Published in:
Proceedings of 4th Solar Air Conditioning Conference, Larnaka, Cyprus, 12.-14.10.2011

Heat rejection systems for solar driven sorption systems

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Abstract

Efficient heat rejection is crucial for the overall primary energy balance of sorption systems, as it dominates the auxiliary energy consumption. Low electrical COP´s of 3.0 or less are still common in sorption system, so that the primary energy efficiency is not significantly better than for conventional compression chillers.

A dry heat rejection systems requires electricity for fan operation, hybrid or wet cooling systems in addition need pumping energy for the cooling water and the water itself. The energy efficiency can be improved for heat rejection to the ground, where only pumping energy is needed for the geothermal heat exchange, however, ground heat exchangers are expensive.

Simulation models were developed for absorption and compression chillers. The chiller models were coupled to a three dimensional numerical ground heat exchanger model or to cooling tower models. All models were validated with operating data of different solar cooling systems.

The paper compares the improvement of primary energy efficiency for different heat rejection systems as a function of cooling load profile, climate and cooling technology and analyses the economics of the system.

Introduction

The performance of solar driven cooling systems strongly depends on the chosen heat rejection system and its control strategy(Kohlenbach, P. 2006). High electricity consumptions caused by suboptimal control in combination with low solar fractions through insufficient system design are critical for the environmental and economical performance of installed absorption cooling systems (ACM), especially if they are compared to highly efficient electrical driven compression chillers(Henning, H. M. 2004). To evaluate the overall efficiency of real installed solar cooling systems within the IEA TASK 38 (International Energy Agency Solar Heating and Cooling Programme) several solar cooling systems are monitored in detail. The results clearly demonstrate, that the electrical COP are still low with values of up to 6 in the best case and values of below 3 in the worst case. Primary energy ratio values obtained are 1.7 in the best case and values clearly below 1.0 in the worst case (Sparber, W. et. al 2009; Núnez, T. et. al 2009). For comparison, systems with good compression chillers with wet cooling tower for heat rejection reach average electrical system COP of 3.0 and primary energy ratios which are slightly above 1.0. The main reasons for the low system efficiencies of the analysed solar cooling systems are low solar thermal fractions and high electricity consumptions mainly due to the chosen heat rejection system.

Methodology

For the development of improved heat rejection systems detailed analyses on the effect of different control options on the primary energy efficiency are performed using a case study of a 15 kW chilii® solar driven absorption cooling system installed at the office building of the SolarNext AG in Rimsting, Germany. Detailed dynamic simulation models of the whole solar cooling system were developed and validated against measured performance data. The developed system models do not only describe the thermodynamic processes but also include the electricity consumption of the ACM, all pumps and of the heat rejection system. These models are used to analyse the effect of different control options of the solar cooling system on the overall system performance and the primary energy efficiency reached.

System Description

The analysed chillii® solar driven absorptioncooling system of the SolarNext AG has been set up and installed as a test facility to cool and to heat their office building in Rimsting, Germany (47.88°North, 12.33° East). The system includes a market available 15 kW chillii® ESC15 (EAWWGRACAL SE 15 LiBr absorption chiller), two 1 m³ hot water storage tanks, one 1 m³ cold storage tank, 37 m² CS-100F flat plate collectors and 34 m² TH SLU1500/16 solar vacuum tube collectors all facing south with an inclination of 30°, a 35 kW EWK wet cooling tower and an additional dry heat rejection system.For the distribution of the cooling energy chilled ceilings and fan coils are used with 16℃ supply and 18℃ return temperature and an automated supply temperature increase for dew point protection.

The cooling load of the single story office building with 566 m² of conditioned space is 8.9 MWh/a (16 kWh/m²a) and a maximum cooling load is 18 kW.

