SIMULATION OF A CENTRALIZED COOLING PLANT UNDER DIFFERENT CONTROL STRATEGIES

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ABSTRACT

The simulated cooling plant equips an office building whose maximum cooling demand is about 5 MW. To meet this load, the cooling plant uses five cooling towers, four twin-screw chillers and four encapsulated ice storage tanks.

The simulation of the cooling plant is carried out with the software TRNSYS. To perform the simulation, the system components are first modeled and identified on the basis of the information given by the equipment manufacturers and the installers.

Once the parameter identification of the components is done, TRNSYS is used to simulate the cooling plant in each operating mode. This paper focuses on two operating modes : the pure discharging and the direct production with possible discharging. For these modes, a control strategy was developed during the cooling plant design phase. This paper discusses the impact on the plant performance of changes brought to the designed control strategies.

INTRODUCTION

A large variety of centralized cooling plants can be found in air-conditioned buildings: the plant can be equipped with thermal storage tanks, the chillers can be air-cooled or water-cooled, they can be equipped with reciprocating, screw, centrifugal compressors, etc.

Likewise, the HVAC designer can develop a large variety of control strategies: the set point temperatures can differ, the flow rates can be fixed or variable, etc.

The cooling plant analyzed in this paper is equipping the building of the Ministry Council of the European Union in Brussels. This building is built on a 40 000 m^2 ground area, with about 54 000 m^2 of useful area, distributed on 10 floors. The cooling plant is equipped with water-cooled twin-screw chillers. The technical solution adopted in order to meet the building cooling loads is to use an ice storage system operating in parallel with the chillers. This allows one to reduce the number and the size of chillers, as well as to take advantage of the reduced cost of electricity during the night when the storage tanks are charged (ASHRAE, 1995).

On the basis of this cooling plant configuration, a control strategy was developed during the design phase. The simulation software, TRNSYS (Klein et al., 1994) is used to give a correct forecast of the cooling capacity available and of the cooling energy stored in the ice storage tanks, but also to evaluate the impact of changes in the designed control strategy on the thermal performance of the cooling plant.

COOLING PLANT DESCRIPTION

A simplified view of the centralized cooling plant is shown in Figure 1.

The plant operates thanks to the adequate set-up of the following main components :

- five counterflow plate heat exchangers in parallel arrangement;

- four horizontal encapsulated ice storage tanks in parallel-series arrangement;

- an equilibrium tank that equilibrates the pressures between the chillers evaporators, the ice storage tanks and the heat exchangers.;

- four R22 twin-screw chillers in parallel arrangement. Capacity control is achieved thanks to a slide valve which can modulate the chiller capacity from 100 % down to about 10 % of full-load. Each chiller has a fixed-speed pump on the evaporator and condenser sides;

- five counterflow cooling towers in a parallel arrangement. Each tower is equipped with a two-speed centrifugal fan.

All these components are inter-connected by three networks:

- the chilled water distribution network to buildings. It is characterized by three fixed-speed pumps (called secondary pumps) and a bypass which controls, thanks to modulating valves, the water flow supplied to the building;

- the glycol water circuit on evaporator and ice storage tank side. It is characterized by three fixed-speed pumps (called primary pumps) and two modulating valves which control the flow in the ice storage tanks and in the equilibrium tank;

- the warm water circuit on condenser side. It is constituted by a reservoir where the make-up water is added, and a bypass which avoids, thanks to a modulating valve, a too low temperature at the condenser inlet.

The cooling plant is permanently monitored by a BEMS (Building and Energy Management System). The BEMS provides the operating status of the valves, pumps, fans and chillers, the pressure drop through the building as well as the temperatures at the inlet and outlet of each component. However, no flow meter is installed in the chilling plant. The only information about flow rate are the chilled water flow rates given by two enthalpy flow meters installed on the chilled water distribution of the buildings. But this information is not handled by the BEMS.

DESIGNED CONTROL STRATEGY

The first step in the control of the chilling plant is the definition of the plant operating mode. Three main operating modes can be selected :

- **direct production** : priority is given here to chillers. If the cooling demand is higher than the maximum cooling capacity of all the chillers, the additional cooling power required is provided by the ice storage tanks.

- **discharge** : priority is given here to the discharge of the ice storage tanks. If the cooling demand is higher than the maximum cooling capacity of the ice storage tanks, the additional cooling power required is provided by the screw chillers.

