Control strategies for heat driven chillers to reduce parasitic electric consumption

J.L. Corrales Ciganda, Felix Ziegler

Institute of Energy Conversion Machines, Technische Universität Berlin, Marchstrasse 18, 10587 Berlin, Germany
fax: +40(0)30314-22253, email: jose.l.corralesciganda@tu-berlin.de

Abstract

One of the possible ways to realise trigeneration is by centrally generating heat and power in CHP plants and producing cold by using the heat produced in the CHP plants and transported via district heat networks in decentralised thermally driven chillers. In this case, usually the efficiency of the CHP plant is out of the scope of optimisation so the main challenge faced to improve the primary energy efficiency of these systems is the reduction of the parasitic electric consumption related with chiller operation.
In this paper different control strategies for an absorption chiller driven by district heating are presented and their performance in terms of thermal and electrical efficiency compared. Performances figures are obtained from simulations of a reference plant under different conditions. The reference plant used for the simulations is based on a demonstration plant installed at TU Berlin, consisting of a 10kW absorption chiller rejecting heat via a dry cooling tower and cooling a server room with an almost constant cooling load of 6,6kW. The presence of a cold water storage tank is expected to improve the chiller performance and has been evaluated within this work too. The results show that with a new presented control strategy power savings of around 10% can be obtained compared with usual control strategies.

Keywords

District heating, absorption chillers, control strategies, heat rejection

Introduction

Hot water driven absorption chillers can be integrated into district heating networks upgrading combined heat and power generation systems (CHP) to systems providing in a combined way heat, cooling and power (CHCP). This kind of CHCP systems have been investigated within the European project PolySMART [7]. One of the main challenges faced to improve the primary energy efficiency of these systems is the reduction of the parasitic electric consumption at the heat cooling plant, related to hydraulic design, plant components installed and control strategy used.
The control of the absorption chiller is usually realized by controlling the chilled water outlet temperature [6]. The desired outlet temperature can be achieved by adjusting the inlet temperature at either generator or absorber/condenser or both of them. Usually the evaporator inlet temperature is given as the return from the cooling load but its value can also be reduced by mixing water from evaporator outlet with that coming to the inlet, including a 3-way valve at the chilled water circuit. Another possibility to to control the cooling load of the chiller is by

adjusting the flow rates of the the external circuits, as has been investigated by Mittermaier [4]. The variation of the solution flow rate inside the chiller or a combination of all methods mentioned before are possible ways to control the chiller load also. A new method adjusting the cooling water temperature by using the characteristic equation has already been proposed by Kühn [3] for solar cooling applications. This method has been adapted and used in this work and some additions have been made. In this paper only chilled water control strategies based on cooling water and chilled water inlet control are investigated.

1. Methodology and main assumptions

A dynamic model of the absorption cooling plant installed at TU Berlin has been developed using Modelica. Component models for pipes, pumps, valves and their related controls have been implemented modifying components from the Modelica Fluid Library. The models have to be fed with parameters extracted from data obtained from manufacturers. The 10 kW absorption chiller is modelled using a modified version of the characteristic equation model [2] that includes thermal masses to simulate the transient behaviour of the chiller. A simplified model for cooling towers proposed by Stabat [1] is used for characterising the dry cooling tower used as heat rejection device. The water storage tank is modelled as a series of discrete water volumes in thermal exchange within their layer and their environment by convection and with conduction between the layers.
The thermal loads are almost constant at around 6,6kW and it is assumed that the ambient temperature for heat losses is 20°C for all plant components. The heat district water flow is assumed to have a constant value in pressure and a temperature of 90°C.

2. District heat driven absorption cooling plant model

Figure 2 shows a scheme of the simulated cooling plant. The main components present in the plant are the thermally driven or absorption chiller (TDC), a chilled water storage tank (TANK), a dry cooling tower and two fan coils that deliver the cold produced to the cooling loads (FC1 and FC2). The driving heat is transferred to the chiller via the district heating circuit (DH) that is not modelled in this work. Instead a constant flow rate of 1,2 m^3/h and an associated constant power consumption at pump P6 is assumed.

