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Cooling Water System Modelling for Control and Energy Optimisation Purposes

Cornelius J. Muller *,** **Ian K. Craig** **

* *Sasol Solvents RSA, Sasolburg, South Africa (e-mail: nelis.muller@sasol.com),*
** *Department of Electrical, Electronic, and Computer Engineering, University of Pretoria, Pretoria, South Africa (e-mail: nelismuller@tuks.co.za, icraig@postino.up.ac.za)*

Abstract: Utilities account for a significant portion of the fixed cost associated with running a petrochemical plant, however, utility optimisation has not received much focus until recently. This change was driven by rising energy prices, stricter environmental policies, and the threat of climate change. Several utilities are used including steam, compressed air, nitrogen, cooling water, refrigeration water, tempered water, electricity, and fuel gas. Generation and preparation of all of these utilities require energy and therefore merits optimisation to reduce waste. This paper describes the modelling of a dual circuit cooling water system containing a combination of discrete and continuous energy components for control and energy optimisation purposes.

Keywords: Modelling, verification, optimisation, energy, hybrid systems.

1. INTRODUCTION

With stricter environmental policies, rising energy costs, a struggling global economy, and the threat of climate change, there has been a revival in the focus on efficiency improvement in the process industries (Craig et al. (2011)). Historically, high product margins far outweighed the price of additional energy input. Currently, the margin between product recovery and energy cost is narrowing which requires a different approach to energy management.

The petrochemical industry accounts for a significant portion of energy usage and greenhouse gas emission globally. Therefore, a seemingly insignificant efficiency improvement in this industry can have a substantial impact on global consumption. From a financial point of view, energy costs in refineries in the U.S. are approximately between 50% and 60% of total fixed cost and at 30% to 40% for chemical plants (Lipták (2007)). In South Africa, the petrochemical industry was responsible for more than 20% of electricity consumption in manufacturing between 1993 and 2006 (Inglesi-Lotz and Blignaut (2011)). Therefore, considerable savings can be realised by reducing energy consumption.

In some cases, plants are designed with energy efficiency in mind. This can have large long-term benefits though it comes at additional capital cost and more complex process dynamics (for instance by using process-to-process heat exchangers for heat integration). The norm though is the opposite and the opportunities for optimisation after commissioning can be attractive as described e.g. in Muller et al. (2011) and Ricker et al. (2012).

Energy is supplied to (or removed from) a plant mostly through utilities such as steam, tempered water, compressed air, electricity, fuel gas, cooling water, etc. and a reduction in the consumption of these utilities results in a direct energy saving. The less frequently considered improvement area is the actual supply or generation side of these utilities. The amount of energy lost in the generation and transmission of utilities is significant (Saidur et al. (2010), Bhatt (2000)). Furthermore, poor focus on control of these utilities results in running unnecessarily large buffer capacities which typically results in additional waste through venting to get rid of over-generation at times of stable operation. Therefore, a control and optimisation scheme focussing on the optimal generation and supply of utilities has definite value-add potential. This paper describes the modelling of a dual circuit cooling water system for control and optimisation purposes which is a prime example of the opportunities that exist with utilities.

2. PROCESS DESCRIPTION

Energy is required to move material through a plant and to change its temperature. Apart from the actual thermal energy consumed/removed through equipment such as heat exchangers, electrical energy is required to transport the utility media through a plant by using pumps, fans, compressors, blowers, etc. The energy consumption of such equipment can be modelled through duty equations or power curves.

A cooling water system is a prime example of a utility system using several pumps, fans, and heat exchangers. There are several types of cooling towers including natural draft, mechanical draft, and evaporative condensers. Mechanical draft towers are further classified as induced or forced draft, either of which can be cross-flow or counterflow (Lipták (2006)).