- **storage** : the chillers are used to charge the ice storage tanks. A characteristic of this installation is that the building cooling loads are never equal to zero. Therefore, during the charge of the ice storage tanks, the chillers have also to supply the building with a (low) cooling capacity.

The goal of the regulation is to act on the control variables in order to meet the set points while respecting some constraints.

The control variables are distributed among the three fluid networks. The warm water network is characterized by the number of operating cooling towers (i.e. the towers whose valve is open), the fan speed of the operating towers (high, low or OFF) and the opening ratio of the bypass valve (see Figure 1). The glycol water network is characterized by the number of operating chillers, the slide valve position of the operating chillers, the opening ratio of the valve VR1 at the equilibrium tank outlet (see Figure 1), the opening ratio of the valve VR2 at the ice storage tank outlet (see Figure 1) as well as the number of operating primary pumps.

It should be noted that only one or two primary or secondary pumps can be utilized; the third one is kept in reserve.

Two set point temperatures appear in the control strategy : the set point on the cold water temperature at the building inlet and the set point on the glycol water temperature at the evaporator outlet. The respect of the first set point is the priority of the cooling plant control strategy. This set point is equal to 7°C. The value of the second set point is function of the plant operating mode: it is equal to -7°C in storage mode whereas it is equal to 4.5°C in other operating modes. This set point is the same for all the chillers.

Four constraints must be respected by the control system:

1. a higher fan speed or an additional cooling tower must be used when the water temperature at the reservoir outlet reaches 26°C in storage mode and 28°C in other operating modes;

2. in the designed control strategy, the minimum water temperature at the condenser inlet is equal to 22°C (to avoid a too low condensing pressure in the chillers). This restriction is more severe than the recommendations of the chiller manufacturer ;

3. the glycol water temperature at the heat exchanger inlet can not be lower than 3°C to avoid the risk of freezing in the heat exchangers;

4. in direct production mode, the glycol water flow rate at the equilibrium tank exhaust (plate heat exchanger side) must be greater than the total glycol water flow rate in the evaporators. This means that a bypass of a fraction of the plate heat exchanger flow rate should always occur inside the equilibrium tank, resulting in a temperature increase at the equilibrium tank exhaust (plate heat exchanger side)

The designed control strategy is, of course, function of the cooling plant operating modes. This paper focuses on two modes :

- **Discharge only** : two primary pumps are running. The valve VR2 is progressively opened to respect the set point temperature at the building inlet while the opening ratio of the valve VR1 is adjusted to respect the third constraint.

- **Direct production (priority) with possible discharge** : when only the chillers are used, the opening ratio of the valve VR1 is adjusted to respect the set point temperature at the building inlet (it is reduced when the cold water temperature at the building inlet is higher than its set point). When the fourth constraint is no more respected, an additional chiller starts up. Only one primary pump is used when the number of operating chillers is lower or equal to two. The minimum number of operating cooling towers is one unity higher than the number of operating chillers. A maximum of three cooling towers can be used when only one chiller is operating. The tower fans start up in succession when the first constraint is not respected: the low speed first, the high speed after. The opening ratio of the bypass valve of the warm water circuit increases when the water temperature at the condenser inlet does not respect the second constraint any more. When the ice storage must be used in addition to the chillers, the valve VR2 is progressively opened to respect the set point temperature at the building inlet while the opening ratio of the valve VR1 is adjusted to respect the third and fourth constraints.

HYDRAULIC ANALYSIS

Since no flow meter equips the cooling plant, a series of energy and mass balances based on the temperatures provided by the acquisition system and the cooling loads « measured » by the enthalpy flow meters is done.

In theory, these thermal and mass balances should allow the determination of the water and glycol water flow rates. However, the analysis of the data provided by the acquisition system points out the inaccuracy of several sensors. This reduces strongly the accuracy of the energy and mass balances. To reduce the uncertainty on the flow rates, a hydraulic analysis of the warm water and glycol water networks has been carried out (Bourdouxhe et al., 1996).

To model these circuits, the pressure drop, Δp, through a component crossed by the mass flow rate $\dot{M}$ is given by :

$$\Delta p = K \dot{M}^2 \qquad (1)$$

where K is the pressure drop coefficient ($kg^{-1}m^{-1}$), assumed to be constant (except for the modulating valves). This coefficient is determined on the basis of the information provided by the installers and the equipment manufacturers.

The hydraulic simulation is done by the software EES (Klein, 1996). The results are used to develop flow rate tables. These tables are then used by TRNSYS during the simulation of the thermal performance of the cooling plant.