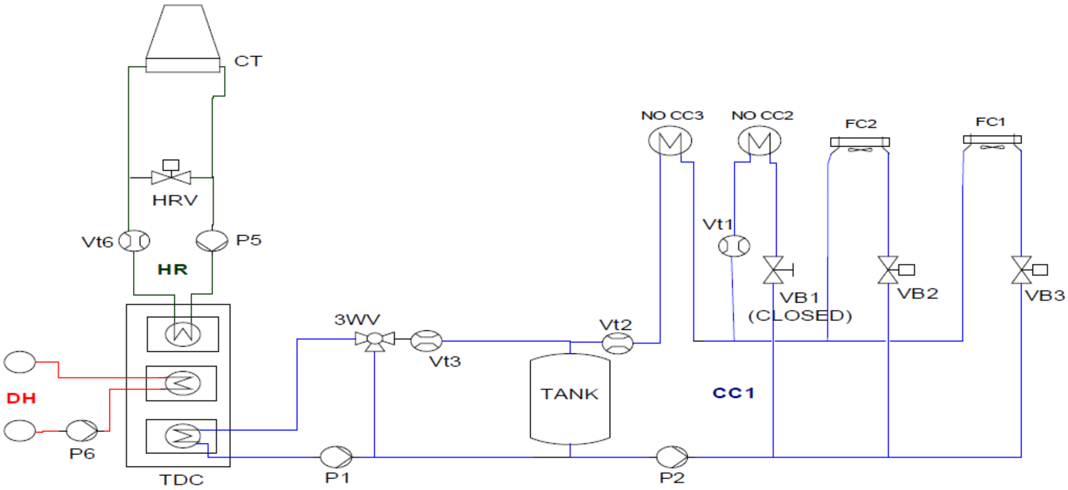

Figure 1: simulated absorption cooling plant

The heat rejection circuit (HR) includes a frequency controlled pump (P5) and a by-pass valve (HRV) and connects the TDC and the cooling tower. The chilled water circuit (CC) includes 2 pumps (P1 and P2) four controlled valves (VB1-3 and 3WV) that drive the flow between chiller, tank and fan coils. In the real plant two additional heat exchangers in the chilled water circuit transfer the cooling heat flow from the main chilled water circuit to two additional circuits (CC2 and CC3), responsible for cooling offices and lecture rooms at TU Berlin. These secondary chiller circuits CC2 and CC3 have not been simulated in this work.

3. Model validation

The performance of the cooling plant has been evaluated using measured temperatures, flows and and power consumption in the frame of the PolySMART project. Additionally, its primary energy related performance has been evaluated [8]. Therefore it is possible to validate the cooling plant model: simulation results are compared with measured for flow rates at the heat rejection and chilled water circuits (see figure 1 for plant location) and related power consumption for the first 10 hours of a summer day operation. In figure 2 simulated and measured values for flow rates at the flow meter Vt1 to Vt6 are plotted against time.

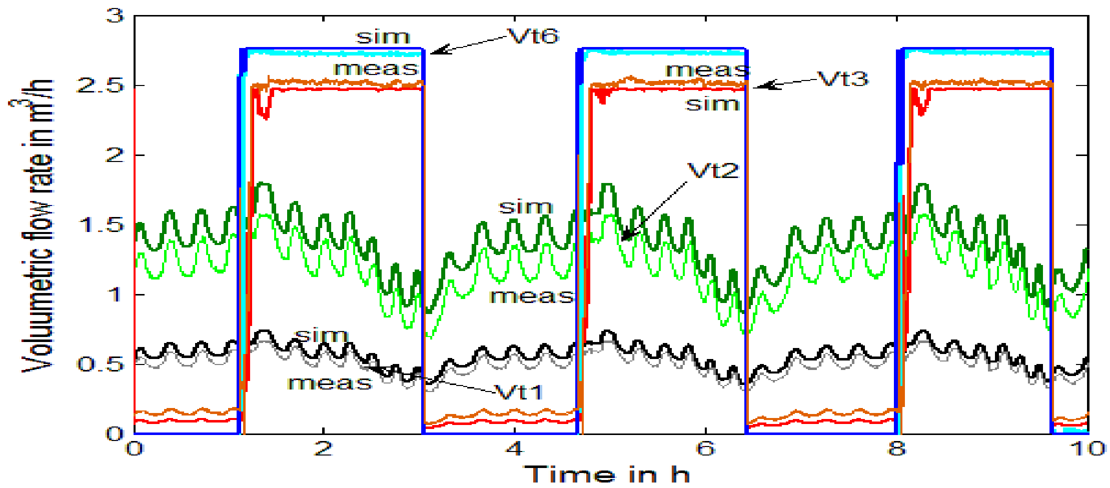

Figure 2: measured and simulated flow rates

Results show that the model can simulate quite accurately the hydraulic behaviour of the heat rejection circuit. The simulated results for flow meters Vt1 and Vt2 show less agreement than those for Vt2 and Vt6. Some hydraulic resistances seem to have been underestimated in the simulation. The obtained accuracy is considered sufficient for the goals of this work. Inaccuracies in the hydraulic performances results will also cause inaccuracies in the power consumption simulations. In figure 3 the simulated and measured values for power consumption at main plant components for the same summer day are presented.

