OPTIMIZED CONTROL STRATEGY OF A COMBINED HEATING, COOLING AND POWER SYSTEM

Matthias Schicktanz[(a)], Nan Kishore Kumuda Rajgopal[(b)], Hannah Neumann[(a)], Tomás Núñez[(a)]

[(a)]Dept. Thermal Systems and Buildings/Fraunhofer Institute for Solar Energy Systems ISE
[(b)]FCPower GmbH

ABSTRACT

Adsorption chillers have been under investigation for many years. Yet, little is known about optimizing the control strategy of a combined heating, cooling and power (CHCP) system where adsorption chillers, pumps, a heat rejection unit, etc. are involved. In this study the question has been analyzed for a system consisting of two adsorption chillers by using a transient model. The aim was to maximize the cooling capacity and at the same time maximizing the overall efficiency by taking heat and electricity consumption of the periphery into account. The primary energy consumption was used to sum up the different energy sources to one source. A parameter study shows the optimized operation states for different ambient temperature conditions.

1. INTRODUCTION

The term CHCP is used usually for two different system concepts. First, a centralized large scale combined heating and power unit (CHP unit) produces electricity and delivers heat via a district heating network to one or several thermally driven chillers (TDC), which produce cooling energy. Second, a decentralized small scale CHP unit producing heat and electricity and the heat is used directly to power a TDC at the same site. This paper is about the second concept and focusing on the cooling. The system was already described in earlier publications (PolySMART 2008), (Schicktanz et al. 2009). It consists of two TDCs, a CHP unit, a cooling tower and a chilled water distribution network. Earlier investigations revealed that the electricity consumption is high, mainly due to the cooling tower fan and the cooling water pump. The question arises how to operate the system at certain ambient temperatures in order to maximize the efficiency.

2. DESCRIPTION OF THE SYSTEM

Fig. 1 gives a graphical overview of the CHCP system. The CHP unit delivers heat to the buffer storage. The two adsorption chillers TDC 1 and TDC 2 are consuming this heat while delivering cold to the chilled water storage. The dry cooling tower with an optional spray function rejects waste heat to the ambient. From the chilled water storage cold is distributed to an open office and five small offices equipped with PCM chilled ceilings. The CHP unit produces electricity with an annual efficiency of η_{el}=26% and an annual thermal efficiency of η_{el}=60% (measured values). The water in the CHP loop must not exceed 60°C at the CHP inlet and reaches 75°C at the outlet. In order to meet this temperature difference the TDCs are connected in series in the hot water loop. Moreover, they are connected in series in the chilled water loop to increase the chilled water temperature spread. In the cooling water circuit both units are connected in parallel in order to assure the lowest possible heat rejection temperature for both units. Fast temperature changes occur at all outlets of the TDCs due to the switching process of the adsorption chillers. To avoid that these fluctuations influence the CHP unit inlet a stratified storage in the return line was installed. A three-way-valve mixes water from the top and the bottom of the stratified storage to meet the required inlet temperature of the CHP unit. The fan of the cooling tower is frequency controlled as well as the cooling water pump. The cooling water loop and the cooling tower are responsible for most of the electricity consumption of this system (Schicktanz et al. 2009). Optimizing the system implies to reduce the parasitic electricity consumption of these two components.

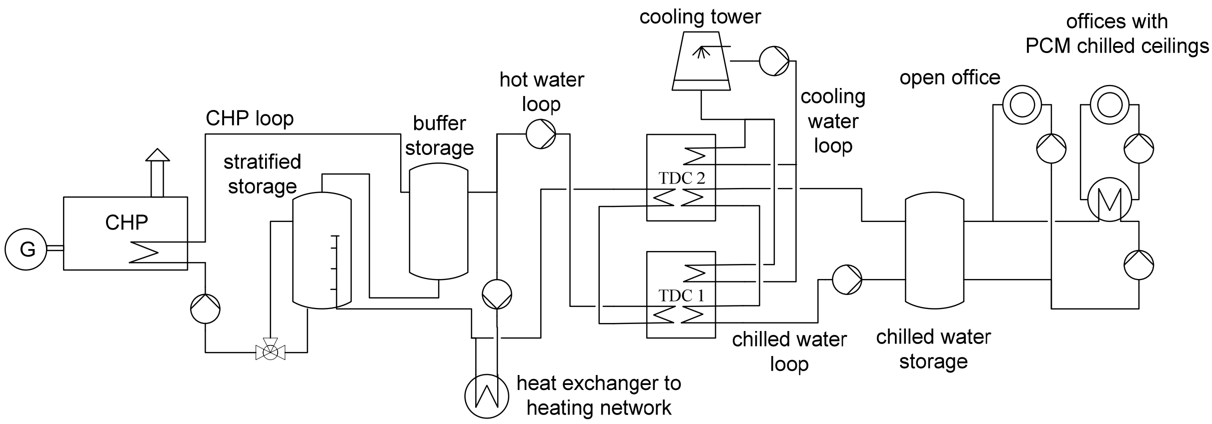

Figure 1 – Hydraulic scheme of the CHCP system.