It should also be noted that the hydraulic analysis of the cold water network has still to be done. Therefore, this side of the system is modeled very simply : the cold water flow rate in the plate heat exchangers is assumed to be only function of the number of operating secondary pumps and only the cold water temperatures at the supply and exhaust of the plate heat exchangers will be simulated.

COMPONENT MODELING AND PARAMETER IDENTIFICATION

The cooling tower and twin-screw chiller models considered in this paper are extracted from the « Toolkit for Primary HVAC System Energy Calculation » (Bourdouxhe et al., 1995). They have been adapted to be used by TRNSYS. The Toolkit models are static, simple, but physically meaningful. Thanks to the modeling approach considered in the Toolkit, a minimum number of parameters are required to reproduce the behavior of the component. These parameters can be identified on the basis of data currently available in the manufacturers' catalogs.

The counterflow cooling tower model is characterized by one parameter, the heat transfer coefficient of the tower. The evolution of this parameter with the air and water mass flow rates is defined on the basis of the manufacturer 's data and the experiments carried out with a similar cooling tower (Jennes, 1996).

The twin-screw chiller is represented by four components: the twin-screw compressor, the evaporator, the condenser and the expansion valve (presumed to be isenthalpic). Both condenser and evaporator are represented as classical heat exchangers. Their heat transfer coefficients are the first two parameters to be identified in order to model the whole chiller.

The following linear relationship is used to define the motor-transmission subsystem :

$$\dot{W} = \dot{W}_{lo} + \alpha \dot{W}_{in} + \dot{W}_{in} \qquad (2)$$

where $\dot{W}$ = power consumed by the compressor (W)

$\dot{W}_{in}$ = internal power of the compressor (W)

$\dot{W}_{lo}$ = third chiller parameter (the constant part of the electromechanical losses) (W)

α = fourth chiller parameter (the electromechanical losses proportional to the compressor internal power) (-).

The « swept » volume flow rate of the screw compressor in full-load and the internal leakage (characterized by an equivalent leakage area) allow the determination of the refrigerant mass flow rate flowing through the chiller. They are the last two parameters identifying the component behavior in full-load regime. These six parameters are identified on the basis of operating points in full-load regime provided by the manufacturer (Bourdouxhe, 1996).

In part-load, moving a slide valve opens a recirculation passage and bypasses a portion of the refrigerant back to suction before much compression occurs. The part-load losses are characterized by the pressure jump, dp_{pump}, encountered by the refrigerant diverted. This is the only parameter characterizing the part-load regime. It is identified on the basis of one operating point in part-load regime provided by the manufacturer (Bourdouxhe, 1996). All these parameters are presumed to remain constant.

The plate heat exchanger is modeled as a classical counterflow heat exchanger. Its global heat transfer coefficient, AU, is assumed to be function of the thermal resistance on the cold water side (R_w), the thermal resistance of the metal (R_m) and the thermal resistance on the glycol water side (R_g) :

$$\frac{1}{AU} = R_w + R_m + R_g \qquad (3)$$

where $R_w = c_1 (\dot{M}_w)^{-0.8}$ (K W^{-1}) ;

$R_g = c_2 (\dot{M}_g)^{-0.8}$ (K W^{-1}) ;

$\dot{M}_w$ is the water flow rate (kg s^{-1}) ;

$\dot{M}_g$ is the glycol water flow rate (kg s^{-1}) ;

The coefficients c_1, c_2 and R_m are identified on the basis of data provided by the manufacturer (Bourdouxhe, 1996).

Such as for the previous components, the model of encapsulated ice storage tank (Bilas, 1995) is a simple semi empirical model. The theoretical heat transfer coefficients used to represent the thermal behavior of the tanks liquid or solid phase or during the freezing or the melting, are calibrated on the basis of the information given by the manufacturer.

SIMULATION OF THE DISCHARGE MODE

The storage mode is selected when moderate cooling loads are expected. Two TRNSYS simulation files are developed in order to analyze the cooling plant in discharge mode. The first one simulates the discharge without any control while the second file simulates the discharge, including the control of the plant (actions on both VR1 and VR2 valves).The first file can be used in order to simulate the plant provided that the actions on the control valves are known : VR1 and VR2 are, in this case, adjusted to their monitored or assumed value. The third required input is the building cooling demand. The second file is designed in order to simulate an hypothetical (and ideal) controller and to optimize the action on the control valves while maintaining the required set point and matching the constraints on the cold side of the heat exchangers. The only required input is the building cooling demand.

