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Energy-Efficient Control of Evaporative Cooling Towers for Small Steam Power Plants

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Abstract: Forced draft evaporative cooling towers are often used for condenser cooling in small steam power plants. Conventional control strategies keep the cooling pump operating at full speed all the time, and control the cooling water temperature by modulating the tower fan speed. In this work, an energy-efficient control strategy is proposed, that minimizes the combined consumption of the tower fans and circulation pump, while also avoiding the sustained operation of the fans within some forbidden velocity range, to avoid resonance effects. The proposed strategy can be easily implemented within standard industrial control system, and is demonstrated to operate satisfactorily by a simulation study.

Keywords: Optimal operation and control of power systems; Control system design; Modeling and simulation of power systems

1. INTRODUCTION AND MOTIVATION

Many industrial processes require to cool some working fluid. For large thermal loads, like for example those encountered in power plants, a primary device used for that purpose is the evaporating cooling tower.

In a forced draft cooling tower, water is sprayed from atop against an ascending air stream, induced by fans. A small fraction of the water evaporates, whence the device name, and as a result the entire stream is cooled. Water then exits the tower at its bottom, and is collected to re-enter the process. A forced draft cooling tower thus consumes power to drive the water pumps and to drive the fans. Said consumption is often of non-negligible entity, so that its reduction yields a relevant benefit to the overall plant operation.

In this paper, we propose a control strategy for such devices, targeted at fulfilling the thermal load required by the process with a combination of water flow rate and tower outlet temperature that minimises the combined consumption of pumps and fans. We also present a validation of the strategy on an accurate dynamic model of the cooling system, subjected to realistically varying thermal loads and environmental conditions.

The paper is organised as follows. Section 2 gives a brief literature overview and further motivates the present work. In Section 3, the dynamic plant model used for the model-based control system design is described. A conventional control system for the system under consideration is described in Section 4; the proposed control strategy is illustrated in Section 5 and validated in simulation in Section 6. Concluding remarks and future work are discussed in Section 7.

2. BRIEF LITERATURE REVIEW

This section briefly reviews some related work, without any claim of being exhaustive, to better situate the present research in the overall scenario of modelling and control of cooling systems based on evaporating towers, thereby providing further motivation for it.

The literature contains many works on the design and correct sizing of cooling systems, to achieve energy savings in both the civil and the industrial field, see e.g. Kintner-Meyer and Emery (1995) or the document by the European Commission (2001). On the other hand, studies on control strategies for the same systems focus primarily on air conditioning. Schwedler (1998) argued that lowering the water temperature is not the most efficient strategy due to the high consumption of the fans, while Ahn and Mitchelly (2001) demonstrated that the exogenous variable that influences consumption in cooling towers is the wet bulb temperature, followed by the thermal load, while the dry bulb temperature has practically no influence.

The simplest and most widely used control strategy is to keep the tower outlet temperature to a constant set point, chosen as low as possible (Briley, 2003). Crowther and Furlong (2004) proposed to maintain a constant temperature difference (or *approach*) between the set point and the wet bulb, demonstrating that this choice is better than a fixed set point.

Along the same reasoning, works like Braun and Diderrich (1990), Ahn and Mitchelly (2001), Yao et al. (2004) and Sun and Reddy (2005) present strategies in which the set point is chosen as a function of the wet bulb temperature and in some cases of the thermal load, determined through multiple simulations. These strategies, however, are in general unable to determine the optimum set point for consumption minimisation, owing to model uncertainties. Also, for each operating condition, a large number of simulation runs is required.

Another research line aimed at cooling energy saving adopts optimal control strategies, as shown e,g, in Koeppel et al. (1995), Lu et al. (2004) and Ma et al. (2008), often based on genetic algorithms or neural networks, the former being considered the most promising (Ma et al., 2009).

Summarising, control-centric literature works are not primarily focused on industrial/power-generation tower-based cooling systems. In the authors' opinion, this leaves room for relevant improvements in that specific area, which motivates the present work.