The cooling water system modelled in this paper is shown in Figure 1, and is an example of a dual circuit cooling water system with induced draft counter-flow cooling towers. There are two water circuits. The first is the tempered water (TW) circuit which is a closed, treated water loop that is pumped by a bank of pumps through the plant heat exchanger network to cool the process and then through another bank of heat exchangers where it transfers the energy to the second circuit, the cooling water (CW) circuit. The cooling water is then pumped by another a bank of pumps to a bank of cooling towers (CTs) where it is cooled mainly through partial evaporation as the water interacts with a counter-current induced air draft. This latent heat transfer accounts for about 80% of heat transfer with the balance occurring through sensible heat transfer between die water and air (Green (1997)). The tempered water circuit has a temperature control valve that bypasses the CW heat exchanger bank if too much cooling is provided (such as during plant load reduction or a sudden rain spell affecting the heat duty). The pumps on the cooling water side are each equipped with a discharge pressure control valve which will throttle back if the discharge pressure of a pump drops too low. The CW circuit also has side stream filters and a water make-up line which are not shown in Figure 1.

The typical disturbances associated with the system include ambient temperature fluctuations, plant load changes, and equipment failures. The available handles for control on the system are the running signals on the pumps and fans, the discharge valves on the cooling water pumps, and the temperature valve on the tempered water circuit (discussed in more detail in Section 4). There are no variable speed drives (VSDs) on this specific process. The scenario for VSDs will be handled in future work.

This system presents a number of energy optimisation opportunities:

- Each pump can be individually optimised based on the required flow rate through it.
- Each pump bank can be optimised based on the amount of pumps required to run to ensure that the total circuit flow requirement is met.
- Each circuit can be optimised based on the required flow and temperature of the water.
- The bank of cooing towers can be optimised with regard to the number of fans running and the temperature of the water exiting the towers.

The amount of optimisation is restricted by the following system constraints:

- The maximum flow through a TW pump (J-101 to J-105) is 2500 t/h.
- The maximum flow through a CW pump (J-201 to J-205) is 2750 t/h.
- The TW supply temperature, $T_{TWS} \geq 26\,^{\circ}\mathrm{C}$.
- The TW differential temperature, $(T_{TWR} - T_{TWS}) \leq 10\,^{\circ}\mathrm{C}$.

The bigger the temperature difference between the supply and return streams to the heat exchangers, the better the efficiency of heat transfer. Therefore, over-cooling of either circuit is not desirable. If downstream processes are temperature controlled, increasing the temperature of the tempered water will cause higher tempered water flow which will counter the initial intent of reduced energy consumption through reduced cooling tower fan speed and cooling water flow. Therefore, the optimal balance between flow and temperature must be determined to meet the cooling requirements of the plant in an optimal way.

3. HYBRID SYSTEMS

The cooling water system described above is a typical example of a hybrid energy system (sometimes referred to as mixed-integer, mixed logical dynamical, or switching systems) where there are continuous and discrete elements. The continuous elements in this case include the control valves whereas the discrete elements include the pump and fan running statuses. The presence of discrete variables complicates modelling and optimisation.

Traditionally, hybrid systems are dealt with in two distinct layers. The bottom layer is concerned with the continuous process and the top layer with the discrete process. In some processes, however, the distinction cannot easily be made due to the level of integration. Therefore, the need arose for a systematic way of modelling and designing controllers for hybrid systems that combines the continuous and discrete functions. Bemporad and Morari (1999) suggest the use of mixed integer quadratic programming (MIQP) with the process described in terms of linear inequalities which are obtained though manipulation of combinational logic. These inequalities, combined with the continuous process model is used to formulate an MPC solution.

Zhang and Xia (2010) describe different optimal control techniques for dealing with systems with switching logic (such as on/off pumps) and continuous components such as equipment fitted with VSDs.

4. SIMULATION

A simulation of the system described in Section 2 was created in MATLAB and validated against plant data. The model provides a framework for future control and energy optimisation studies. The simulation provides a baseline for how the plant is normally run with all (or most) pumps and fans running continuously, except when a failure occurs. Control is performed with a temperature controller on the TW side and pressure controllers on the discharges of the CW pumps. All of these are constraint handling controllers that are not active during normal operation. The temperature controller will only open the heat exchanger bypass valve in the event of too much cooling (a too low TW supply temperature). The discharge pressure controllers will only throttle back on the discharge valves in the event of the discharge pressure of a pump dropping (i.e. the pump is in danger of running over capacity in terms of flow).

