INTERNATIONAL JOURNAL OF ENERGY RESEARCH
Int. J. Energy Res. 2000; **24**:1171–1179

Thermal performance of a closed wet cooling tower for chilled ceilings: measurement and CFD simulation

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SUMMARY

A closed wet cooling tower, adapted for use with chilled ceilings in buildings, was tested experimentally. The thermal efficiency of the cooling tower was measured for different air flow rates, water flow rates, spray flow rates and wet bulb air temperatures. CFD was also used to predict the thermal performance of the cooling tower. Good agreement was obtained between CFD prediction and experimental measurement. Copyright © 2000 John Wiley & Sons, Ltd.

KEY WORDS: cooling tower; chilled ceiling; thermal performance; CFD

1. INTRODUCTION

Increased use of IT equipment coupled with thermal comfort requirements has led to an increased demand for cooling in offices. A simple and economic way of cooling offices is to use an indirect-contact closed wet cooling tower combined with chilled ceilings.

Chilled ceilings are a relatively new approach to cooling. A chilled ceiling system has several advantages over conventional cooling systems. The use of chilled ceilings allows the ventilation rate to be reduced to a minimum level, and so reduces ventilation losses. Heat transfer from indoor space to chilled ceilings is accomplished mainly by radiation. Consequently, chilled ceilings enable offices to have low air movement and a comfortable indoor environment. Radiative heat transfer also allows chilled ceilings to remove considerable heat loads, up to 75 W m^{-2}, at a relatively small temperature difference between the room air and the ceiling. This makes it possible to operate the system with water supply temperatures of about 18–20°C.

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Contract/grant sponsor: Commission of the European Union (DG XII).

Received 5 October 1999
Accepted 7 January 2000

Because high water temperatures can be used in chilled ceilings, it is possible to use water from a closed wet cooling tower during most of the cooling period. The cooling tower could be combined with a supplementary refrigeration plant, or used alone if a short period of overheating is allowed or energy storage or night-cooling techniques are used.

Closed wet cooling towers have been used to remove excess heat from various industrial processes with hot water temperatures between 32 and 46°C and typical cooling capacities above 40 kW. For chilled ceilings, cooling towers with lower capacities (cooling loads $\leqslant 10$ kW) and smaller tower dimensions can be used with hot water temperatures ranging from 22 to 25°C.

Most of the existing thermal models for tower design make use of empirical correlations for heat and mass transfer coefficients obtained for large towers. The validity of such correlations cannot be extended to all geometries or air flow patterns and so tests are often required for individual tower designs. Computational fluid dynamics (CFD) is a useful alternative tool for designing both small and large cooling towers. CFD models give information on air flow and spray water distribution, which cannot be obtained with simplified models. Such information is important for good tower design and operation. An unsatisfactory distribution of spray water over the tubes would reduce evaporative cooling and tower cooling capacity. Simultaneous non-uniform distribution of air and spray water would further degrade the tower-thermal performance.

This project is concerned with designing and testing a prototype closed wet cooling tower for use with chilled ceilings. CFD has been used for the design and performance evaluation. This paper presents the experimental and computational results of the tower performance.

2. EXPERIMENTAL MEASUREMENT

Figure 1 shows the schematic of the closed wet cooling tower, with notation for the main flow variables. The cooling tower had a horizontal cross-section of 0.6 m × 1.2 m and a height of 1.55 m. The heat exchanger was composed of 228 staggered tubes of 10 mm outside diameter

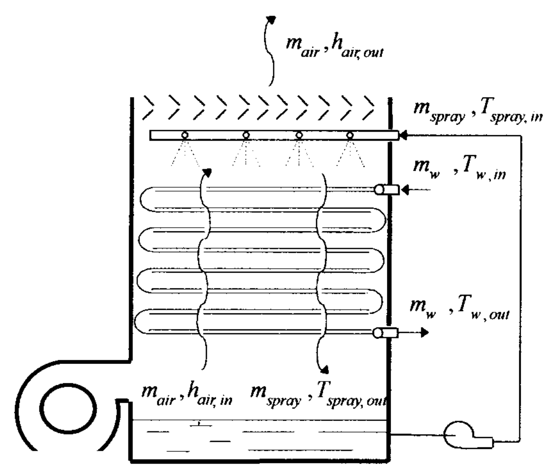

Figure 1. Schematic diagram of the closed wet cooling tower and main variables.