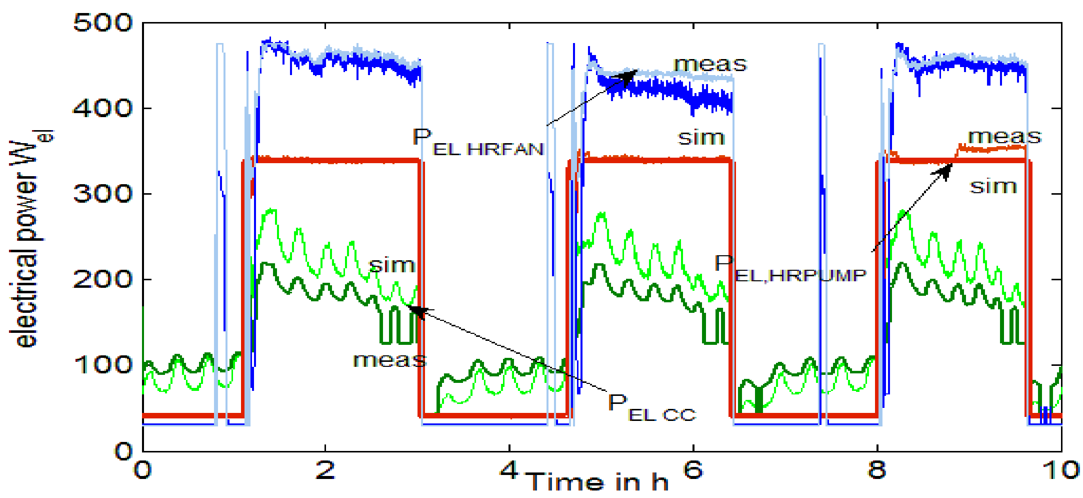

Figure 3: measured and simulated power consumptions

$P_{EL\ HRFAN}$ is the power consumption of the fan at the heat rejection unit (dry cooling tower, CT). $P_{EL\ HRPUMP}$ is the combined power consumption of the pumps at heat rejection and driving heat circuits (P5+P6) and $P_{EL\ CC}$ is the consumption of the pumps at chilled water circuit (P1 + P2). Simulated values for $P_{EL\ HRFAN}$ and $P_{EL\ HRPUMP}$ are in very good agreement with measured ones. The simulation for the power consumption at the chilled water circuit $P_{EL\ CC}$ seems to fit not so well, showing a discrepancy of about 30% with measured results. It was not possible to find an explanation for this difference, as the measured values are above the maximal values supplied by the pumps manufacturer To calculate the total electric consumption associated with the cold production, the electric consumption for internal chiller pumps and control ($P_{EL\ TDC}$) must be also considered. This value is constant during operation (150W) and stand-by times (49W), and can be reproduced without deviation at all for simulation. For that reason their curves are not shown in figure 3.
The total electrical power demand of the plant, the consumption for a given period, and, knowing the cold produced at the chiller the electrical performance factor for this period can be calculated (see table 1).

$$P_{EL\ PLANT} = P_{EL\ HRFAN} + P_{EL\ HRPUMP} + P_{EL\ CC} + P_{EL\ TDC}$$

$$W_{PLANT} = \int P_{EL\ PLANT}\, dt$$

$$COP_{EL\ PERIOD} = Q_E / W_{EL\ PLANT}$$

In the 10 hours period taken for validation the cold production was $Q_{TH\ E}$ =48,4 kWh.

Table 1: Measured and simulated electrical performances

	Measurement	Simulation	Deviation
$W_{EL\ PERIOD}$ [kWh]	7,06	7,04	0,30 %
$COP_{EL\ PERIOD}$ [-]	6,85	6,83	0,30 %

The agreement between measured and simulated values is extremely satisfying.

4. Simulated system configurations and control strategies

After having shown that the numerical model is valid for simulation, it is used to compare different control strategies.

4.1 Cooling tower outlet control *(T_{CW} C)*

In the strategy implemented at the experimental plant at TU Berlin the water temperature at the outlet of the cooling tower (T8 in figure 5) is maintained at a constant value via a PI-controller that changes the fan speed of the tower. The temperature of the chilled water leaving the chiller (T2) is maintained at its set value using a three way mixing valve in the chilled water circuit. If the 3-way mixing valve is completely closed and the chilled water outlet temperature falls below the set temperature a valve at the heat rejection circuit (HRV) bypasses part of the flow to the cooling tower and increases the value of the cooling water temperature over the water temperature value at the cooling tower outlet, thus reducing the cooling power furthermore. Depending on the demanded load and the set value for the cooling tower outlet temperature a given position will be adopted by the controlled values.
In figure 5 a control scheme for this control for a plant including a chilled water tank is shown. The chiller and related circuits will stop operation once the chilled water at the top of the tank is lower than a set value. In this way, the chiller in the plant operates in an on-off mode.