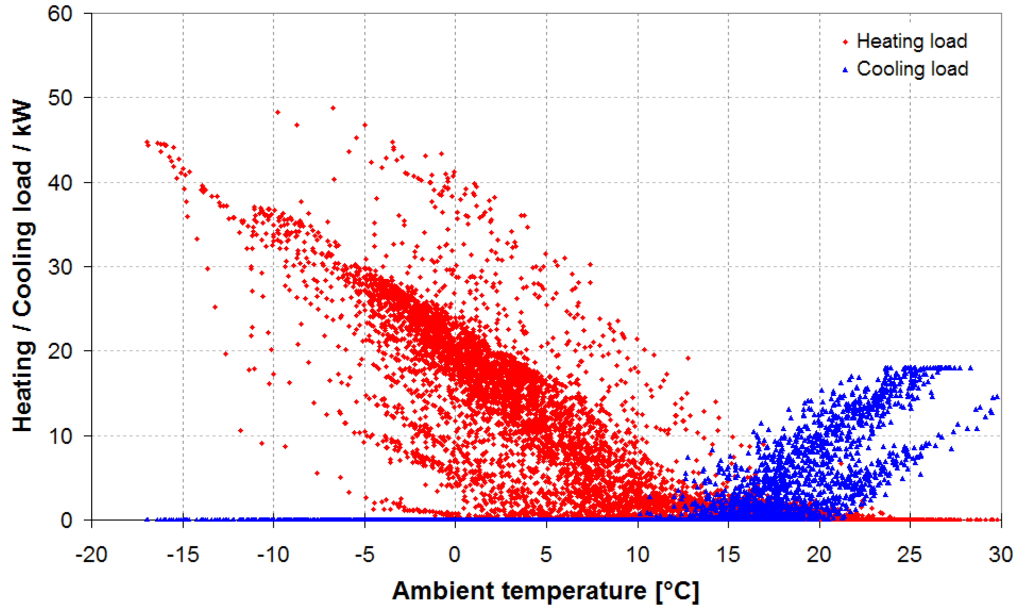

Figure 1: Ambient temperature dependent cooling and heating load distribution of the case study office building in Rimsting, Germany

The installed absorption chiller is able to provide the required cooling power of 18 kW if the temperature set point of the heat rejection system is reduced by 3 K from 30℃ design conditions to 27℃. To avoid stagnation in the collector circuit during periods of chiller shut down, a water to air cooler (KAMPMANN air heater TOP 474136) has been implemented in the system, which rejects the solar heat to the environment if the temperature in the collector return increases above 100℃. After activation the solar circuit cooler remains in operation until the temperature in the collector return decreases below 90℃. Table 1shows the installed pumps and their electricity consumption.

Table 1: Electrical consumption of main components of the absorption chiller system

Component description	Type	Nominal volume flow [m³/h]	Pressure drop [mbar]	Electrical power demand [W]

Absorption chiller with solution pump	EAW WEGRACAL SE 15			300
Wet cooling tower	AximaEWK 035	5.0 1420 m^{-1} fan rotation		330
Absorber / Condenser pump	Wilo-IP-E 40/115-0,55/2 3~ PN10 Wilo-VeroLine-IP-E	5.0	1200	550
Generator pump	High efficiency pump Wilo-Stratos ECO 25/1-5 PN10	2.0	250	56
Evaporator pump	High efficiency pump Wilo-Stratos 25/1-6 PN 10	1.9	350	52
Primary solar pump	WILO Stratos 30/1-12 PN 10	0.388	450	160
Secondary solar pump	WILO TOP-S 30/7	0.388	350	130
TOTAL				1578

The open wet cooling tower installed in the solar cooling system is an Axima EWK cooling tower with a nominal cooling capacity of 35 kW at 30℃ water supply and 36℃ cooling water return temperature at ambient conditions with a wet bulb temperature of 24℃.

The control of the cooling water supply temperature reduces assures a stable cold water temperature at variable generator inlet temperatures (Kühn, A. et. al, 2008 and Albers, J. et. al, 2009). For the overall system efficiency the increase of cooling water temperature makes only sense, if the electricity consumption of the cooling tower is significantly reduced through a fan speed control and no backup system is used for heat supply. A control of the cooling water temperature only through a three way valve is not recommendable in this case.

For systems with high temperature cold distribution (e.g. 16℃ supply temperature) the cold storage offers a useful storage capacity, if the produced cold water temperature is allowed to drop significantly below the cold water supply temperature setpoint at part load conditions. In this case generator inlet temperature control or evaporator mass flow control are not really required.

System simulations and validation

For the development and analysis of new innovative control strategies, a detailed dynamic simulation model of the installed system which also considers the electricity consumption of all installed components (fans, pumps, etc.) has been developed in the simulation environment INSEL (Schumacher, 1991). The component models used include dynamic models for the solar collectors and the hot and cold storage tank. No inertia is considered for the absorption chiller, the piping, the wet cooling tower and the dry heat rejection.