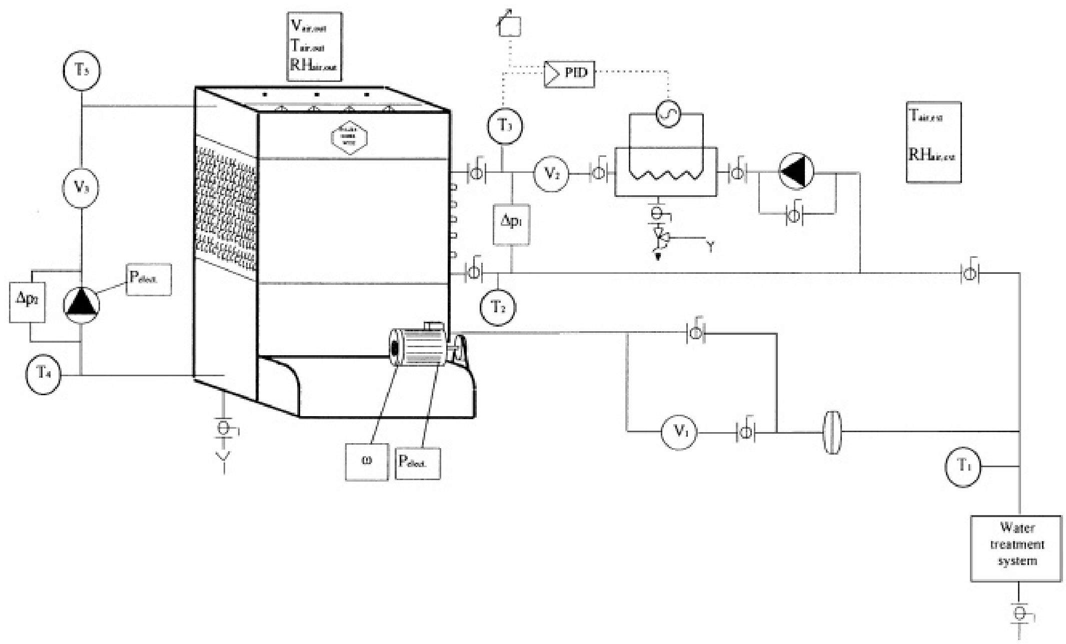

Figure 2. Experimental facility for cooling tower testing. A-Anemometer, $V_1V_2V_3$-Flowmeters, T-Thermocouple or PT'10.0, Δp-Pressure loss, ω-Fan speed, $P_{elect.}$-Power, RH-Relative humidity.

and had a total heat transfer area of 8.6 m^2. The design conditions for the tower had a cooling capacity of 10 kW, inlet water temperature of 21°C, water flow rate of 0.8 kg s^{-1} and wet bulb air temperature of 16°C. The tower was manufactured by Sulzer Escher Wyss (Lindau, Germany) and was much smaller than the traditional cooling towers for industrial use. A forced draft configuration was chosen with a crossflow fan located at the air entrance. This arrangement gave rise to a low-noise level and a low-pressure drop. It also led to good air distribution and facilitates air flow measurements.

An experimental facility was assembled at Porto to test this cooling tower. Figure 2 shows the schematic diagram of the test facility including the main instrumentation used. The facility allowed various flow parameters such as the water flow rate and temperature, spray water flow rate and air flow rate to be varied at controlled conditions. The thermal load was modelled with an electrical heater located in a water tank. The tower inlet water temperature was controlled by varying the heating power through a PID controller. The fan speed, and so air flow rate, was controlled by means of a frequency controller. The spray water flow rate could be changed manually through a valve with seven opening levels. The cooling water flow rate could also be changed by varying the pump speed and by adjusting flow valves.

The air flow rate was measured with a vane anemometer. The anemometer could be traversed across the tower outlet section to measure the air speed at different points. Several thermocouples were connected to the tubes to measure cooling water temperature evolution. The data acquisition system included a data logger HP 34970A with software HP VEE.

The thermal performance of the cooling tower was expressed by means of tower thermal efficiency. This is defined as

$$\varepsilon = \frac{T_{\mathrm{w,in}} - T_{\mathrm{w,out}}}{T_{\mathrm{w,in}} - T_{\mathrm{wb}}} \tag{1}$$

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

3. CFD SIMULATION

A commercial CFD software package (FLUENT User's Guide, 1993) was employed to simulate the two-phase flow of air and water droplets in the cooling tower. Air flow was modelled as a continuous phase and water droplets as a dispersed phase. The impact of the dispersed phase on the continuous phase flow was accounted for by coupling the momentum and the heat and mass transfer between the two phases (Gan, Riffat, 1999).