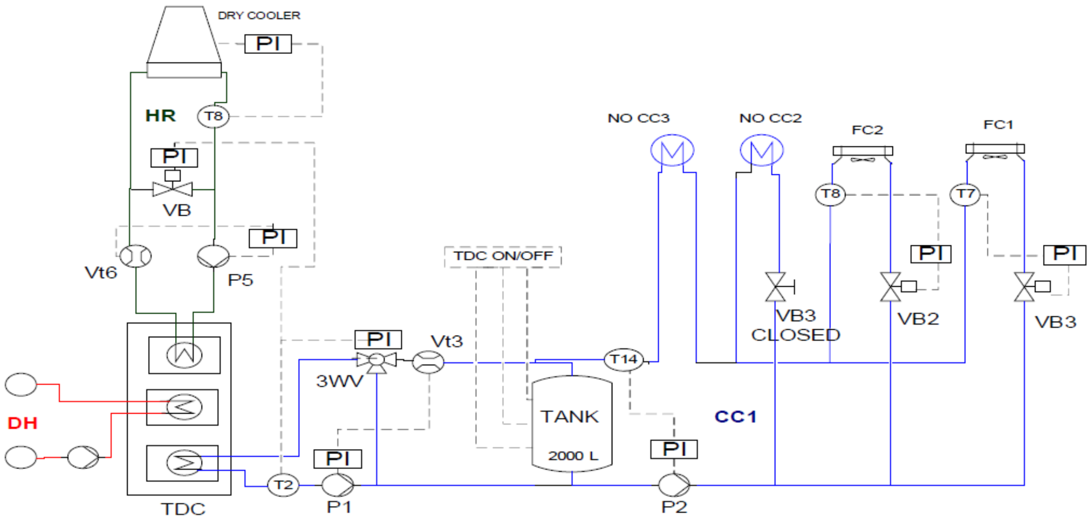

Figure 4: constant cooling tower control with storage tank

Figure 5 shows the flow diagram for the control logic.

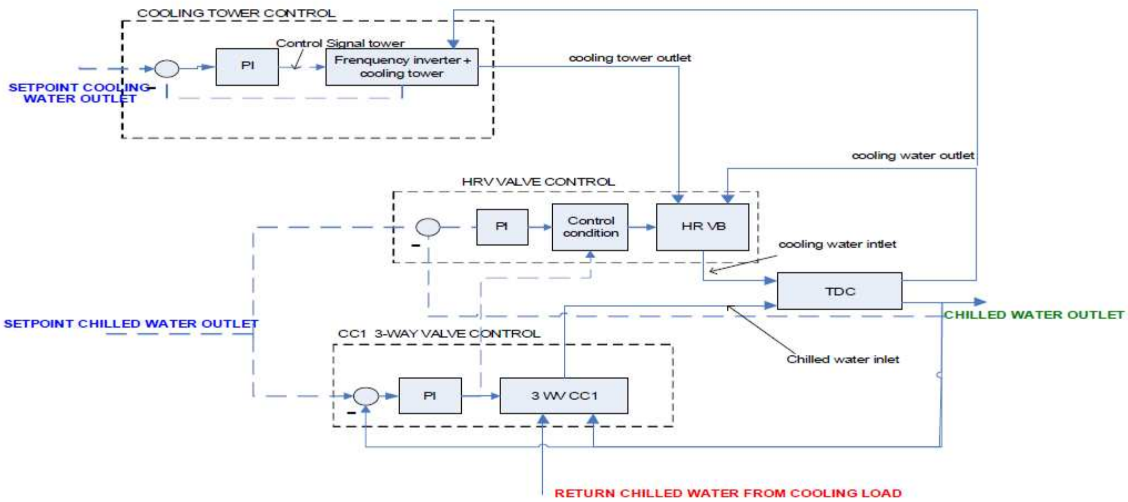

Figure 5: flow diagram for the constant cooling tower control with storage tank

4.2 Adjustable cooling tower outlet control *(T_{CW} V)*

Is presented in figure 6. A controller based on the characteristic equation [3] describing the chiller partial load calculates the cooling water temperature which for nominal flow rates and given inlet hot and chilled water temperatures set the outlet chilled water temperature at the desired value. Using this strategy the 3-way valve at the chilled water circuit is no longer needed. The bypass valve at the heat rejection circuit will start to open only if even with no operation of the cooling tower fan the chilled water at the chiller outlet falls below the set value. The flow rate at the heat rejection circuit is maintained constant at its nominal value during operation.

As for all other control strategies presented a control for a tank including a tank is also the developed. The controlled constant flow rate at the evaporator changes if a chilled water tank is present. With a tank, the nominal flow rate for the chiller is used. With no tank, the evaporator flow rate is adapted to the value required by the load.