3. PLANT MODEL

The presented study refers to the plant configuration shown in Figure 1. The study requires a detailed model of the cooling towers and of their thermal load, which is the steam condenser of a biomass-fired steam power plant. The boundary conditions for the system are given by the mass flow rate and specific enthalpy of the steam turbine exhaust. Most of the required components, such as the piping elements, the pump, the collecting reservoir and the condenser, were taken from the ThermoPower Modelica library (Casella and Leva, 2005, 2006), while the tower model was created on purpose for this work. Component parameters were taken from commercial datasheets. The specific application refers to an 18 MWe biomass-fired power plant, but the concepts presented here are general and applicable to a wide range of similar plants.

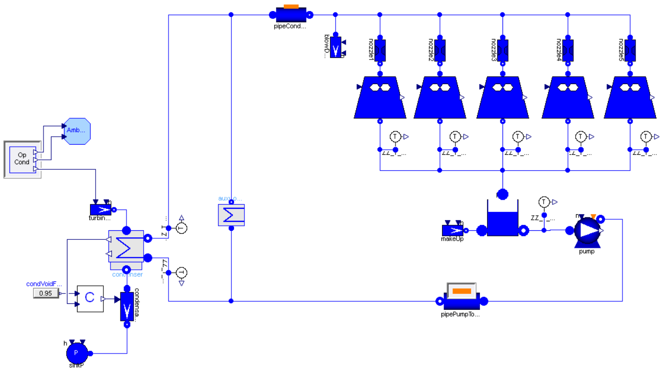

Fig. 1. The considered plant scheme as represented in Modelica.

The first component, on the left of Figure 1, is the condenser. Given the scope of the study, the steam inlet conditions are prescribed as know functions of the plant load. The condenser is a two-pass shell-and-tube heat exchanger, which is represented by two counter-current heat exchangers plus volumes to represent the energy storage in the two plena at the shell ends. The tube side heat transfer coefficient varies according to the Dittus-Boelter correlation, while the shell side (condensation) coefficient is assumed constant. A condensate extraction pump removes the condensed water from the bottom hot well in order to keep the condensate level constant; this is only needed to avoid the flooding of the condenser, but the actual way the condensate is extracted has no influence on the condenser behaviour. The condenser model has been validated successfully against design and off-design data from the manufacturer.

Continuing with Figure 1, the piping from the condenser to the towers is represented with enough detail to reproduce its dynamic effect (energy storage, transport delay) on the quantities of interest. Piping sections that are not relevant for the study, for example because they are excluded after the plant start-up, are not represented. The thermal input from auxiliary loads that are connected in parallel to the condenser is synthetically modelled by the injection of a fixed thermal load. This is justified by the fact that their heat duty is a small fraction of the condenser load, and by the lack of detailed information about them.

The cooling towers are modelled according to Merkel's theory, which is well-established in the field. The tower body is divided into N segments; within each of them, the forced air flow, travelling upwards, exchanges mass and heat with the downcoming water flow. The heat transfer is proportional to the difference between the enthalpy of saturated humid air at its wet-bulb temperature, and the enthalpy of saturated humid air at the water temperature. In order to represent the dynamic response of the temperature of the out-flowing water, energy storage in the liquid water hold-up and in the tower packaging is accounted for, while storage in the humid air is neglected. The tower fans are described by a simple model based on kinematic similarity, which states that the volume flow rate is proportional to the rotational speed, and the power consumption is proportional to the cube of the rotational speed. Ideal speed control is assumed. The heat transfer coefficient and the fan coefficients have been tuned with the tower manufacturer's design data, and the off-design performance of the has been validated against manufacturer's datasheets.

The outlet flows from the five towers mix together in a single pool, modelled as a well-mixed open tank. The pump and the piping to the condenser complete the loop. A small, fixed make-up flow rate compensates the water lost by evaporation.

The atmospheric dry bulb and wet bulb temperatures are exogenous known signals. The control signals are the pump and the fans rotational speeds, while the controlled variables are the tower outlet temperatures and the condenser steam-side pressure, i.e., the steam turbine discharge pressure. The overall plant model was calibrated in open-loop, based on steady-state design data.