3.1. Continuous phase model

The continuous phase model for turbulent air flow consists of the conservation equations for mass, momentum, enthalpy, concentration and turbulence. For steady-state incompressible flow, the model can be represented by the following equation:

$$\nabla \cdot (\rho \bar{V} \phi - \Gamma_{\phi} \nabla \phi) = S_{\phi} + S_{\phi}^{p} \tag{2}$$

where ϕ is the flow variable, $\bar{V}$ is the mean air velocity (m s^{-1}), ρ is the air density (kg m^{-3}), Γ_{ϕ} is the diffusion coefficient (N s m^{-2}), S_{ϕ} is the source for the continuous phase and S_{ϕ}^{p} is the source due to the interaction between air and water droplets.

3.2. Dispersed phase model

The flow of dispersed water droplets is characterised by the trajectories. The trajectory of a water droplet is computed using the following equation:

$$\rho_p \frac{\mathrm{d}V_p}{\mathrm{d}t} = \frac{3}{4}\frac{\rho C_D |V - V_p|}{d_p}(V - V_p) + g(\rho_p - \rho) + \frac{1}{2}\rho \frac{\mathrm{d}}{\mathrm{d}t}(V - V_p) + \frac{\partial P}{\partial r_p} \tag{3}$$

where V is the instantaneous air velocity (m s^{-1}), V_p is the instantaneous velocity of the droplet (m s^{-1}), C_D is the drag coefficient, d_p is the droplet diameter (m), ρ_p is the droplet density (kg m^{-3}), P is the static pressure (Pa) and r_p is the droplet trajectory (m).

3.3. Assumptions

The following assumptions were used to predict the thermal performance of the cooling tower:

- The tower was quite long compared with the width, so two-dimensional flow was considered. The tower casing was adiabatic and impermeable.
- For the two-dimensional flow on one vertical cross-section of the tower, the tubes representing the heat exchanger were taken as separate entities. The heat transfer from the tube surface to air

involves evaporation and convection. However, because the software could not take account of evaporative cooling of tubes, only the sensible heat transfer was used in terms of a surface heat flux. The estimated sensible heat transfer for a complete wet cooling tower was about 24 per cent of the total heat transfer from the heat exchanger to air and spray water.

- On the same vertical section, the spray water composed of 100 trajectories of droplets was injected into the tower through a nozzle at the centre line above the heat exchanger. The injection velocities of the water droplets from the nozzle varied such that the spray water would cover the entire width of the heat exchanger when falling on top of it. The mean diameter of the water droplets was estimated from the terminal velocity of the droplets at a mean velocity of air flowing over the heat exchanger.

4. MEASUREMENT RESULTS

Experiments were first carried out to analyse the influence of inlet water temperature. It was found that this parameter had very little effect on the tower efficiency. Tests were then carried out at various spray water flow rates. The effect of spray water flow rate can be seen from Figure 3. An increase in spray rate increases the efficiency up to a certain level. Above a rate of about 1 $kg\,s^{-1}$, an increase in spray rate does not significantly improve tower performance because the tube surface is almost completely wet. This means that, there exists an optimum spray rate for a given air flow rate, with regard to cooling capacity and water/energy (pumping) consumption. When both air and spray water flow rates are considered, the mass flow ratio of spray water to air can be employed and this ratio can also be optimized according to the cooling capacity and energy consumption.