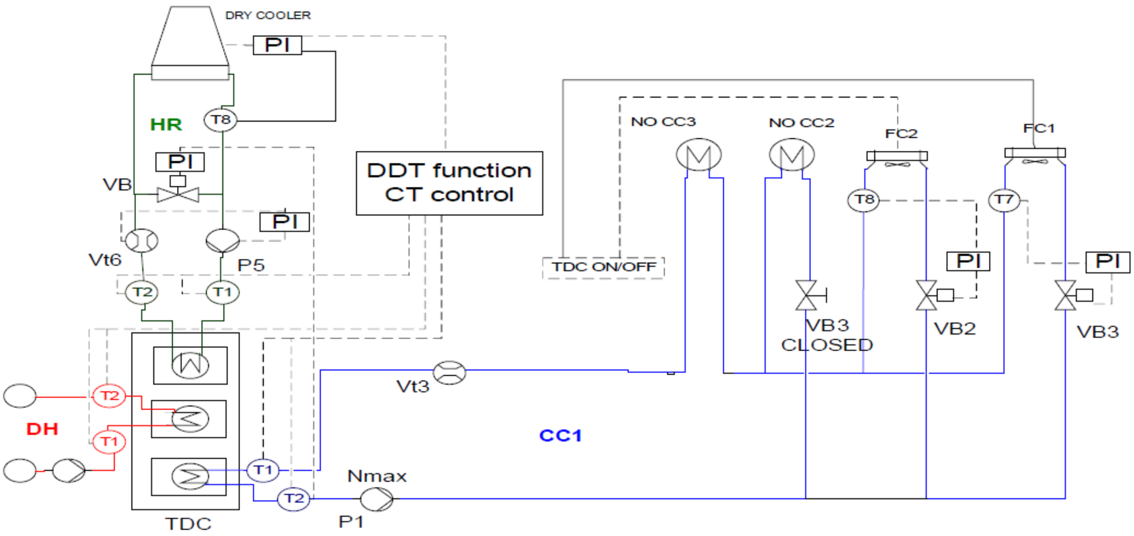

Figure 6: adjustable cooling tower control without storage tank

The flow diagram for this control is shown in figure 7.

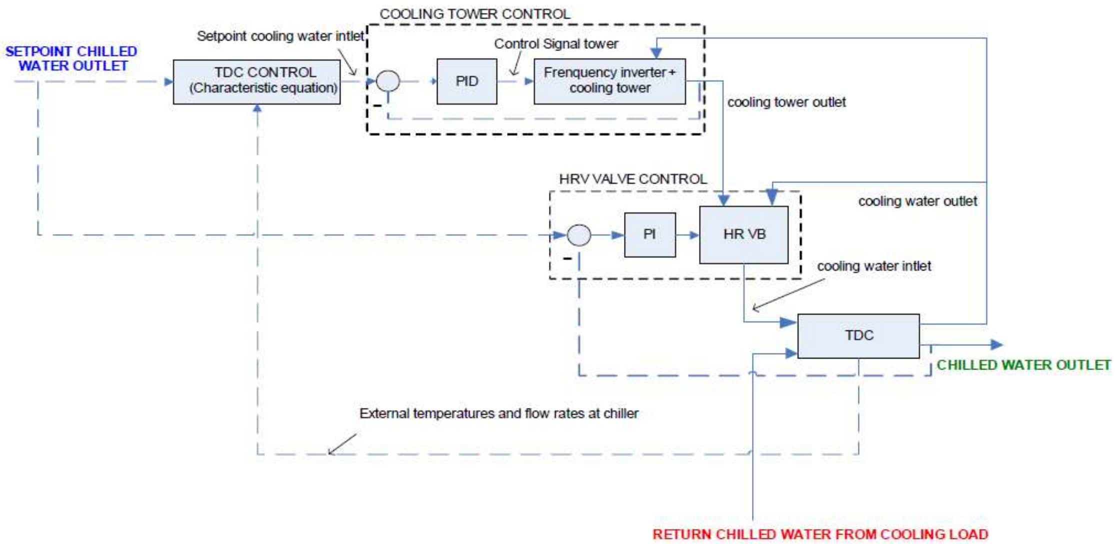

Figure 7: flow diagram for the adjustable cooling tower control without storage tank

4.3 Improved adjusted cooling tower outlet control *(T_{CW} V+)*

It has been observed that power consumption at the heat rejection circuit can be lower for constant cooling control for ambient temperatures. The hydraulic behaviour of the rejection heat circuit has been found to be the reason. For low ambient temperatures the constant cooling tower outlet control opens the bypass valve HRV in order to ensure that the chilled water outlet temperature stays at the set value. When that happens, power consumption at the heat rejection pump is reduced as part of the water from the circuit follows a path with smaller hydraulic resistances. If a constant cooling water temperature have to be maintained, the power consumption at the cooling tower fan rises because now the partial flow sent to the tower has to reduce its temperature to a lower level, but if the reduction in pump electric consumption is bigger than the increase in fan electrical consumption, the overall heat rejection electrical consumption can be reduced.