The first simulation file was used to simulate a real discharge test performed on the plant (Bourdouxhe et al., 1997). During this test, the action of the BEMS on the control valves was modified with respect to the design strategy. Figure 2 shows the evolution of the building cooling load (measured by the enthalpy flow meters at the inlet of the buildings) together with the opening ratios of the VR1 and VR2 valves, which are activated as follows:

- VR1 is limited above 80% opening rate in order to prevent cavitation problems with the primary pumps ;
- VR2 is manually controlled according to the schedule represented by Figure 2.

The figure shows that VR1 is continuously maintained at the 80% opening ratio while VR2 is progressively opened in order to admit more and more glycol water at the heat exchanger inlet. Using this strategy, the set point on the building inlet water temperature is more or less maintained (see Figure 3). However, at the end of the test, the cooling loads are too important and the cold water temperature slightly increases above the 7°C set point. The simulation results pointed out a default with one sensor, located at the outlet of the ice storage tank (a 3.6° C temperature difference). Taking that problem into account, the simulation results match the monitored values with a very good agreement (see Figure 3).

In the second simulation file, the controller model is designed in order to find a kind of "optimization" of the operation of the valves VR1 and VR2, while maintaining the set point. The results of this strategy are compared to the results obtained with a manual operation of the VR2 valve during the discharge test. Figure 4 shows the building inlet temperature calculated by the first and second simulation files. The figure shows that, with the second file, the set point of 7°C is almost perfectly maintained during the whole discharge test. Major differences between both simulations occur during the first hour of discharge.

Furthermore, oscillations are limited around an average value, which corresponds to the set point. Comparison between the opening ratio of both valves is shown in Figure 5. The controlled simulation continuously maintain the valve VR1 totally opened. Concerning VR2, the simulation including control appears as more "economical" as far as the use of the storage is concerned. This results in a better control of the building inlet temperature.

The comparisons presented here above show that the building inlet temperature is heavily depending on the activation strategy of the VR2 valve. With the controller active, the supply temperature is maintained at the required set point (7°C). With manual control, fluctuations of the supply water temperature are higher. As a consequence, the use of the ice storage is slightly different and this might be a critical factor for days where the energy content of the store matches the cooling load of that day. This shows that there is some room for optimization, considering two opposite aspects : cost (storage consumption) and comfort (building inlet set point). This optimization problem appears still more crucial if more complex operating modes are concerned. As an example, the next section considers the operation of the cooling plant as the combination of direct production and discharge modes.

SIMULATION OF THE DIRECT PRODUCTION MODE

The direct production mode with/without discharge is selected when high cooling loads are expected. Such as for the discharge mode, two TRNSYS simulation files are developed in order to analyze the cooling plant in this operating mode. The first one simulates the plant thermal performance provided that the user gives as inputs the time evolution of the outside air dry-bulb temperature and relative humidity, the building cooling loads and the plant control variables. This file allows the validation of the simulation when the inputs are the values recorded by the BEMS. The second simulation file allows the simulation of the plant with its regulation. The required inputs are the time evolution of the outside air dry-bulb temperature and relative humidity, the building cooling loads. The values of the control variables are determined by the regulation routine. For each time step, this routine calculates the values of the control variables that respect the set point temperatures and the constraints. The implemented control is not an optimal one, but follows the strategy developed at the plant design stage (e.g. the opening ratio of the valve VR1 is reduced if the chillers do not meet the set point temperature at the building inlet. This reduces the glycol water temperature at the plate heat exchangers, but it also reduces the glycol water flow rate in these heat exchangers. Consequently, the minimum opening ratio of the valve VR1 could not necessarily lead to the minimum water temperature at the building inlet). Nevertheless, the regulation routine is flexible : it allows the change of the set point temperatures and of the constraints on the fluid temperature. This allows the simulation user to point out the impact of the value of the set points, of the constraints and of the number of available components on the thermal performances of the cooling plant (available cooling capacity and power consumption).