The modelling is performed under the following assumptions:

- Pumps on a bank are identical and balanced.
- The cooling towers are identical and balanced.
- The plant heat exchanger network has a constant flow coefficient (i.e. a constant system curve).
- No switching of heat exchangers occurs (all five heat exchangers are in use all the time).
- Side-stream filters and the dosing system are omitted from the model.

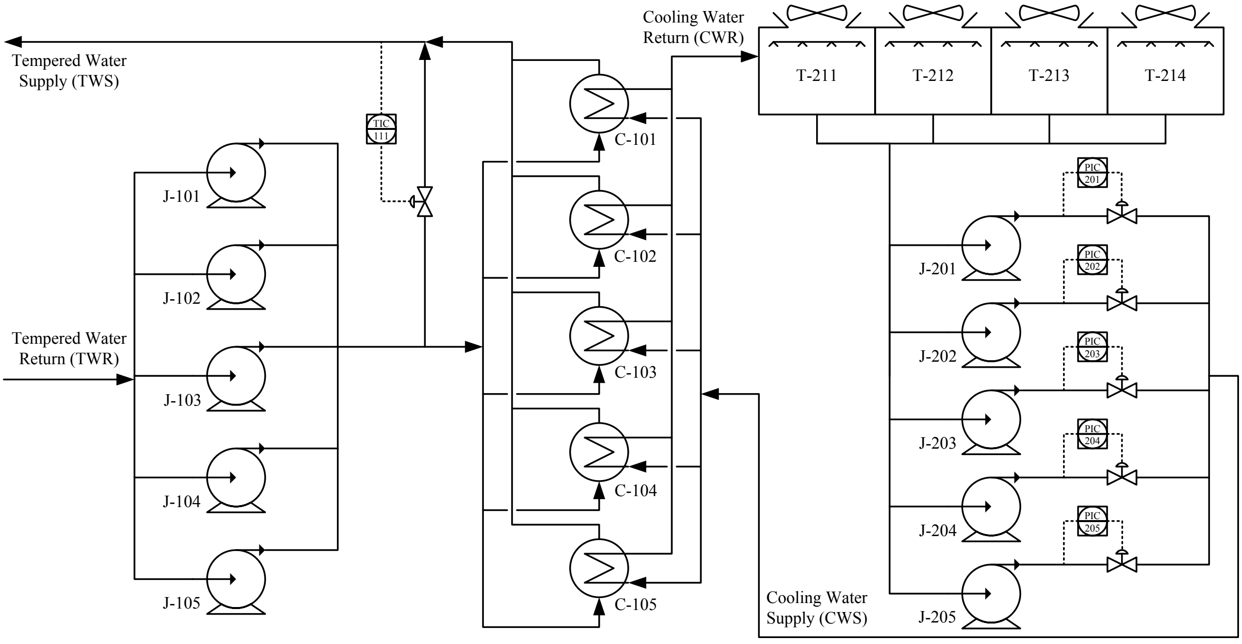

Fig. 1. Dual circuit induced draft counter-flow cooling water system.

- The suction pressure of the TW pumps is fixed at 230 kPa-g.
- The suction pressure of the CW pumps is fixed at 20 kPa-g (i.e. a 2m water level in the cooling towers).
- Heat addition by the pumps is negligible.

The model inputs are

- the pump running signals, $u_i^{TW}(t)$ and $u_j^{CW}(t)$,
- the CT fan running signals, $u_k^{CT}(t)$, and
- the temperature control valve and discharge pressure valve openings

with $i = 1$ to n_{TW}, $j = 1$ to n_{CW}, and $k = 1$ to n_{CT} where n_{TW} and n_{CW} represent the numbers of TW and CW pumps and n_{CT} is the number of CT fans. The model disturbance variables include

- the plant duty, $Q_P(t)$ (MJ/h),
- the ambient air wet-bulb temperature, $T_{wb}(t)$ (°C),
- the make-up water flow to the cooling towers, $F_{mu}(t)$ (t/h), and
- the availability of the pumps, fans, and heat exchangers.