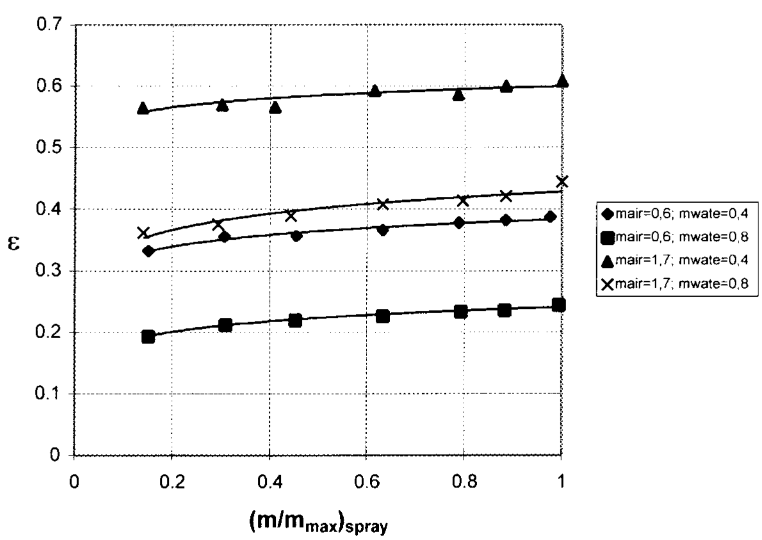

Figure 3. Tower efficiency as a function of spray water flow rate for different air and water rates [$(m_{max})_{spray}$ = 1.39 $kg\,s^{-1}$, wet bulb temperature = 12.6°C].

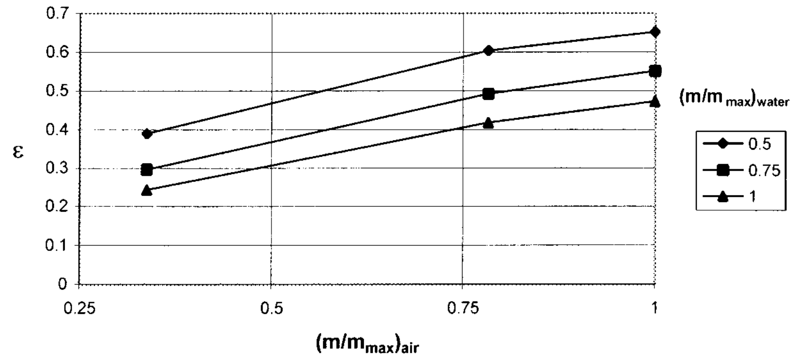

Figure 4. Tower efficiency as a function of air and water mass flow ratios for a wet bulb temperature of 15.8°C [$(m_{\max})_{\text{water}} = 0.8\ \text{kg s}^{-1}$, $(m_{\max})_{\text{air}} = 1.7\ \text{kg s}^{-1}$].

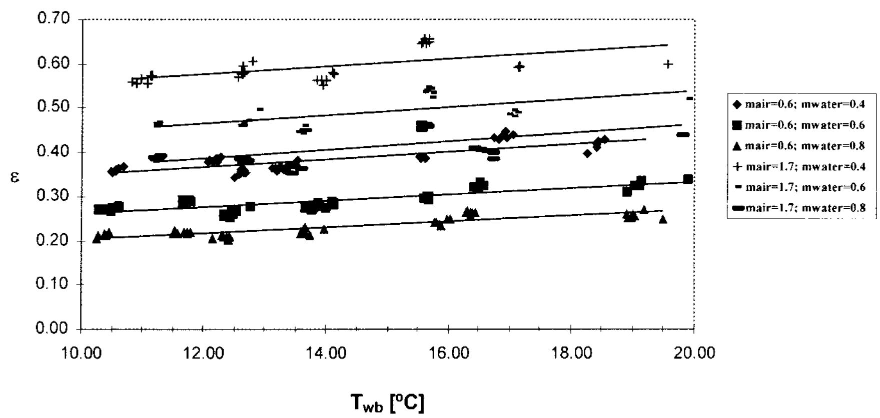

Figure 5. Influence of wet bulb temperature on tower efficiency for different air and water mass flow rates.

If the spray rate is kept close to the maximum or optimum value, the tower efficiency is a function of three parameters: air flow rate, cooling water flow rate and wet bulb temperature. Figure 4 shows the measured efficiency values for different air and water flow rates at a fixed wet bulb temperature. As expected, the efficiency increases with the air flow rate and decreases with the increasing water flow rate (due to a decrease in water temperature difference).

The wet bulb temperature of air has also an influence on the tower efficiency as shown in Figure 5. The efficiency increases slightly with the wet bulb temperature: about 8 per cent for temperatures varying between 10 and 20°C. The increase is linear for different air and water flow rates. However, increasing the wet bulb temperature would increase the outlet water temperature from the heat exchanger, thus decreasing the cooling capacity of the chilled ceiling.

A global coefficient of performance (COP) can be calculated for the cooling tower. It is equal to the cooling capacity divided by the total energy input (pumping and fan power consumption). The total energy input can be determined by measuring the electricity input to the fan motor and spray pump motor. The calculated COP at maximum air and spray flow rates was 6.