4. CONVENTIONAL CONTROL SCHEME

The conventional control strategy for cooling systems like those addressed herein consists of regulating the tower outlet temperature so as to provide the condenser with the necessary cooling capacity to keep the turbine discharge pressure between convenient limits. The water pump is not controlled and always operates at full speed. The block diagram for this strategy is shown in Figure 2, that depicts only one of the five identical fan controls.

In the case under consideration, the fans could not operate within a certain speed range, in order to avoid the excitation of the first bending mode of the blades. To cope

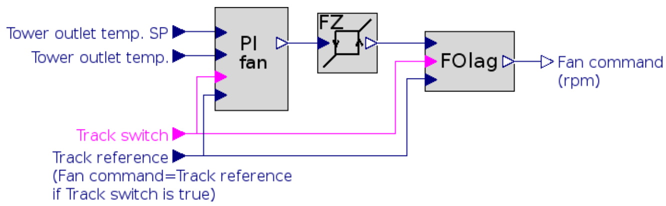

Fig. 2. Modelica scheme of the traditional control strategy.

with that, a hysteresis block and a unity gain first-order lag were introduced. The input-output characteristic of the hysteresis block, dubbed Forbidden Zone (FZ), is shown in Figure 3.

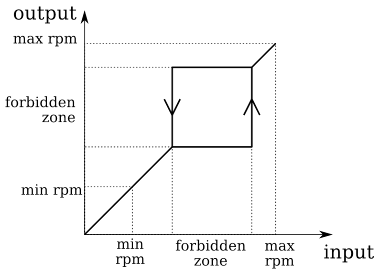

Fig. 3. I/O characteristic of the hysteretic block for the tower outlet temperature.

In Figure 2, cascaded to the PI and the hysteresis block FZ, a first-order lag (FOlag) is introduced. The role of this block is to shape the limit cycle that the closed-loop system will eventually enter when the fan rotational speed as requested by the PI falls into the forbidden range. The FOlag block has (obviously) a unity gain, and its time constant is related to the time that the control takes to make the fan speed command traverse the forbidden range when this is necessary. Both the PI and the first-order lag have a tracking mechanism, which can be used to have both block outputs follow a prescribed signal when the track switch is activated.

5. ENERGY-EFFICIENT CONTROL STRATEGY

The idea behind the proposed energy-efficient control strategy is that when the wet bulb temperature is low enough, the required cooling power can be transferred to the condenser with a reduced rate of colder water. Compared to the conventional approach, the reduction of power consumption of the pump (which is close to the maximum where the consumption curve is steeper) more than compensates the increased consumption of the fans, which are operating at low speed where the consumption curve is flatter, resulting in an overall electrical power saving. The system has now an extra degree of freedom (the pump speed), which can be exploited to minimized the combined consumption of fans and pump. Bear in mind that the nominal value of this consumption at full load for the plant considered in this paper is about 400 kW, so that a reduction of, say, 30% of that value corresponds to 0.7% of the rated plant power.

It has already been mentioned in the introduction that the main factor affecting the cooling towers performance is the wet bulb temperature of the cooling air, rather than its dry bulb temperature. This is also confirmed in this case by running different simulations where the dry bulb temperature only is changed, which give the same results. The plant load, which determines the heat duty of the condenser, is also obviously affecting the cooling tower performance. The new proposed control strategy can then be summarized as follows:

- for each combination of wet-bulb temperature and plant load, find the value of the towers water outlet temperature that minimizes the combined consumptions of fans and pump at steady state;
- schedule the set point of the conventional tower controllers according to the previously found map;
- use the variable speed pump to control the condenser pressure.

The control scheme is therefore composed of one PI controller with FZ for each tower fan, plus another one, also of the PI type, for the pump. The tower fans controls are identical to those of the traditional strategy, with the only addition of a set point scheduling mechanism. The pump controller is used for the regulation of the condenser pressure. The block diagram of the new strategy is shown in Figure 4.