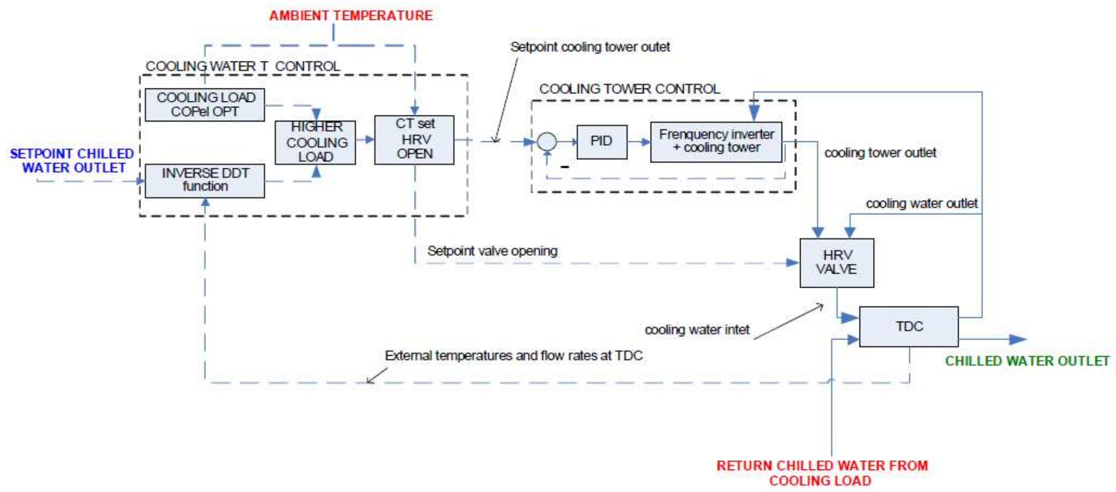

Figure 8: modified adjustable cooling water control strategy with storage tank

Consequently a third control strategy combines an adjustable cooling water temperature as presented in the last section with a controlled opening of the by-pass valve at the heat rejection circuit. The controller calculates (using a table in a first approach) for each ambient and cooling water set temperature the position for the by-pass valve HRV that optimises the power consumption at the heat rejection circuit. If a storage tank is included, the strategy goes one step further and tries to make the TDC work at a cooling load (or cooling water temperature) which optimises the electrical COP if this cooling load exceeds the demand and the tank is not yet fully loaded.

5. Simulation results

The control strategies described in section 4 have been used to simulate a period of 24 hours plant operation for the hottest and coldest of the year in Berlin using data taken from the Meteonorm database. [8]
The seasonal thermal coefficient of performance to is calculated in analogy to the electrical one defined in section two with the amount of district heat consumed for operation (Q_{DH}).

$$COP_{TH\ PERIOD} = Q_E / Q_{DH}$$

Table 2 shows the results for the hot day: a chilled water storage tank seems to improve the plant thermal performance whereas there is not a clear trend for electrical performance. The chilled water tank becomes full and the TDC is turned off at least two times for the sets including a tank, and thermal performance is expected to remain at these level for successive days for these systems.

Table 2: Thermal and electrical performances for the hottest day simulation

	TANK			*NO TANK*		
	T_{CW} *C*	T_{CW} *V*	T_{CW} *V+*	T_{CW} *C*	T_{CW} *V*	T_{CW} *V+*
TDC on/off cycles	2	3	3	-	-	-
$COP_{TH\ PERIOD}$	0,73	0,73	0,73	0,7	0,7	0,7
$COP_{EL\ PERIOD}$	8	8,4	8,8	7,6	8,3	9,1

TDC units in plants including a chilled water tank work at higher cooling loads when they are working and have longer shut down periods, in which no heat is consumed, therefore the thermal COP raises. Regarding control, electrical performances seems to be better for control strategies adjusting the cooling water temperature.