5. COMPARISON BETWEEN SIMULATION AND MEASUREMENT

A number of CFD simulations were performed to compare with the experimental measurement. The thermal performance of the cooling tower was predicted according to the temperature distribution of the tubes representing the temperature of cooling water in the heat exchanger. Simulations were divided into two groups, one for tests at the design water flow rate of 0.8 $kg\,s^{-1}$ and the other for tests at a lower water flow rate of 0.4 $kg\,s^{-1}$.

Table I presents a comparison between the predicted and measured tower performance for four test cases at the design water flow rate of 0.8 $kg\,s^{-1}$. In the first two tests, the temperature of supply air was lower than that of spray water and cooling water at the outlet of the heat exchanger. The measured air flow rate for the second test (1.08 $m^3\,s^{-1}$) was higher than that for the first test (0.48 $m^3\,s^{-1}$). In the third test, the temperature of supply air was slightly higher than the spray water temperature. In the fourth test, the temperature of supply air was higher than that of spray water and cooling water. The measured temperature drop of cooling water between the inlet and outlet of the heat exchanger for these tests was approximately 1.5 K. It is seen that the simulations (Preds. 1–4) are in good agreement with the measurement in terms of the temperature difference of cooling water at the inlet and the outlet.

In Table II, prediction is compared with the measurement for one test at the low water flow rate of 0.4 $kg\,s^{-1}$. The predicted temperature difference between the top and the bottom tube rows (1.5 K for Pred 1) was much smaller than the measured temperature drop of cooling water (3 K for test). For a given cooling load, the temperature drop of the cooling water is inversely proportional to the water flow rate in the heat exchanger. The measured water temperature drop would therefore be 1.5 K for a water flow rate of 0.8 $kg\,s^{-1}$ compared with 3 K for 0.4 $kg\,s^{-1}$.

Table I. Comparison between the predicted and measured thermal performance of the cooling tower at the design water flow rate of 0.8 $kg\,s^{-1}$.

	Supply air			Spray water			Chilled water		
Case	$Q_{air,in}$ (m^3 s)	$T_{air,in}$ (°C)	$RH_{air,in}$ (%)	m_{spray} (kg s)	$T_{spray,in}$ (°C)	m_r	$T_{w,in}$ (°C)	$T_{w,out}$ (°C)	ΔT (K)
Test 1	0.48	13.1	85	1.38	16.2	2.34	18.53	17.02	1.51
Pred 1	0.48	13.1	85	1.38	16.2	2.34	19.1	17.5	1.6
Test 2	1.08	13.0	87	1.38	13.6	1.04	15.86	14.35	1.51
Pred 2	1.08	13.0	87	1.38	13.6	1.04	16.3	14.9	1.4
Test 3	0.48	15.7	51	1.38	15.0	2.36	17.24	15.76	1.48
Pred 3	0.48	15.7	51	1.38	15.0	2.36	18.1	16.8	1.3
Test 4	0.48	20.7	45	1.38	18.1	2.41	20.38	18.90	1.48
Pred 4	0.48	20.7	45	1.38	18.1	2.41	20.6	19.2	1.4

$m_r = m_{spray}/(Q\rho)_{air}$: mass flow ratio of spray water to air
$\Delta T = T_{w,in} - T_{w,out}$

Table II. Comparison between the predicted and measured thermal performance of the cooling tower at a water flow rate of 0.4 $kg\,s^{-1}$.

	Supply air			Spray water			Chilled water		
Case	$Q_{air,in}$ (m^3 s)	$T_{air,in}$ (°C)	$RH_{air,in}$ (%)	m_{spray} (kg s)	$T_{spray,in}$ (°C)	m_r	$T_{w,in}$ (°C)	$T_{w,out}$ (°C)	ΔT (K)
Test	0.48	14.3	92	1.23	16.1	2.10	18.15	15.19	2.96
Pred 1	0.48	14.3	92	1.23	16.1	2.10	19.1	17.6	1.5
Pred 2	0.48	14.3	92	0.59	16.1	1.00	19.5	17.1	2.4
Pred 3	0.48	14.3	92	0.29	16.1	0.50	19.5	16.8	2.7
Pred 4	0.48	14.3	50	1.23	16.1	2.10	19.1	17.1	2.0
Pred 5	0.48	14.3	50	0.59	16.1	1.00	19.2	16.2	3.0
Pred 6	0.48	14.3	50	0.29	16.1	0.50	18.2	15.7	2.5
Pred 7	0.48	14.3	50	0.35	16.1	0.60	18.4	15.6	2.8

The predicted tube temperature difference would therefore agree with the measurement at the design water flow rate of 0.8 $kg\,s^{-1}$.