3. OPTIMIZATION PROCEDURE

3.1. Optimization method

A computer model was used to calculate the optimized operation conditions of the system. Since the TDCs are connected in series in the hot and chilled water loop fast temperature fluctuations occur at the inlet of the second chiller in the loop and a transient model is required to calculate the performance. The used model was introduced in an earlier ISHPC conference (Schicktanz and Núñez, 2008), (Schicktanz and Núñez, 2009). To simplify the optimization problem the complexity of the system was reduced. Basically only the block of the TDCs with the cooling tower was modelled. In order to reduce computing time the block was subdivided into two separated tasks. The first task is the modelling of the cooling tower. A physical model as described in (Kumuda Rajgopal 2008) was used. States were calculated for different ambient and cooling water inlet temperatures such as different air and water flow rates. The air-side volume flow rate is described by the voltage signal of the frequency converter. A mesh was calculated with step size as given in tab. 1. Altogether 38720 points were calculated.

Table 1 – Calculation mesh of the cooling tower

	unit	min	max	step	points
Water flow rate	L/s	0.5	1.5	0.1	11
Water temp	°C	24	45	1	22
Amb. temp	°C	20	35	1	16
Fan speed	V	1	10	1	10
Total					38720

The second task is the modelling of the TDC block for different operation conditions. This implies variations of the hot and cooling water flow rates (the chilled water flow rate was kept constant), variable cooling and chilled water inlet temperatures (the hot water temperature was kept constant to 75°C) and variable sorption cycle lengths. Tab. 2 shows the used step sizes of the calculation mesh with altogether 3600 points. The flow rates and temperatures at the interface of the cooling tower and the cooling water loop of the TDC block were matched in a later step taking into account that the cooling water loops of the TDCs are connected in parallel, which means that the cooling tower flow rate is twice the individual TDC flow rate. Moreover, the hot water outlet temperature had to fulfil the requirement that the mean value over a whole cycle is 60°C, which is the maximal allowed CHP inlet temperature.

Table 2 –Calculation mesh of the TDC-block

	unit	min	max	step	points
Hot water	L/s	0.13	0.25	0.03	5
Cooling water	L/s	0.5	0.75	0.05	6
Cooling water	°C	25	30	1	6
Chilled water	°C	15	18	1	4
Half cycle length	s	800	2400	400	5
Total					3600

The two calculation meshes were then combined to find the optimized control strategy for given chilled water inlet and ambient temperatures. The task was done by a computer optimization tool which maximized the optimization criterion while simultaneously fulfilling the boundary conditions.

3.2. Optimization criterion

As optimization criteria the maximization of the cooling power and the primary energy efficiency was chosen.

The primary energy efficiency COP_{PE} was used since the CHCP system produces electricity and consumes natural gas. By referring to the primary energy a common base was set for these two kinds of energy. The primary energy efficiency is calculated as the amount of cold produced per unit of primary energy consumed whereas the electricity produced is computed as a bonus. According to EnEV 2009 the consumed natural gas has a primary energy factor of PEF_{gas}=1.1 J_{PE}/J_{gas} and the electricity in the grid has a primary factor of PEF_{grid}=2.6 J_{PE}/J_{el}. Further, the electric η_{el}=0.26 and thermal η_{th}=0.60 efficiency of the

CHP as well as a seasonal buffer storage factor of η_{BS}=0.85 is taken into account. Altogether the primary energy efficiency is calculated as

$$COP_{PE} = \left(\frac{PEF_{grid}}{COP_{el}} + \frac{PEF_{gas} + PEF_{grid} \cdot \eta_{el}}{COP \cdot \eta_{th} \cdot \eta_{BS}} \right)^{-1} \tag{1}$$