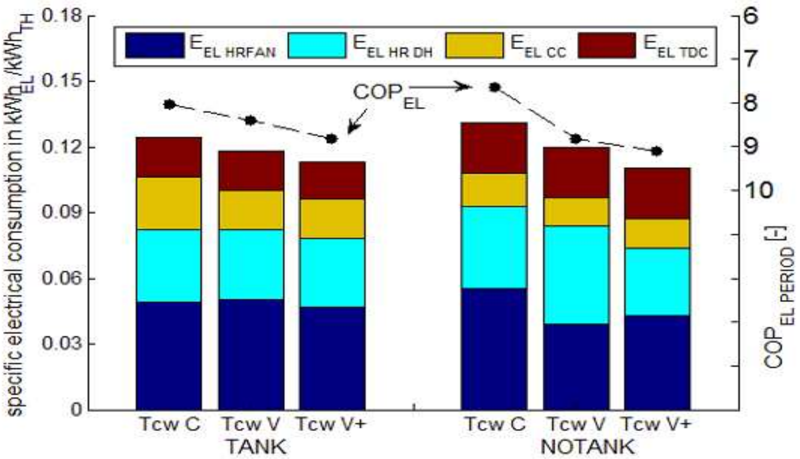

Figure 9: specific electrical consumption for a hot day simulation

In figure 9 contributions to specific electric power consumption of the plant are separated into terms proportional to the chiller cooling load and ambient temperature and others only dependent on the operation time of the TDC unit. The terms proportional to ambient temperature and load include the power consumption for heat rejection and district heating circuit, while the second one includes the internal electric consumption of the TDC and the power consumption for the chilled water circuit. The contribution of the second group is reduced when using a chilled water tank: with a tank the TDC operates at a higher cooling load in an ON/OFF mode being the internal power consumption at the TDC and auxiliary pumps reduced. Consumptions at the chilled water circuit are however lower in systems with no tank because they need one pump less at this circuit. Adjustable cooling water temperature also reduces the consumption at the chilled water circuit compared with constant cooling water temperature strategies, as they do not include a 3-way valve in the chilled water circuit.
The terms proportional to the cooling load and temperature represent over 60% of the overall consumption, and change with the control strategy used. It can be better understand how this terms change by looking at figure 10.

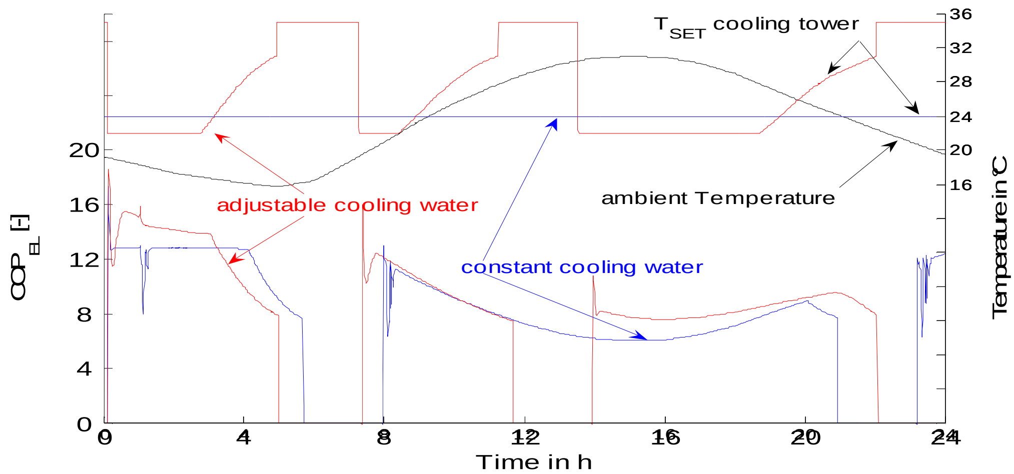

Figure 10: COP_{EL} in relation with set value for cooling water and ambient temperature

In figure 10 the electrical coefficient of performances are related with the set value for the cooling water temperature for each time of the hot day, for the constant and adjustable cooling water controls. It shows that for low ambient temperatures (at the beginning of the day) reducing cooling water temperature improves electrical performances as compared to letting the cooling water temperature at a fixed value of 24°C: the electrical COP can be improved from 13 to 15 by reducing the cooling water temperature to 22°C. For higher ambient temperatures, as it is the case for the period around 16h in figure 10, the electrical COPs are lower but still can be improved from 6 to 8 by adjusting the cooling water temperature. The power consumption of the heat rejection pump is better for control strategies that control the by-pass valve HRV. The results of the systems operating continuously at low ambient temperatures are shown in Table 3.

Table 3: Thermal and electrical performances for a cold day simulation

	TANK			TANK		
	T_{CW} *C*	T_{CW} *V*	T_{CW} *V+*	T_{CW} *C*	T_{CW} *V*	T_{CW} *V+*
TDC on/off cycles	4	4	4	-	-	-
$COP_{TH\ PERIOD}$	0,74	0,74	0,74	0,7	0,7	0,7
$COP_{EL\ PERIOD}$	12,4	12,8	13,8	12,8	9,9	12,2

For low ambient temperatures the system sets including a tank show not only better thermal but also better electrical performance. Now, without ambient temperature limitations, systems having a tank can work at higher TDC loads, reducing the time of operation in the ON/OFF cycles (more cycles in 24h) and reducing in this way the electrical consumption related with operation time. Comparing those sets not including a tank the simple adjustable cooling water control has a power consumption at the heat rejection pump almost double than for strategies that open and close the bypass valve.(see Figure 11). In this figure it can also be seen how for winter operation, the specific power consumption at the heat rejection fan been is reduced in more than 50% with respect to summer operation for all control strategies.