The cooling plant and its regulation are simulated on the basis of the design conditions (i.e. 4.5 °C for the set point temperature at the evaporator exhaust, 7 °C for the set point temperature at the building inlet and 22 °C for the outside air wet-bulb temperature during the peak loads). Figure 6 shows the time evolution of the cold water temperature at the building inlet, of the glycol water temperatures at the evaporator outlet and at the ice storage tanks outlet. A tolerance of ± 0.3 °C is taken into account for the respect of the set points. As can be seen, the set point temperature at the evaporator exhaust is always respected contrarily to the set point temperature at the building supply. Indeed, after the peak cooling loads (i.e. after 16 hr), the cooling capacity of the storage tanks becomes insufficient to meet the cooling demand while respecting the set point at the building supply. As a result, the cold water temperature increases and overpasses slightly the set point. Figure 6 also shows that the glycol water temperatures at the ice storage tanks outlet reaches 7 °C meaning that almost all the cooling energy is extracted from the tanks. Figure 7 shows the time evolution of the cooling capacity provided by the chillers and the ice storage tanks.

The same simulation is repeated, but with a lower set point temperature at the evaporator exhaust (3 °C). Figure 8 shows that, during the peak hours, the set point temperature at the evaporator exhaust is not always respected, meaning that the chillers are working at their maximum cooling capacity. Now, the set point temperature at the building supply is never overpassed. However during the low cooling demand period, the building supply water temperature is too low ; this is due to the fact that, with 3 °C at the evaporator exhaust, the opening of the valve VR1 is not sufficient to increase enough the level of glycol water temperature in the plate heat exchangers. The time evolution of the glycol water temperatures at the ice storage tanks outlet (see Figure 8) and of the cooling capacity provided by the ice storage tanks (see Figure 9) shows that the design cooling demand can be met without extracting completely the cooling energy stored in the ice storage tanks.

CONCLUSIONS

After a hydraulic analysis and a parameter identification of the equipment models based on the data provided by the installers and the manufacturers, the simulation software TRNSYS is used to simulate the cooling plant and its regulation.

The validation of the discharge mode shows that the ice storage tank and plate heat exchanger models work very satisfactorily. A good regulation of this operating mode is also important in order to reduce as much as possible the cooling energy extracted from the ice storage tanks.

The simulation of the direct production mode with/without discharge on the basis of the set points defined during the design phase leads to a slight overpassing of the set point temperature at the building inlet and to an almost complete discharge of the ice storage tanks. It seems therefore that the cooling equipment is correctly sized. However, the cooling plant can “do more” if the set point temperature at the evaporator outlet is reduced. But, during low cooling demand period, this can lead to a too low building supply water temperature. It is therefore essential to adjust continuously the set point of the chillers in order to maximize the cooling capacity of the cooling plant and to control accurately the cold water temperature supplying the buildings.

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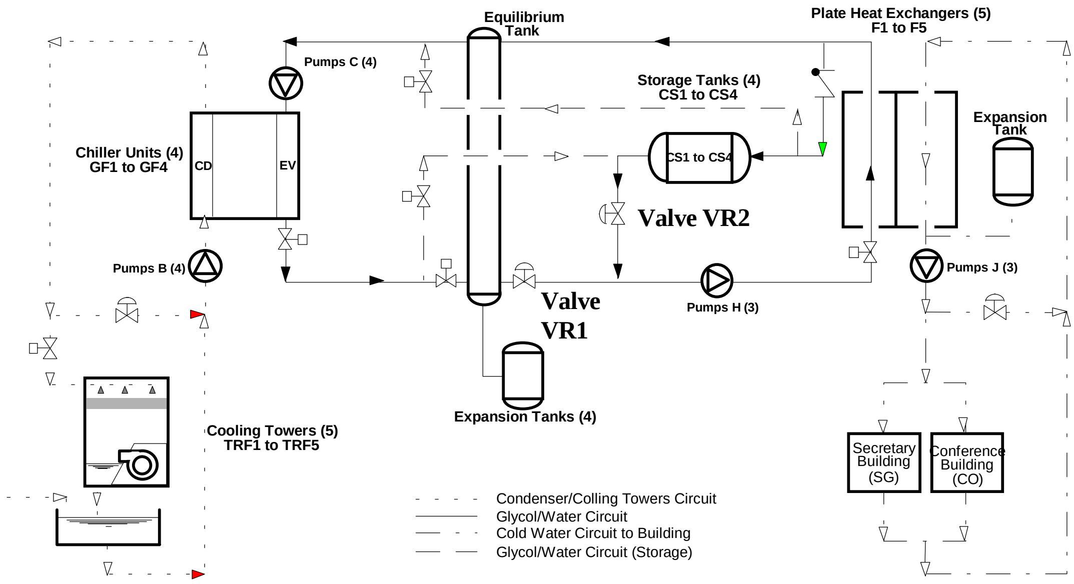
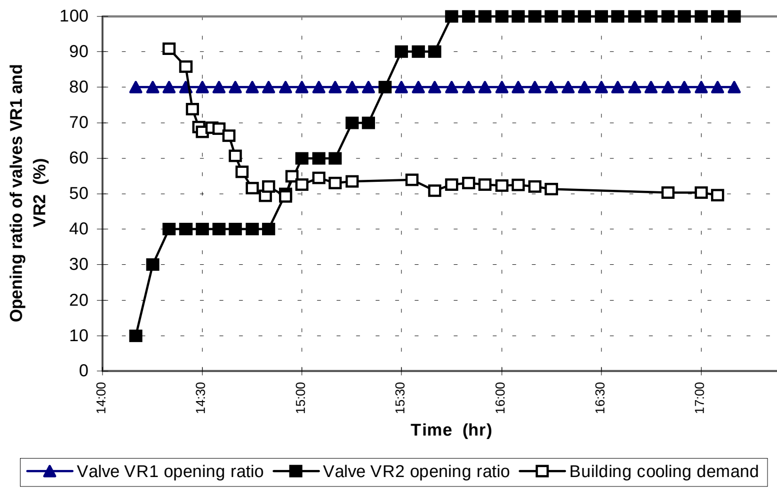