A detailed derivation can be found in (Schicktanz et al. 2011). The electrical coefficient of performance COP_{el} is calculated as the cold produced per unit of electricity consumed P_{el} in order to power the TDCs, the pumps and the cooling tower. It is assumed that the electricity consumption of the pumps is a function of the volume flow rate to the third power (Schicktanz et al. 2009). Therefore, the electricity consumption is

$$\begin{aligned} P_{el} &= 1000W \cdot \left(0.19 - 2.4 \cdot 10^{-2} V^{-1} \cdot U + 9.9 \cdot 10^{-2} V^{-2} \cdot U^2\right) \\ &+ 252 \frac{W}{kg/s} \cdot \dot{m}_{HT}^3 + 3732 \frac{W}{kg/s} \cdot \dot{m}_{MT}^3 + 180W \end{aligned} \tag{2}$$

where the first part is a fit function for the electricity consumption of the cooling tower as a function of the voltage signal U, followed by the hot water pump (HT), the cooling water pump (MT) and an offset for additional unregulated pumps.

Only maximizing the primary energy efficiency would lead to a very efficient operation mode that hardly produces cooling energy. An optimized control strategy should also maximize the cooling capacity if required. The product of the efficiency and the cooling capacity was thus chosen as optimization criterion. This product was maximized.

$$COP_{PE} \cdot \dot{Q}_C \rightarrow \max \tag{3}$$

This optimization criterion implies two assumptions: the heat production of the CHP is larger than the heat consumption of the TDC and the cold demand is larger than the produced cold. These assumptions hold for the system investigated. If one of this assumptions is not achieved the power term in the optimization criterion would be replaced. For a heat limitation the new criterion would then be as follows: Use the available heat as efficient as possible. For a cold demand lower then the available cold the criterion would be: cover the cold demand as efficient as possible. These two cases are not in the focus of this investigation.

4. RESULTS

Fig. 2 shows the maximized optimization criterion for a chilled water inlet temperature of 15°C or 18°C vs. the ambient temperature. The value of the optimization criterion drops almost linear with the ambient temperature. This means that at higher ambient temperatures the chillers should produce cold at a decreased power and lower efficiency. Moreover, at lower chilled water inlet temperatures the optimization criterion is lower.

The optimization criterion however, is not a common figure. Therefore fig. 3 shows the primary energy efficiency, the thermal and the electrical COP and the cooling power of the TDCs. The nominal cooling power of the TDCs is 5.5kW each. But due to the unusual series connection in the hot water and the chilled water loop 11kW can not be reached.

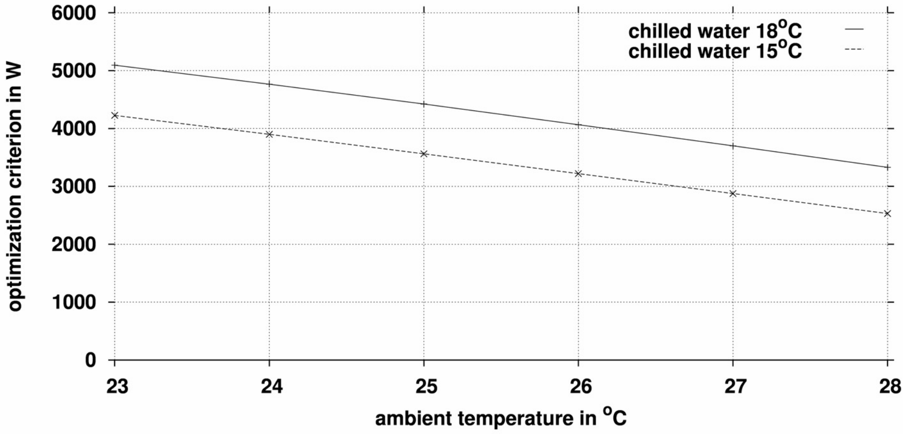

Figure 2 –Maximized optimization criterion at different ambient temperatures and for chilled water inlet temperatures of 15°C and 18°C

