mathrm{spray}} = 1.39$ kg/s.

4. Model results

The different thermal models described previously were applied to the new cooling tower geometry.

Model 2 was implemented with a fixed number of nodes, equal to 13 — number of tubes in one column + 1. Model 3, a finite differences model, can be implemented with any number of nodes. However, above 25 nodes no significant difference was noted in the calculated tower efficiency.

Fig. 7 shows experimental and calculated efficiencies using Mizushina's correlations — Eqs. (9) and (10). Results for models 1, 3 and 4 are quite close, but all calculated efficiencies are above experimental values. The lower experimental efficiencies must be due to the particular geometrical conditions of the cooling tower: smaller tower dimensions and non-uniformity in air velocity in the tower section, which is caused by fan position. This non-uniformity of the air flow over the tube bundle was noted during air velocity measurements.

Fig. 8 shows experimental and calculated efficiencies when using experimental correlations for mass and heat transfer coefficients — Eqs. (12) and (13). Calculated results are closer to experimental ones, and the difference between the 4 models is smaller. Model 1 and 2 seem to give the best results in this case.

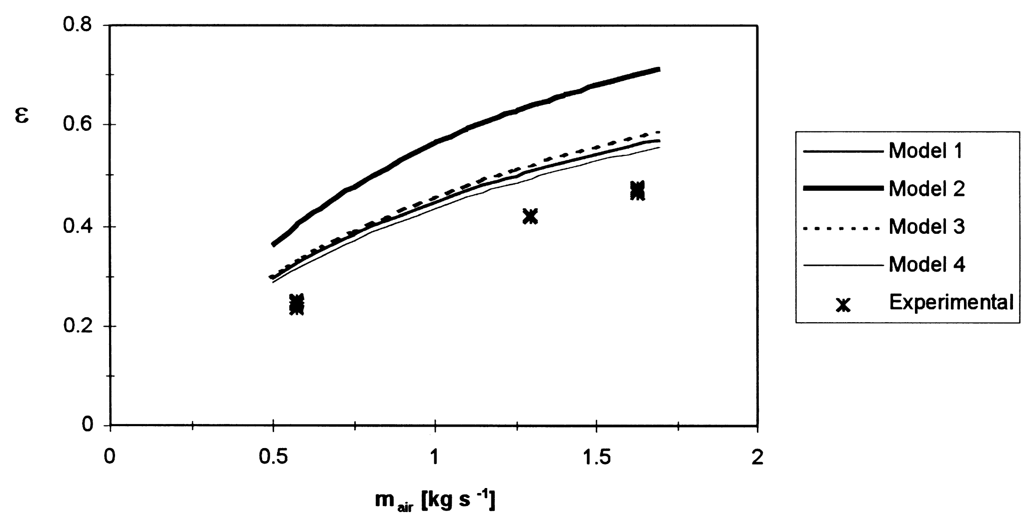

Fig. 7. Comparison of experiment and models using Mizushina's correlations for mass transfer and heat transfer between tube surface and water film. Water rate = 0.8 kg/s, spray rate = 1.37 kg/s, wet bulb temperature = 16°C.

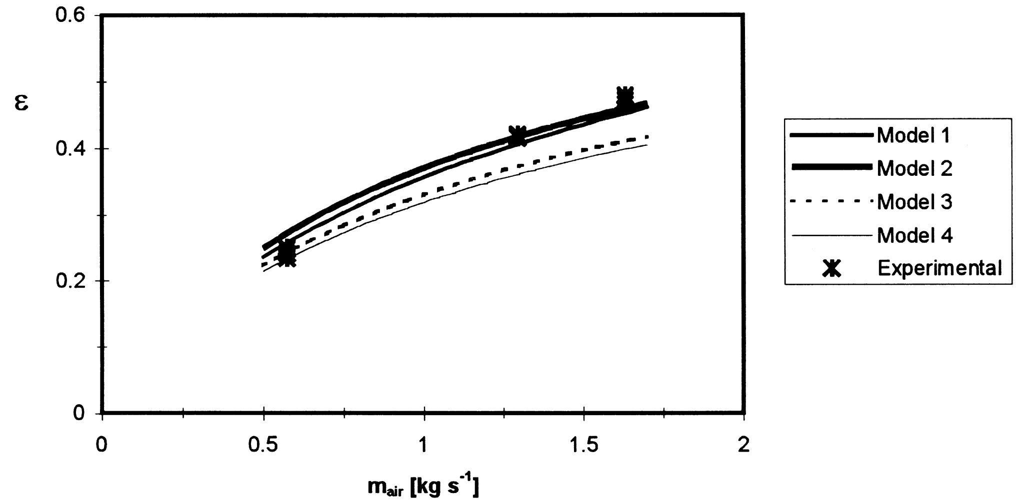

Fig. 8. Comparison of experiment and models using experimental correlations for mass and heat transfer. Water rate = 0.8 kg/s, spray rate = 1.37 kg/s, wet bulb temperature = 16°C.

Fig. 9 shows experimental and calculated efficiencies while varying air wet bulb temperature, using experimental correlations for mass and heat transfer coefficients. A good accuracy is also obtained, particularly with models 1 and 3.