Figure 1 : Schematic view of the centralized cooling plant



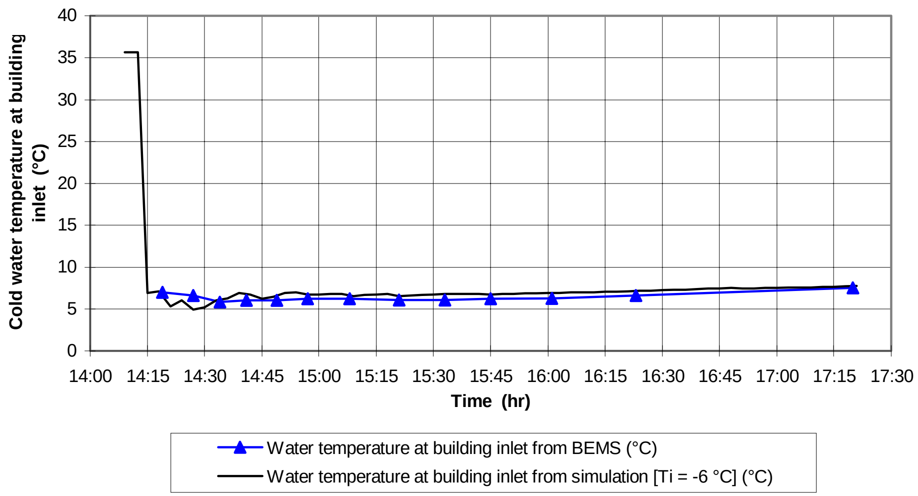
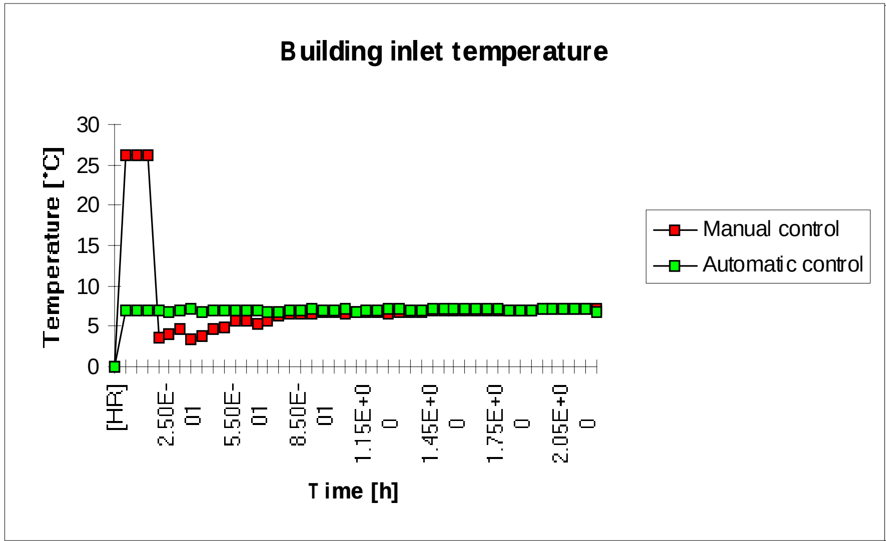