In order to find the optimal values of the tower outlet temperature for each selected operating condition (wet-bulb temperature and plant load), a batch of simulations was performed in which the tower outlet temperatures set points, all equal, were changed at an extremely slow rate, so that the system performed a quasi-static transformation, where the dynamic terms played a negligible role. The total power consumption of fans and pump was computed, and the set point value yielding the minimum consumption was marked.

This procedure in principle produces a double-entry table relating the couple (wet bulb temperature, plant load) to a set point value for the tower outlet temperatures. In practice, the study revealed that the dependency on the plant load is negligible, as it is clearly visible in Figure 5, where the optimal tower outlet temperature set points are plotted against the wet bulb temperature for three different plant loads. The rightmost point of the curves is the design point, when both the fans and the pump are working at full speed. The range is limited to quite high loads ($\geq 90\%$) as these are the expected (and profitable) operating conditions for this kind of plant.

The three curves are almost perfectly coincident, meaning that the optimal choice of the tower outlet temperature set point does not depend on the plant load; furthermore, they can be very well approximated by a straight line, at least for wet-bulb temperatures greater than 5°C .

An important factor to consider is how much the cooling flow rate through the cooling towers can be reduced with respect to the design value, when this optimal strategy is applied. This can be assessed by starting from the steady-state energy balance on the condenser cooling side

$$Q_c = w_c c_p (T_{ic} - T_{oc}), \tag{1}$$

where Q_c is the condenser thermal load, w_c the cooling water mass flow rate, c_p the water specific heat capacity, T_{ic} and T_{oc} the inlet and outlet temperature of the condenser cooling water. At steady state, T_{ic} is approximately equal to the tower outlet temperature, which is controlled by the

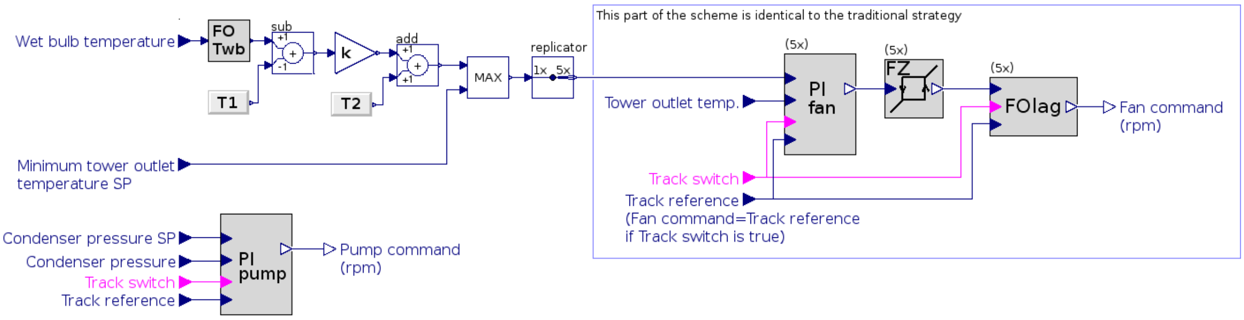

Fig. 4. Block diagram of the proposed control strategy.

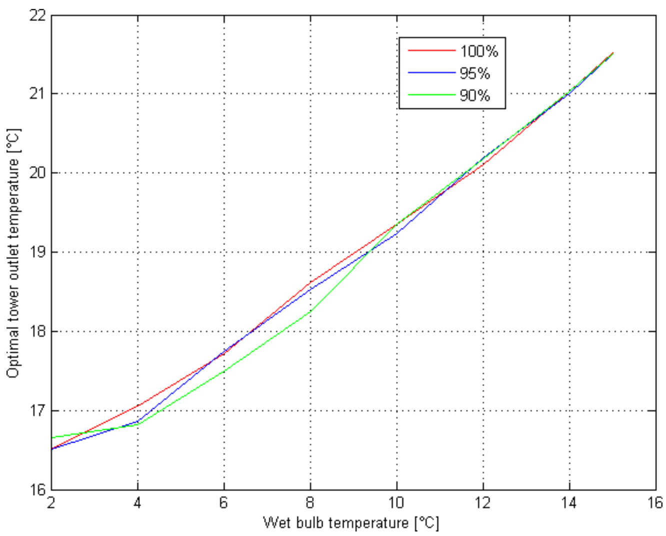

Fig. 5. Optimal tower outlet temperature set point for three plant loads.