Water temperature profile along the tubes (water flow direction) was calculated with model 3. As Fig. 10 shows, the calculated temperature profile is quite close to experimental one.

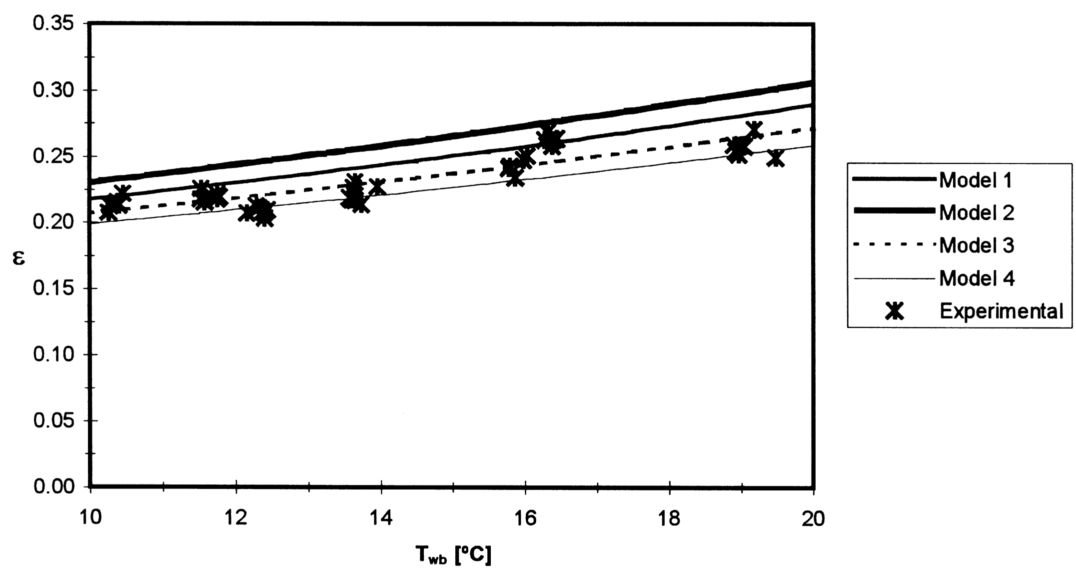

Fig. 9. Influence of air wet bulb temperature in tower efficiency; air rate = 0.6 kg/s, water rate = 0.8 kg/s, spray rate 1.37 kg/s. Models with experimental correlations.

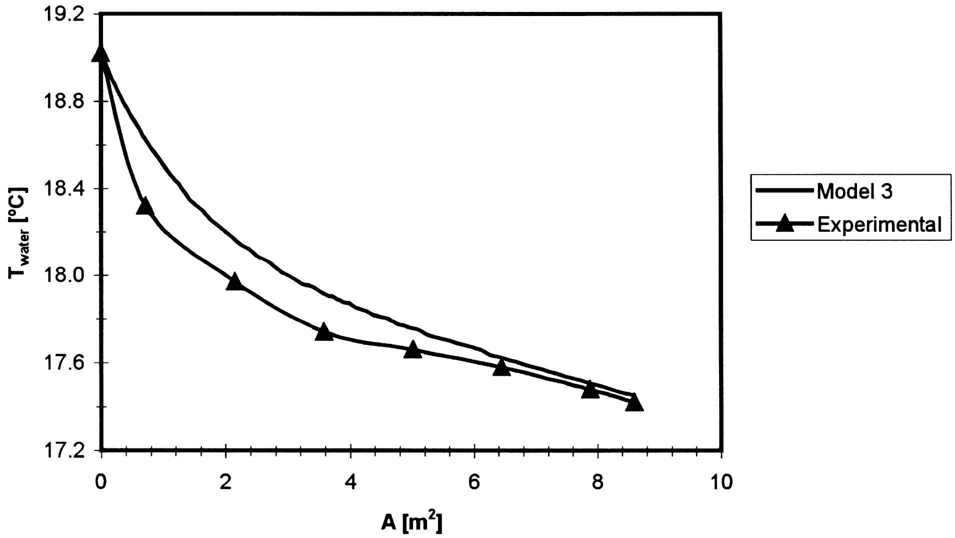

Fig. 10. Variation of water temperature along tube banks. Air rate = 0.7 kg/s, water rate = 0.8 kg/s, spray rate = 1.39 kg/s, wet bulb temperature = 15.6°C. Model with experimental correlations.

The use of the thermal models also confirmed one experimental conclusion: the influence of water inlet temperature in tower efficiency is negligible.

5. Conclusions

Thermal models are very useful both for designing and predicting cooling tower performance. This is not always possible with experiment, since it is difficult to reproduce all possible operating conditions. Different thermal models to predict the performance of closed wet cooling towers were presented.

A new closed wet cooling tower, adapted for use with chilled ceilings, was tested and experimental correlations for mass and heat transfer coefficients were obtained.

Different models were used and their results compared to experimental ones. It was found that, for such a small tower, the models give better results when using the new correlations. It was also found that simpler models, with a global approach, can give as good, or even better, results as models based on finite difference techniques.

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