Figure 2 : Cooling load evolution and opening ratios of VR1 and VR2 valves

Figure 3 : Comparison simulated/monitored building inlet water temperatures



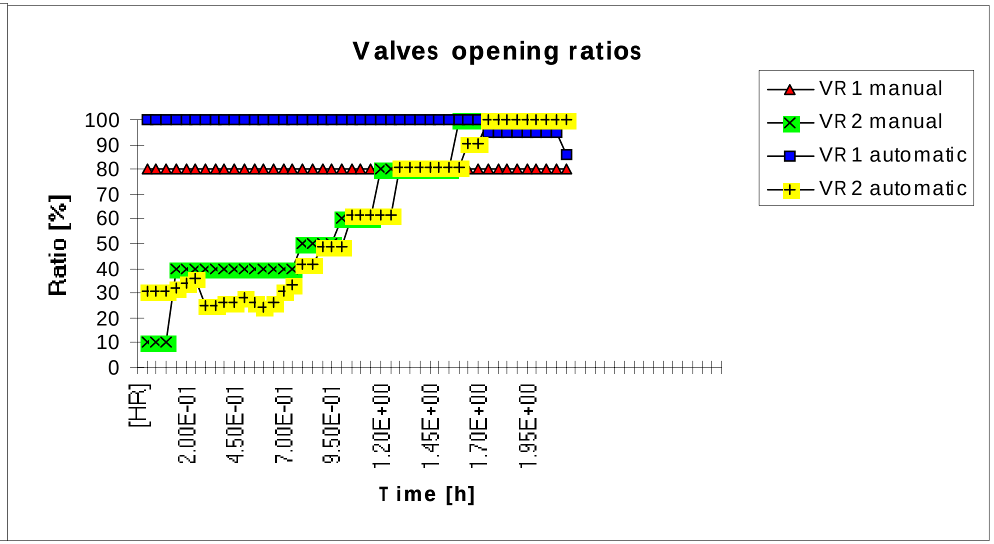


Figure 4 : Building inlet temperatures given by both simulation files (manual and automatic control respectively)

Figure 5 : Evolution of the VR1 and VR2 opening ratios for the manual and automatic control cases.

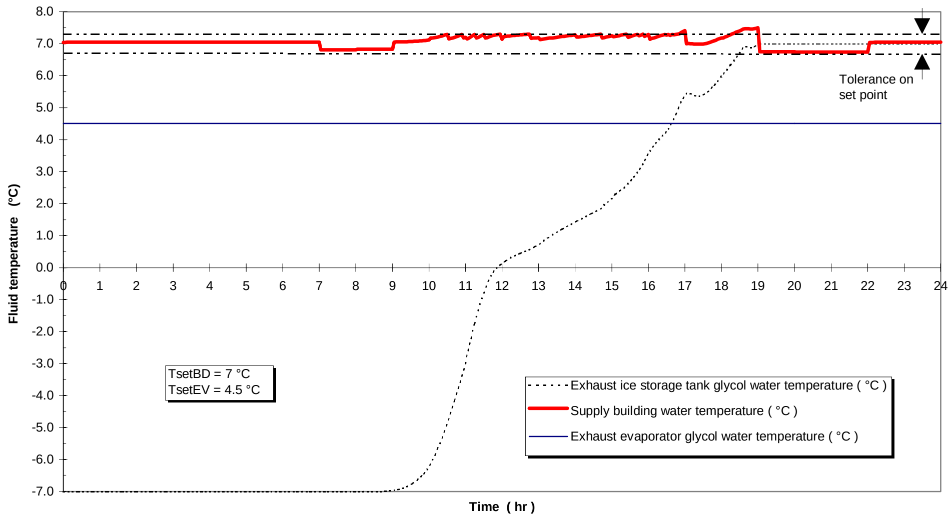


Figure 6 : Cold water temperature at the building inlet, of the glycol water temperatures at the evaporator outlet (set point : 4.5 °C) and at the ice storage tanks outlet