fan speed, save for the small additional cooling given by the make-up cold water and by the additional evaporation on the pool surface. If the condenser pressure is controlled at a fixed value, the condenser internal temperature T_c is also fixed to the corresponding saturation temperature. Given the relatively high number of heat transfer units of the condenser, the relative changes in T_{oc} will be small, compared to the changes of Q_c, w_c and T_{ic}, as T_{oc} will always stay close to T_c. Hence, by neglecting them entirely, the linearized version of equation (1) reads:

$$\frac{\Delta Q_c}{Q_c} = \frac{\Delta w_c}{w_c} + \frac{\Delta T_{ic}}{T_{ic} - T_{oc}}. \tag{2}$$

This equation relates the relative changes of the cooling load and of the tower outlet temperature set point with the relative change of the pump flow rate. The cooling tower performance is normally not guaranteed below (say) 80% of the nominal cooling flow rate. This corresponds to a relative change $\Delta w_c/w_c$ of 20%. At nominal load, this reduction corresponds to a reduction in the set point of the tower outlet temperature by 20% of $(T_{ic} - T_{oc})$, i.e., 2 K in the considered case. If the plant load (and thus the cooling load) is reduced, then the margin of reduction of the tower outlet temperature set point will correspondingly be reduced.

Given the plant load, (2) allows to indirectly enforce the maximum reduction of the cooling flow simply by introducing a lower limit on the optimal tower outlet temperature set point. Moreover, one can reasonably expect that the tower cooling efficiency does not degrade too quickly below 80% of the nominal flow. As a consequence, it is very likely that when the wet bulb temperature is very low, it is possible to obtain an additional reduction of power consumption by lowering the tower cooling flow rate below 80%, even though the model used for the optimization will no longer be valid in that range. This can be done easily by further reducing the lower limit on the tower outlet temperature set point, or by introducing a piecewise linear wet bulb to tower outlet temperature set point curve, coinciding with that of Figure 5 for wet bulb temperatures above, say 19.5°C, and having a reduced slope for lower wet bulb temperatures.

Summing up, the scheduling of the tower outlet temperature set point can be done based on the wet bulb temperature only, and has the form

$$SP_{T_{to}} = max(K(T_{wb} - T_1) + T_2, T_{to,min}) \tag{3}$$

where $SP_{T_{to}}$ is the tower outlet temperature set point, T_{wb} the measured wet bulb temperature, $T_{to,min}$ is the lowest possible value of the set-point, according to the above-discussed considerations, and K, T_1, T_2 suitable parameters to match the curve in Fig. 5. The wet bulb temperature should be suitably low-pass filtered to reduce noise effects and to avoid problems in the case of outliers or transient measurement artefacts.

To end the section, a few words are in order on how to switch back and forth from the conventional to the new strategy, both for plant start-up, and in the case the former has to be maintained as a fall-back option. In both strategies, the fan control is in fact identical, save for the set point scheduling. Switching from one strategy to the other hence means just switching the set point source (constant or scheduled) with no intervention on the controllers. There will just be a short transient involving the fan drives, but owing to the large heat capacity of the pool, the rest of the plant will not be affected at all. As for the pump control, in the new strategy the pump PI is in automatic mode, while in the conventional one it is in tracking mode with its output forced at the maximum value. Hence, here too the switching is straightforward.

There is only one remark in this respect, concerning the switching from the new to the traditional strategy. If, prior to that switching, the pump speed command is well below the maximum, setting it to the maximum abruptly will feed the condenser with a suddenly increased flow of water that is colder than needed, since during the operation of the new strategy the pool has been accumulating water with a reduced and colder tower outflow. If this phenomenon produces an acceptable upset of the condenser operation, there is no action to take. Otherwise, it will be sufficient to schedule the change of the tracking signal with a ramp of duration comparable to the pool residence time.