The controlled outputs are

- the TW supply temperature, T_{TWS},
- the TW differential temperature (T_{TWR} - T_{TWS}),
- the electricity consumption of the system, and
- the energy cost of the system.

The first two outputs are for constraint handling whereas the last two can be used for optimisation.

4.1 Pump Calculations

The pump discharge pressures are determined by the pump performance curves and the system curves. When no throttling element is used (such as a discharge throttling valve), the operating point is the intercept between the pump curve and the system curve. When more than one pump runs in parallel, the flow is distributed between pumps and the higher combined discharge pressure results in a higher total flow. Figure 2 illustrates the operation of a single pump versus two identical pumps in parallel (Daugherty and Franzini (1977)). When running a single pump, the operating point is at A. When running two pumps in parallel, the combined operating point moves to B with the operating point per pump illustrated by C. Therefore, the more pumps are run in parallel, the less the progressive increase in flow becomes.

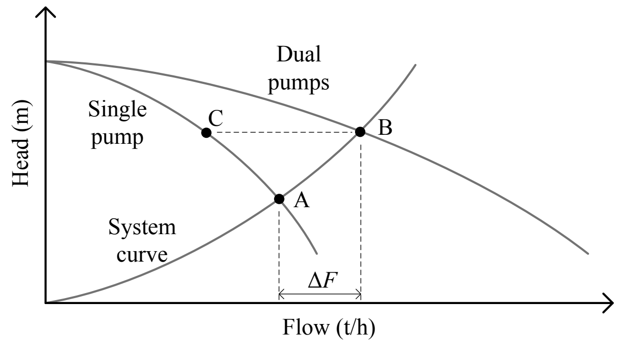

Fig. 2. Pumps in parallel.

The pump performance curves were captured in lookup tables for the modelling in Simulink. Given a mass flow rate, F (t/h), the table generates the corresponding head, h (m). The head is translated to discharge pressure, P_d (kPa-g), with $P_d = h.g.\rho + P_s$ where P_s is the suction pressure (kPa-g), $g = 9.81$ m/s^2 is the gravitational constant, and $\rho = 1.0\,\text{kg}/\ell$ is the specific gravity of water.

The pump running signals indicate the number of pumps in operation. In the following sections, the system is reduced to single equipment for simplicity. For the complete model, however, the calculations are done per device and the results combined for the system outputs. The same applies for the cooling tower fan sunning signals (i.e. a cooling tower is viewed as out of operation if its fan is not running).

4.2 Flow Calculations

The flow in the TW circuit is determined by the TW pumps' common discharge pressure and the flow coefficients of the plant heat exchanger network, the CW heat exchangers' TW side, and the temperature control valve (with valve opening). The flow in the CW circuit is determined by the CW pumps' discharge pressure and the flow coefficients of the CW heat exchangers' CW side, the cooling towers, and the discharge pressure control valves (with valve openings). Figure 3 shows a simplified system diagram indicating the key variables. The total flow in the TW circuit is described by

$$\begin{aligned} F_{TW} &= F_{TV} + F_{TWI} \\ &= C_{ETW}.\sqrt{\Delta P_1} + C_{TV}.OP_{TV}.\sqrt{\Delta P_1} \end{aligned} \tag{1}$$

and

$$F_{TW} = C_G.\sqrt{\Delta P_2} \tag{2}$$

where F_{TV} is the mass flow through the temperature valve, F_{TWI} is the intermediate mass flow thought the TW side of the heat exchanger bank, C_{TV}, C_{ETW}, and C_G are the flow coefficients of the temperature valve, the TW side of the heat exchangers, and the plant, OP_{TV} is the valve opening of the temperature valve, and $\Delta P_1 = P_2 - P_3$ and $\Delta P_2 = P_3 - P_1$ are differential pressures. The differential pressure over the pumps is $\Delta P_3 = P_2 - P_1$ and is determined from the pump curve. Then

$$\Delta P_2 = \Delta P_3 - \Delta P_1 \tag{3}$$

and by setting (1) = (2) and substituting for (3) results in

$$\Delta P_1 = \frac{\Delta P_3}{\left(\frac{C_{ETW} + C_{TV}.OP_{TV}}{C_G}\right)^2 + 1} \tag{4}$$

which allows for the calculation of F_{TWI} and F_{TV}. On the CW loop, the flow F_{CW} is calculated in a similar fashion. Here, the combined flow coefficient of the cooling towers is the sum of the flow coefficients of the towers that are in operation.