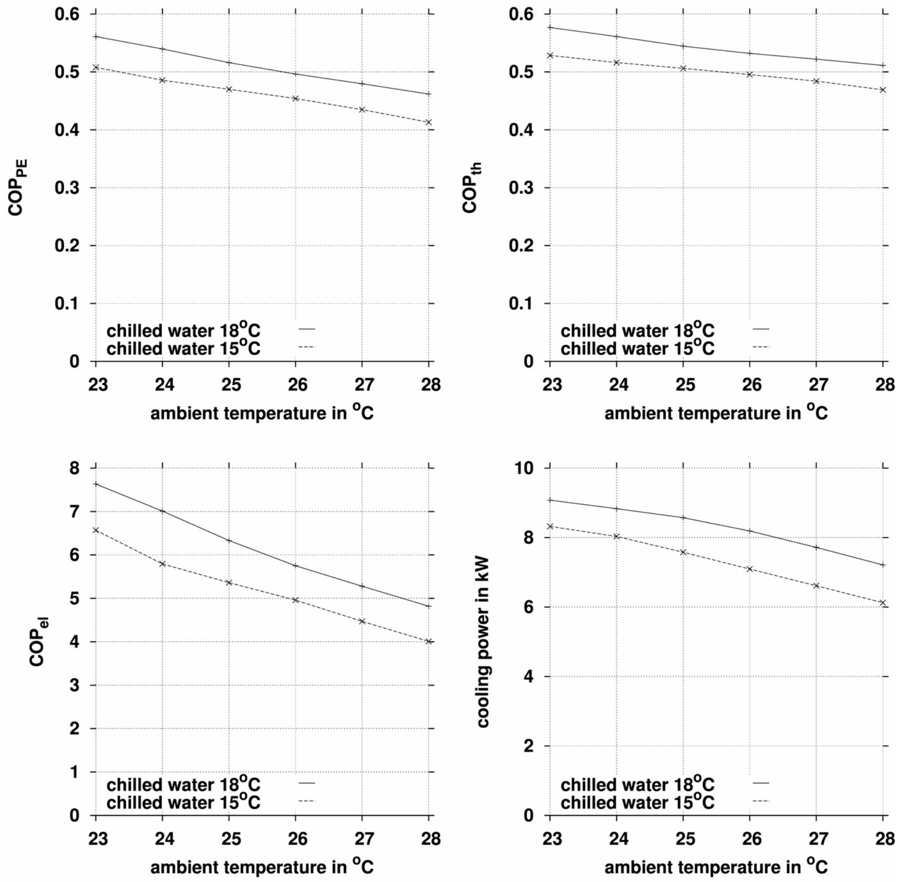

Figure 3 – Primary energy efficiency (COP_{PE}), thermal COP_{th}, electrical COP_{el} and the cooling power of the TDCs

For a chilled water inlet temperature of 18°C at low ambient temperatures 9kW can be achieved and 7 kW at high ambient temperatures. For 15°C chilled water inlet the cooling power is always about 1kW lower. The COP_{th} decreases from almost 0.6 to about 0.5 with higher ambient temperature and is always about 0.05 lower for a chilled water inlet temperature of 15°C. The COP_{PE} shows an almost similar profile like the COP_{th} shifted slightly to lower values. The COP_{el} reveals the high electricity demand of the cooling water loop. Starting at a value of 7.5 for a chilled water temperature of 18°C the COP_{el} drops three units at an ambient temperature increase of 5K. For a chilled water inlet temperature of 15°C the COP_{el} is always about one unit lower.