6. SIMULATION RESULTS

Since the new strategy is expected to be active only when the plant is in normal operation at sufficiently high load, the worst-case expected transient is caused by a sudden variation of the wet bulb temperature (e.g., produced by a storm). To assess the capability of the new strategy to cope with that, the available meteorological data were searched for the fastest wet bulb temperature rate, and that value was for safety further increased by 20%, leading to a rate of $10°C/h$. Tests were then performed by applying ramp variations to the wet bulb air temperature with that rate, and by verifying the absence of undesired effects. In particular, those tests allow to further verify that the hysteresis block and the first-order lag, introduced in the fan control loops to cope with the forbidden speed range, are working correctly, especially as for error and control activity when traversing the forbidden range.

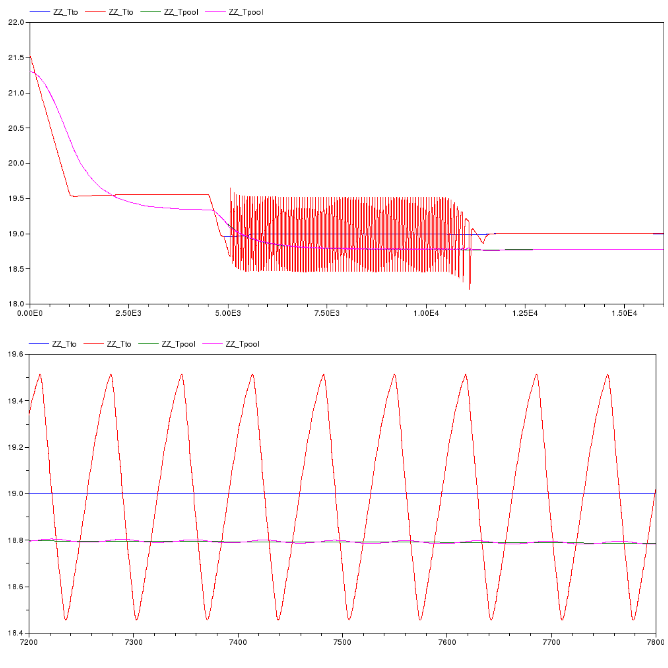

Fig. 6. Rapid variations of the wet bulb temperature: relevant temperature transients (top) and 10 minutes detail (bottom).

To start, Figure 6, top, shows the tower outlet temperature with and without the hysteresis block, the latter situation corresponding to a case in which the fans have no forbidden operating zone. The blue curve corresponds to the case without hysteresis and follows the set point almost perfectly. The red curve refers to the (real) case with forbidden speed range and hysteresis, and shows some oscillations when those effects come into play. However, said oscillations are completely smoothed out by the low-pass action of the pool, as shown by its temperature (green and magenta curve in the case without and with hysteresis, respectively) that does not reveal any significant variation. The detail shown in Figure 6, bottom, allows to better appreciate this fact.

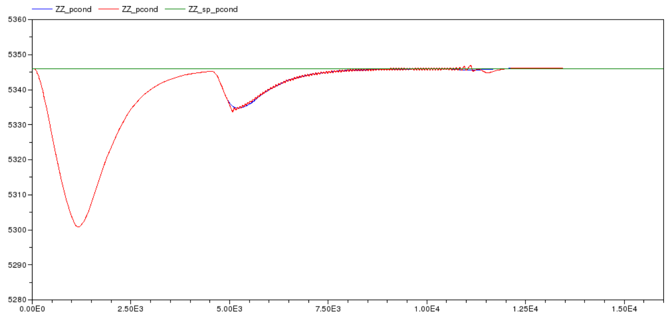

Fig. 7. Rapid variations of the wet bulb temperature: condenser pressure.