4.3 Duty Calculations

The plant duty, Q_P, is the energy to be removed by the cooling system. This energy is transferred from the plant to the TW stream through the plant heat exchanger network. It is then transferred from the TW to the CW through the CW heat exchanger bank (shown in Figure 1). The energy is then expelled from the CW stream via the cooling towers. The heat transfer mechanism in the cases for the plant heat exchanger network and the CW heat exchangers

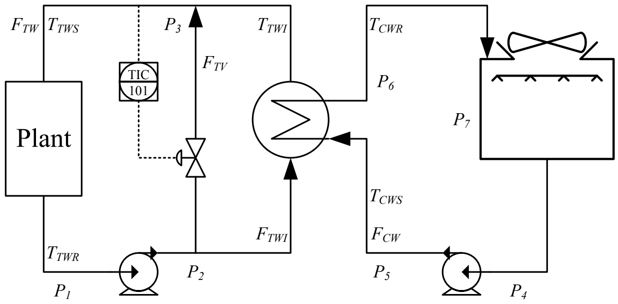

Fig. 3. Heat exchanger streams.

(assuming no phase change) is sensible heat transfer and is described by

$$Q = F.C_p.\Delta T \tag{5}$$

where F is the mass flow of the fluid (t/h), C_p is the specific heat of the fluid (kJ/kg.°C), and ΔT is the differential temperature between the inlet and the outlet of the heat exchanger (°C). This is valid for both the process and utility sides of the exchanger. In this case, both sides are water streams with $C_p = 4.18$.

The heat exchanger duty is determined by the heat transfer coefficient, U (kg/m^2.°C), the heat exchange area, A (m^2), and the log mean temperature difference between the process and utility streams, ΔT_{lm} (°C). With reference to Figure 3, the heat exchanger duty is described as

$$Q = U.A.\frac{(T_{TWR} - T_{CWS}) - (T_{TWI} - T_{CWR})}{\ln\left(\frac{T_{TWR} - T_{CWS}}{T_{TWI} - T_{CWR}}\right)} \tag{6}$$

where T_{CWS} and T_{CWR} are the CW supply and return temperatures, T_{TWR} is the TW return temperature, and T_{TWI} is the TW intermediate temperature (the heat exchanger outlet temperature on the TW side).

By substituting (5) (on the TW side) into (6), T_{TWI} is calculated as

$$T_{TWI} = \frac{T_{CWS} - \frac{Q}{F_{TWI}.C_p} - e^k.T_{CWR}}{1 - e^k} \tag{7}$$

with

$$k = \left(\frac{Q}{F_{TWI}.C_p} - T_{CWS} + T_{CWR}\right).\frac{U.A}{Q}. \tag{8}$$

The TW supply temperature is then calculated as

$$T_{TWS} = \frac{T_{TWI}.F_{TWI} + T_{TWR}.F_{TV}}{F_{TW}} \tag{9}$$

where F_{TWI}, F_{TV}, and F_{TW} are discussed in Section 4.2. Note that when the TV is closed, $F_{TW} = F_{TWI}$ and therefore $T_{TWS} = T_{TWI}$.

On the CW circuit, a simplified model was used to determine the heat exchange in the cooling towers. This required the wet-bulb temperature (T_{wb}) of the air. The ambient temperature, T_a is measured and used together with the average relative humidity (RH) of the area to calculate the approximate T_{wb} as follows (Stull (2011)):

$$\begin{aligned} T_{wb} = {} & T_a.\tan^{-1}(0.151977.(RH+8.313659)^{0.5}) \\ & + \tan^{-1}(T_a+RH) - \tan^{-1}(RH-1.676331) \\ & + 0.00391838.(RH^{1.5}).\tan^{-1}(0.023101.RH) \\ & - 4.686035 \end{aligned} \tag{10}$$