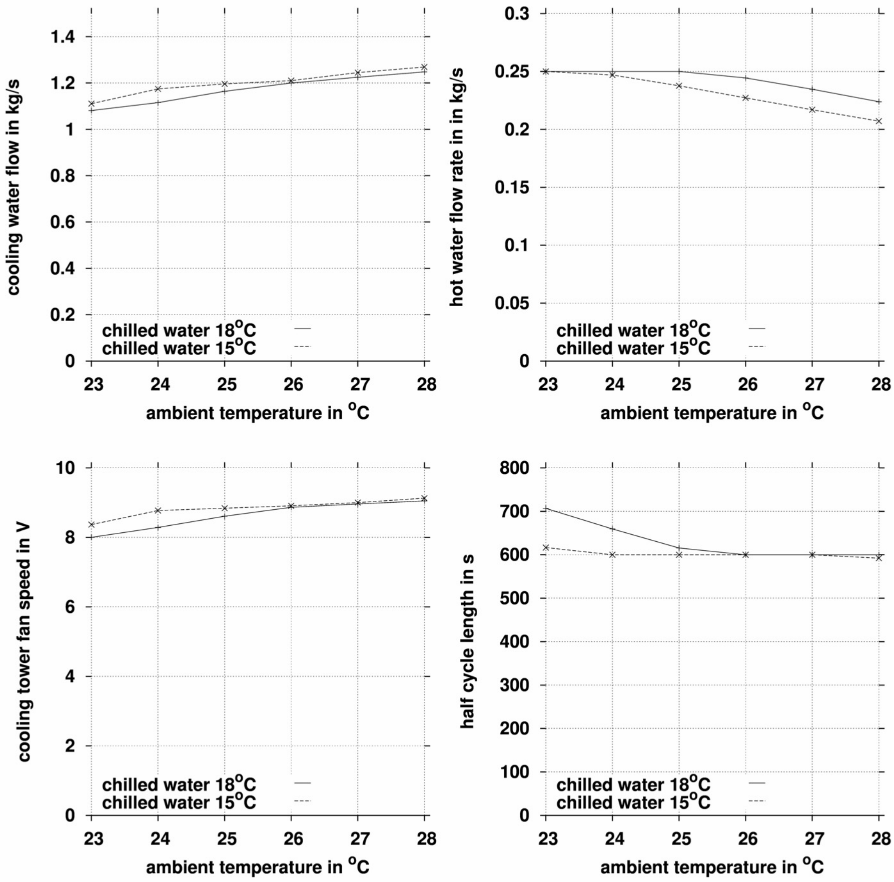

Figure 4 –Control strategy for the hot water and cooling water pump as well as the cooling tower fan signal and the half cycle length of the TDC for different ambient temperatures and chilled water inlet temperatures.

Fig. 4 shows the optimized control strategy for the individual components. With higher ambient temperatures the cooling water flow rate and the cooling tower fan speed should be increased. Although the changes in the control strategy look small the corresponding electric power is high as it was assumed that the flow rate in the tubes correlate with the third power to the electricity consumption. Increasing the cooling water flow rate by 15% increases the electricity consumption by more than 50%.

The optimization strategy recommends a half cycle length of 600s for the adsorption chillers at ambient temperatures above 25°C. In order to meet the boundary condition of a hot water outlet temperature of 60°C the hot water volume flow rate increases with rising ambient temperatures. For a chilled water inlet temperature of 18°C the maximum hot water flow rate is achieved for ambient temperatures below 25°C. To fulfil the criterion of 60°C in the hot water outlet pipe the adsorption half cycle length is increased. Half cycle length over 600s occur when the hot water mass flow rate reaches 0.25kg/s.

5. CONCLUSION

The question of an optimized control strategy of a CHCP system in cooling operation was analyzed and a possible solution was found for a particular system. For this task, a physical model of a cooling tower and an adsorption chiller was used. For both components a mesh of performance values for different operation conditions was calculated and both meshes were combined in order to take into account the boundary conditions and maximize the optimization criterion. The primary energy efficiency was chosen as an criterion to combine the different forms of energy to a single key figure. As optimization criterion the product of the primary energy efficiency and the cooling power was chosen. The new operation strategy requires the control of the TDC half cycle length, the hot and cooling water flow rate and the cooling tower fan speed.

NOMENCLATURE

COP	Coefficient of performance	*BS*	buffer storage
η	annual energy conversion factor	*C*	cold
$\dot{m}$	water mass flow rate, kg/s	*el*	electric
PEF	primary energy factor	*gas*	gas
T	temperature, K	HT	hot water
U	fan speed set point, V	MT	cooling water
		PE	primary energy
		th	thermal

ACKNOWLDEDGEMENT

The authors are grateful to the European Commission (grant no. TREN/06/FP6EN/S0764055/01988) for funding this work.

REFERENCES

1. EnEV 2009, Energieeinsparverordnung, url: http://www.enev-online.de/, German Energy Performance of Buildings Directive
2. Kumuda Rajgopal N.K., 2008, *Physical Simulation of a Dry Cooling Tower Using Modelica*, Master thesis, Fachhochschule Aachen (Dept. Jülich) and Fraunhofer ISE (Dept. Thermal Systems and Buildings)
3. PolySMART 2008, project web page http://www.polysmart.org/cms/upload/publications/Deliverables_PU_year_2/PolySMART_Deliverable_D3-24_SP4a_Assembled.pdf
4. Schicktanz M., Núñez T., 2008, Modelling of an adsorption chiller for dynamic system simulation. In: International Sorption Heat Pump Conference (ISHPC 2008).
5. Schicktanz M., Núñez T., 2009, Modelling of an adsorption chiller for dynamic system simulation. In: *International Journal of Refrigeration*, 32 (4), p. 588-595
6. Schicktanz M., Sondermann N., Wapler J., Núñez T., 2009, First Results of a Micro-CHCP system with two adsorption chillers. In: *Heat powered cycles conference*, 2009.
7. Schicktanz M., Wapler J., Henning H.-M., 2011, Primary Energy And Economic Analysis Of Combined Heating, Cooling And Power System. In: *Energy* 36, p. 575-585, doi:10.1016/j.energy.2010.10.002