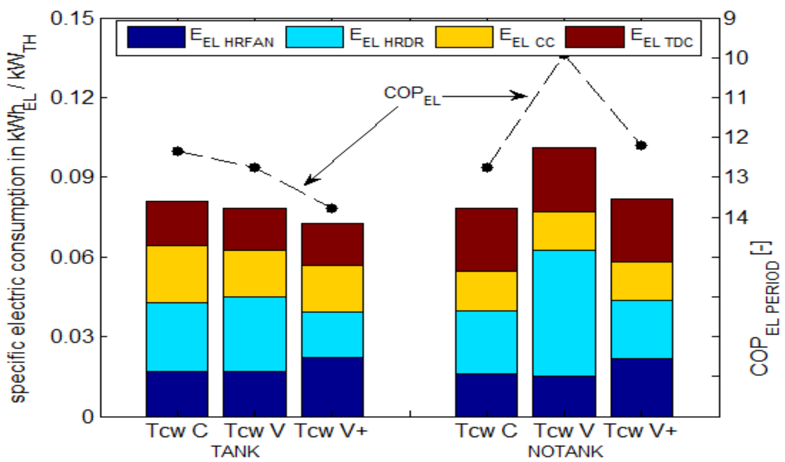

Figure 11: specific electrical power consumption for a cold day simulation

The new adjustable cooling water strategy with controlled by-pass valve opening show the best electrical performances. for the two simulated cases (high and load ambient temperatures).If the ambient temperatures is very high, the available cooling power is high enough to load the storage only at the expense of high parasitic power. For this reason simulated sets including a tank show worse performances for this hot summer day operation than sets not including it.

6. Conclusions

The simulation model for the absorption cooling plant has been validated in its hydraulic and electrical behaviour with measured data of the real plant. Using the simulation model, three different possibilities to control the cooling load have been evaluated. The presence of a chilled water storage tank in the plant have been also considered. The use of adjustable cooling water control strategy has demonstrated to save electrical power at the fan at the rejection heat device. The adjustable cooling water strategy has been modified to include a controlled by-pass of the heat rejection devices that reduce the power consumption of the rejection heat circuit pump. The presence of a chilled water storage tank in the plant improves its thermal performance and if the ambient temperature is not very high its electrical performance also.
With the presented control strategies savings in the specific electrical consumption of around 10% can be obtained. They are dependent on the components present in the heat rejection circuit, and especially on the relationship between air flow rate and power consumption of the cooling tower fan. If cooling towers with frequency controlled fans are used, larger savings can be achieved. First simulations made assuming this type of cooling tower being used obtained specific power consumption savings of up to 30%.

Nomenclature

COP: coefficient of performance [-]
E: specific electrical consumption [kW_{EL}/kW_{TH}]
m: mass flow rates [kg/s]
P: power [W]
Q: heat [kWh]
T: temperature [K]
W: electrical work [kWh]

Subscripts

CC chilled water circuit
DH district heating
HR heat rejection
EL electrical
TH thermal

References

[1] Stabat P., Marchio D.., 2004, "Simplified model for indirect-contact evaporative cooling tower behaviour", *Applied Energy*, Vol 78, Issue 4, pp. 433-451.
[2] Hellman H-M., Schweigler C., Ziegler F. 1989, "A simple method for modeling the operating characteritic of absorption chillers", Proceedings of Eurotherm Seminar No 59 Nancy-Ville, France 6-7 July 1987, pp. 219-226.
[3] Kühn A., Clauß V. Ziegler F, 2007, "A new control strategy for solar driven absorption chiller", Proceedings of the International Conference Solar Air-conditioning, Tarragona, Spain.
[4] Mittermaier M, Petersen S., Ziegler F., 2010: Primary Energy optimized operating strategies of Absorption Chillers in CHCP networks, Sustainable Refrigeration and Heat Pump Technology Conference, Stockholm, Sweden.
[5] Zegenhagen T., Corrales J., Petersen S., Ricart C., Ziegler F, 2010, "Best Practice: Data Center Cooling using CHCP Technology",Sustainable Refrigeration and Heat Pump Technology Conference, 13-16 June, 2010, Stockholm, Sweden.
[6] Kohlenbach P., 2006, "Solar Cooling with absorption chillers: control strategies and transient chiller performance", Dissertation Technische Universität Berlin.
[7]PolySMART, 2010: Polygeneration with advanced small and medium scale thermally driven air-conditioning and refrigeration technology, www.polysmart.org
[8] Meteonorm, 2009: Meteonorm software Version 6.1, www.meteonorm.org

Acknowledgements

This work has been developed withing the frame of the EU-funded project PolySMART