The difference between the achievable T_{CWS} and T_{wb} is called the approach and is dependent on the tower design. The amount of water being evaporated in the cooling towers (t/h) can be estimated as

$$F_e = x.F_{CW}.(T_{CWR} - T_{CWS}) \tag{11}$$

and

$$F_e = F_{mu} - F_b - F_d \tag{12}$$

where F_{mu} is the make-up water added to the system on CT level control, F_b is the blow-down to prevent solids build-up in the system, and F_d is the drift loss through splashing and entrainment. Combining the recommendations from Green (1997) and Lipták (2006) resulted in $x = 0.0015$ (%/°C), $F_d = 0.001.F_{CW}$, $F_b = F_e/2$ (assuming three cycles of concentration), and an approach of 4.2 °C. Equation (12) can then be used to calculate the approximate F_e though it does not contain any reference to the ambient conditions in its current form. Therefore, it is multiplied by the feedback term, $(T_{CWS}/(T_{wb}+4.2))$, which serves to compensate for ambient condition changes and also takes the approach into account. For example, if T_{wb} drops due to a drop in RH, it will result in a higher F_e which means more cooling. Equation (11) can now be used to calculate the new value of T_{CWS}. The portion of the duty of the CTs due to the partial evaporation is calculated as

$$Q_e = F_e.\lambda \tag{13}$$

where $\lambda = 2260$ kJ/kg is the approximate heat of vaporisation for water.

4.4 Energy Model

An energy consumption model of the system can be formulated based on the running statuses and power consumptions of the equipment. A simplified model can be constructed by assuming a constant power consumption equal to the rated power per piece of equipment when the equipment is running and a zero consumption when it is off (Zhang and Xia (2010)). The running signal of the ith component can be denoted by

$$u_i(t) = \begin{cases} 1 & \text{for component ON} \\ 0 & \text{for component OFF} \end{cases} \tag{14}$$

and the power by

$$W_i(t) = u_i(t).W_i^{\max} \tag{15}$$

where $W_i^{\max}$ denotes the rated power of the ith component. Then the total power consumption at time t is

$$W^T(t) = \sum_{i=1}^{N} W_i(t). \tag{16}$$

An objective function for energy optimisation can now be formulated for time period $[t_0, t_f]$ as

$$J_W = \int_{t_0}^{t_f} W^T(t)dt \tag{17}$$

or for energy cost

$$J_C = \int_{t_0}^{t_f} W^T(t)p(t)dt \tag{18}$$

where $p(t)$ is the electricity price at time t. If no TOU tariff is applicable, $p(t)$ reduces to a constant.

5. RESULTS AND CONCLUSION

The model was verified against plant data for a period of 6 days (144 hours) using a one minute sampling interval which yielded promising results. Some further model tuning may be required and unmeasured disturbances such as heat exchanger isolation, relative humidity changes, rain events, etc. are not included at this stage. Figure 4 shows the data inputs to the model for $Q_P(t)$ and $T_{wb}(t)$ (calculated using (10) with the measured amient temperature and an average humidity of 35%). Other inputs include $\mathbf{u}_{TW}(t)$, $\mathbf{u}_{CW}(t)$, $\mathbf{u}_{CT}(t)$, and $F_{mu}(t)$. Figure 5 shows the model response (solid line) versus the plant data (dotted line) for $T_{TWR}(t)$ and $T_{TWS}(t)$. Figure 6 shows $Q_e(t)$ (black line) versus $Q_P(t)$ (grey line) and the ratio of $Q_e(t)$ over $Q_P(t)$ which indicates that the evaporative duty is indeed approximately 80% of total duty as stated in Section 2. The correlation coefficients between the plant and model outputs were used as a simple measure of similarity between the data sets (i.e. the validity of the model). The correlation coefficients for $T_{TWR}(t)$, $T_{TWS}(t)$, and $Q_e(t)$ are shown in Table 1 which indicate an adequate accuracy for the purposes of this simplified model.