As a further check of the correct control operation, Figure 7 reports the condenser pressure, controlled by acting on the pump, in the case without and with hysteresis. The blue and the red curves correspond to those cases in the order just mentioned, while the green curve is the set point value. As can be seen, there is virtually no difference between the two cases, as the amplitude of the induced oscillations is significantly lower than that of the overall transients caused by the reaction of the control system to the applied wet bulb temperature variations. Notice also that the maximum deviation of the controlled pressure from its set point is extremely low, although the control activity (further investigated later on) is acceptable.

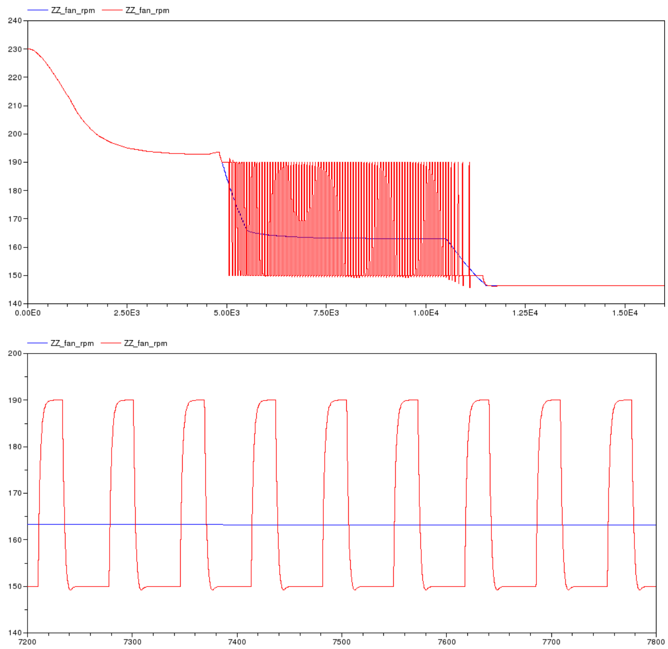

Fig. 8. Rapid variations of the wet bulb temperature: fan speed transients (top) and 10 minutes detail (bottom).

Coming to the control activity, Figure 8, top, reports the fan speed during the same transients of the previous figures. Here too the blue plot corresponds to the idealised case without hysteresis, and the red plot to the real one. As can be seen, the proposed control makes the fan traverse the forbidden rpm range in a PWM-like fashion, resulting in an effect on the plant that is very similar to what would be obtained in the absence of the FZ. Figure 8, bottom, reports a 10-minutes detail of the overall fan speed behaviour, when the command is within the FZ.

As can be seen, that zone is traversed rapidly and safely. Also the frequency of the fan switching is acceptable, and if necessary, could be modified by acting on the time constant of block FOlag. A larger time constant would reduce the switching frequency at a cost of larger temperature oscillations, that are however smoothed out by the pool. The proposed parameters are a good initial compromise, but some fine tuning on the plant can improve the results as usual.

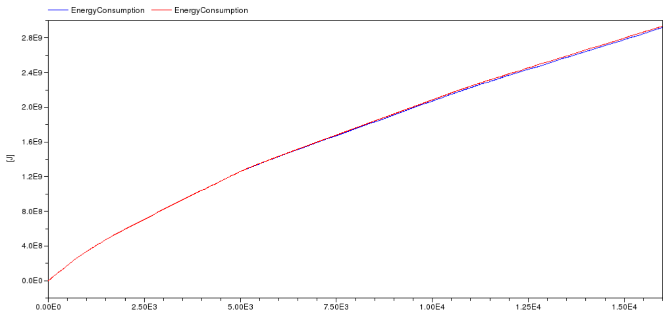

Fig. 9. Rapid variations of the wet bulb temperature: consumed energy.

Finally, it was checked that the real system (with hysteresis) consumes more or less the same energy of the idealised one, in a view to verifying that the proposed control and actuation strategy cannot jeopardise the envisaged energy optimisation. In fact, as shown by the energy consumption curves of Figure 9, where again the blue is the ideal and the red the real one, the effect of the limit cycle induced by the hysteresis block is hardly noticeable, leading to a difference in the total consumed energy of less than 1%.