The difference between the predicted and the measured tube temperatures at the top and bottom rows was partly attributable to the high water-to-air mass flow ratio and high relative humidity of supply air. Consequently, air was saturated throughout the tower, at temperatures much higher than the wet bulb temperature of the supply air. Furthermore, in the wet cooling tower, evaporative cooling would also occur on the tube surfaces but this effect could not be included in the simulation as described in the assumption for the heat exchanger. To improve the prediction, the mass flow ratio and the humidity of supply air were artificially reduced.

When the mass flow ratio was halved (Pred 2 in Table II), the relative humidity of air through the heat exchanger was still very high (over 99 per cent) but was below the saturation level. Evaporative cooling of air could take place along the heat exchanger. The difference in the tube temperatures at the top and bottom of the heat exchanger increased to 2.4 K. When the mass flow ratio was further reduced to about one quarter of the original value, from 2.1 to 0.5 (Pred 3), the relative humidity varied between 95 and 96 per cent and the temperature reduction of the tubes increased to 2.7 K.

When the relative humidity of supply air was reduced from 92 to 50 per cent, while the mass flow ratio remained unchanged at 2.1 (Pred 4), the air in the tower except for the area near the bottom of the heat exchanger was still saturated and the temperature difference of the tubes at the inlet and the outlet increased from 1.5 to 2.0 K. For the same supply air at a relative humidity of 50 per cent but for a reduced mass flow ratio of one (Pred 5), the air was not saturated with a minimum relative humidity of 83 per cent at the bottom and a maximum of 98 per cent at the top of the heat exchanger. The predicted tube temperatures for the top and bottom rows were 19.2 and 16.2°C, respectively. The tube temperature difference became the same as the measured temperature drop for cooling water of 3 K. When the mass flow ratio was reduced to 0.5 (Pred 6), the predicted tube temperatures for the top and bottom rows were 18.2 and 15.7°C, respectively, which were not far from the measured cooling water temperatures at the inlet and outlet. The tube temperature difference between the top and bottom rows was 2.5 K. For a mass flow ratio of 0.6, the predicted tube temperature difference was 2.8 K (Pred 7).

Therefore, for a completely wet cooling tower operating at water flow rates lower than the design value, the prediction of thermal performance can be improved by artificially decreasing the mass flow ratio and supply air humidity.

6. CONCLUSIONS

The performance of a closed wet cooling tower for chilled ceilings has been measured experimentally. The measurements show that, there exists an optimum spray rate for a given air flow rate or an optimum mass flow ratio of spray water to air according to the cooling capacity and energy consumption. The efficiency of the cooling tower increases with the air flow rate and decreases with the increasing water flow rate. The efficiency increases slightly with the increasing wet bulb temperature of the supply air. However, increasing the wet bulb temperature would increase the outlet water temperature from the heat exchanger and decrease the cooling capacity of chilled ceilings.

CFD has also been used to predict the thermal performance of the cooling tower. The predicted thermal performance has been compared with experimental measurement. As the CFD package does not have the capacity to simulate the heat transfer between and within the tubes of the heat exchanger and the evaporative cooling of the tube surfaces, the effect of water flow rate on the temperature drop of the cooling water in the heat exchanger could not be predicted. Despite these shortcomings, the thermal performance of the cooling tower, in terms of the cooling water temperature drop, could still be predicted at the design water flow rate but the tower performance would be under-predicted when the water flow rate was lower than the design value and the tower was over-flooded with spray water. To improve the prediction for performance evaluation at water flow rates lower than the design value, appropriate measures could be adopted to compensate for the simplifications in the model so that the air through the heat exchanger would not be over-saturated. One such measure would be to reduce the mass flow ratio if it is higher than the optimum value.

ACKNOWLEDGEMENTS

The authors wish to thank the Commission of the European Union (DGXII) for partially funding the project, under Joule IV research programme. All partners of the ECOCOOL research project are greatly acknowledged including Sulzer Infra Lab (CH), Sulzer Escher Wyss Lindau (DE), Helsinki Univ. Technology (FIN)—HVAC Lab.

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