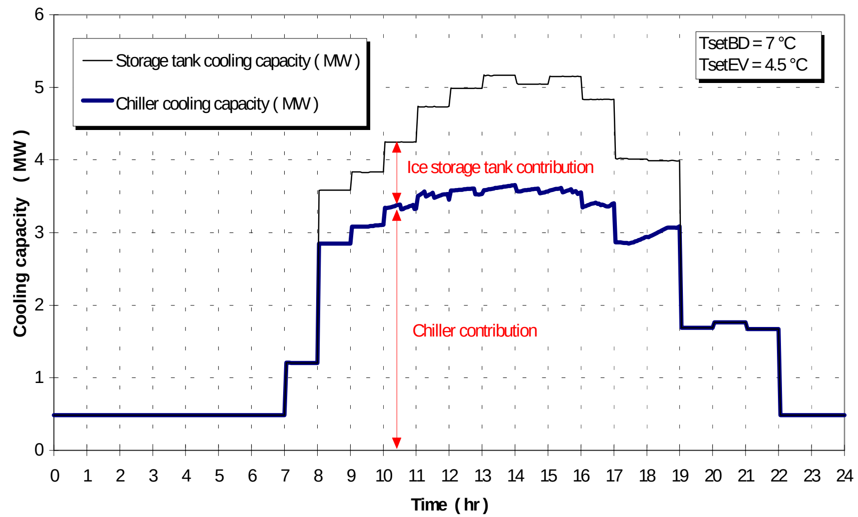


Figure 7 : Cooling capacity provided by the chillers and the ice storage tanks (set point at evaporator exhaust : 4.5 °C)

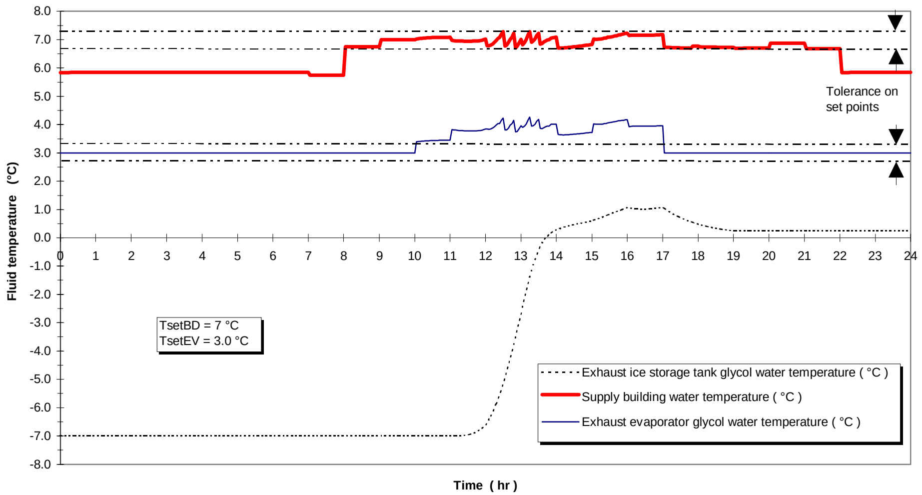


Figure 8 : Cold water temperature at the building inlet, of the glycol water temperatures at the evaporator outlet (set point : 3.0 °C) and at the ice storage tanks outlet

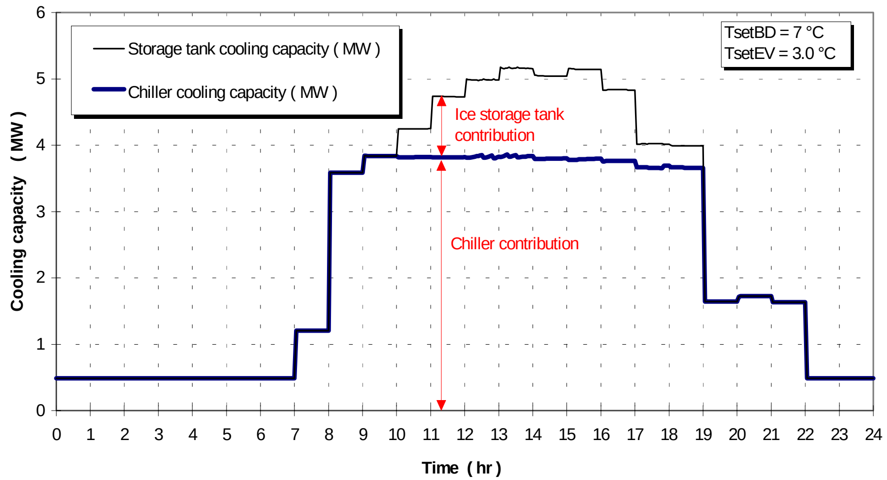


Figure 9 : Cooling capacity provided by the chillers and the ice storage tanks (set point at evaporator exhaust : 3.0 °C)