In the plant that was the subject of this study, based on historical atmospheric data, energy savings worth about 20000 EUR/year were estimated. While this figure is not dramatic in relative terms, its absolute magnitude is more than enough to pay for the improvements in the control system, i.e., the installation of a variable speed drive for the pump, and a few modifications on the plant DCS.

7. CONCLUSIONS AND FUTURE WORK

This paper presents an energy-efficient control strategy for the operation of forced draft cooling towers used in small steam power plants, aimed at minimizing the combined energy consumption of the fans and of the circulation pump. The analysis supported by a detailed physical model of the system eventually led to a simple control strategy, that can easily be implemented in a conventional industrial control system. It is also shown how is it possible to implement the controller so that the fall-back to the conventional control strategy is possible.

The proposed controller has been implemented in an 18 MWe biomass-fired steam power plant, which is currently in operation. The control system performs as expected, though it has not yet been possible to carry out a comprehensive assessment of the actual energy savings.

Future developments of this work might evolve along two research lines. On one hand, it would be interesting to verify if the energy-optimal scheduling always turns out to be independent of the load and a linear function of the wet-bulb air temperature also for other plants of the same kind. On the other hand, it might be interesting to explore the opportunity of using model-free control strategies, that minimize the actual power consumption without the need of a sophisticated model to calibrate the scheduling curve, by using, e.g., extremum seeking control techniques.

REFERENCES

Ahn, B. and Mitchelly, J. (2001). Optimal control development for chilled water plants using a quadratic representation. *Energy and Buildings*, 33(4), 371–378.

Braun, J. and Diderrich, G. (1990). Near-optimal control of cooling towers for chilled-water systems. *ASHRAE Transactions*, 96(2), 806–813.

Briley, G. (2003). Increasing operating efficiency. *ASHRAE Journal*, 45(5), 73–???

Casella, F. and Leva, A. (2006). Modelling of thermo-hydraulic power generation processes using Modelica. *Mathematical and Computer Modelling of Dynamical Systems*, 12(1), 19–33.

Casella, F. and Leva, A. (2005). Object-oriented modelling & simulation of power plants with modelica. In *Proceedings 44th IEEE Conference on Decision and Control and European Control Conference 2005*, 7597–7602. IEEE, EUCA, Seville, Spain.

Crowther, H. and Furlong, J. (2004). Optimizing chillers and towers. *ASHRAE Journal*, 66(7), 34–44.

European Commission (2001). Reference document on the application of best available techniques to industrial cooling systems, technical report.

Kintner-Meyer, M. and Emery, A. (1995). Cost-optimal design for cooling towers. *ASHRAE Journal*, 37(4), 46–55.

Koeppel, E., Mitchell, J., Klein, S., and Flake, B. (1995). Optimal supervisory control of an absorption chiller system. *HVAC and R Research*, 1(4), 325–342.

Lu, L., Cai, W., Soh, Y., Xie, L., and Li, S. (2004). HVAC system optimization – condenser water loop. *Energy Conversion and Management*, 45(4), 613–630.

Ma, Z., Wang, S., and Xiao, F. (2009). Online performance evaluation of alternative control strategies for building cooling water systems prior to *in situ* implementation. *Applied Energy*, 86(5), 712–721.

Ma, Z., Wang, S., Xu, X., and Xiao, F. (2008). Supervisory control strategy for building cooling water systems for practical and real time applications. *Energy Conversion and Management*, 49(8), 2324–2336.

Schwedler, M. (1998). Take it to the limit ...or just halfway? *ASHRAE Journal*, 40(7), 32–39.

Sun, J. and Reddy, A. (2005). Optimal control of building HVAC systems using complete simulation-based sequential quadratic programming (csb-sqp). *Building and Environment*, 40(5), 657–669.

Yao, Y., Lian, Z., Hou, Z., and Zhou, X. (2004). Optimal operation of a large cooling system based on an empirical model. *Applied Thermal Engineering*, 24(16), 2303–2321.