Table 1. Model Correlation Coefficients

Variable	Corr. Coeff.
$T_{TWR}(t)$	0.8264
$T_{TWS}(t)$	0.7630
$Q_e(t)$	0.9365

Future work could include:

- Further improvements on model accuracy and scope.
- Applying MPC to the system using only the available valves (i.e. conventional MPC using only continuous handles).

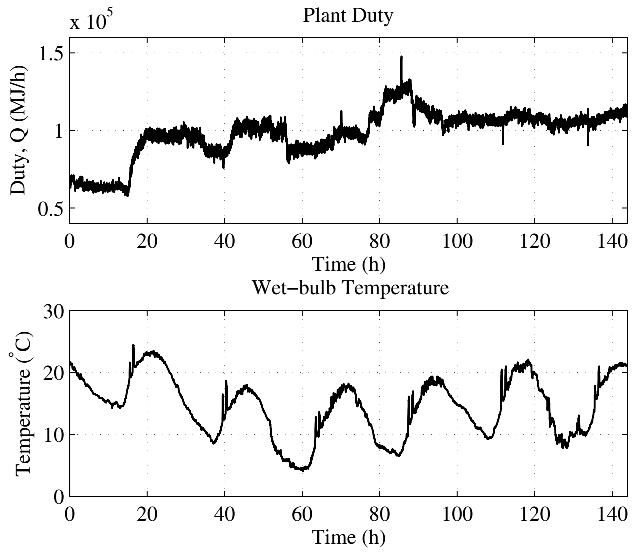

Fig. 4. Model disturbance inputs.

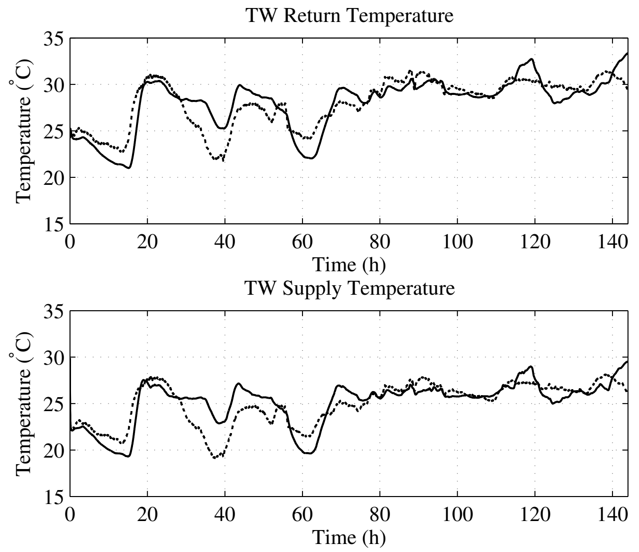

Fig. 5. Model TW results.

- Considering the use of RTO on top of MPC in the traditional hybrid control fashion (two distinct layers).
- Applying mixed-integer MPC that includes continuous and discrete elements into the same optimisation problem.
- Considering the case where one or more of the equipment is fitted with a VSD.

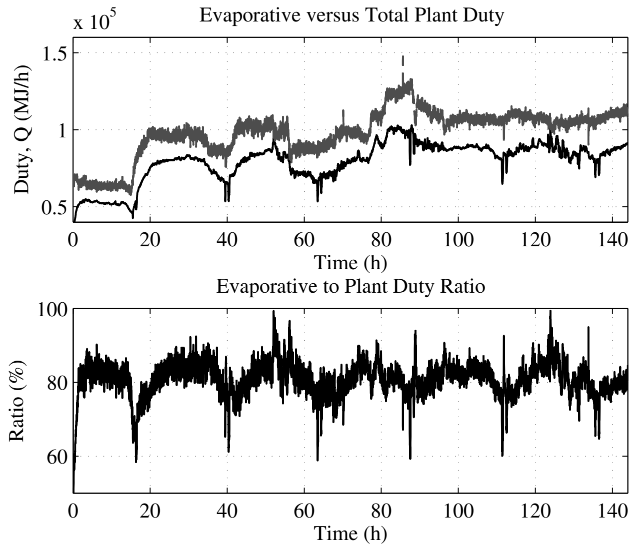

Fig. 6. Model duty results.

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