Measurement dataof the solar driven absorption chiller in summer 2007 was used to validate the developed simulation model of the installed system. A comparison of the simulated and measured outlet temperatures of the generator, condenser and evaporator of the ACM and of the collector field show that the performance of the installed system is well described by the developed simulation model (see Figure 2).

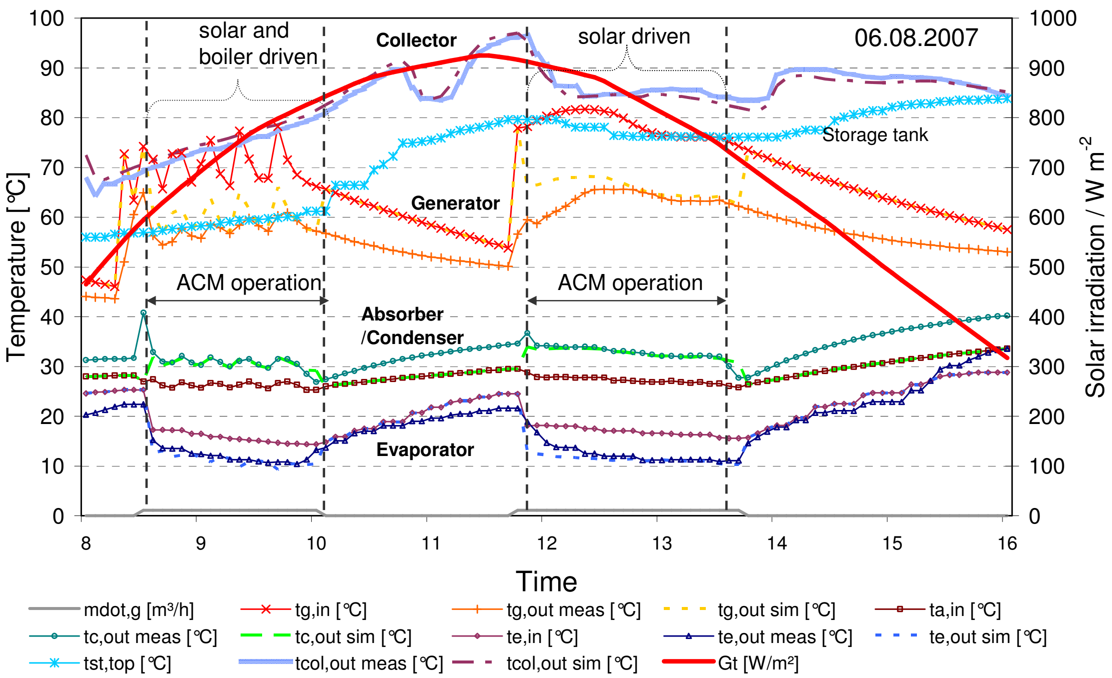

Figure 2: Measurements and simulation of the absorption chiller system with

The deviation between the predicted and measured solar heating power of the collector circuit (dynamic model) is below 1% for the analysed day. For the absorption chiller larger differences only occur at system startup and system shutdown, which is due to the omitted inertia of the absorption chiller in the model. Otherwise the deviation between simulated and measured generator and evaporator power is below 4% and below 3%for the heat rejection power. During start-up and shut down the deviation between measured and predicted performance increases to 16% for the generator, 12% for the evaporator and 10% for the heat rejection circuit of the absorber and condenser. These rather large deviations can be partly attributed to the fluctuations of the generator inlet temperature for the combined operation mode with auxiliary heating. These fluctuations result from a bad hydraulic integration of the auxiliary heater and a poorly controlled three way mixing valve in the original system setup (optimised during the heating period 2008 / 2009). If only the purely solar driven part is considered, the deviations between measurement and simulation are reduced to 12% for the generator, 10% for the evaporator and 6% for the heat rejection circuit of the absorber and condenser.

Analysed Control Options for heat rejection systems

The validated simulation model is used to analyse the effect of different control strategies for the installed ACM, the solar thermal system and the cold distribution on the overall performance of the solar cooling system. The analysed cases with different control options for either constant or variable generator supply temperature, with and without fan speed control of the cooling tower etc. are described in **Fehler! Verweisquelle konnte nicht gefunden werden.**.

The pumps of the solar collectors are considered as On/Off controlled in all cases. The collector pump is set in operation as soon as the collector temperature is 10 K above the temperature at hot storage bottom and is switched off again if the collector outlet temperature is 5 K above the temperature at hot storage bottom or if the temperature in the upper part of the hot storage increases above 95℃. The minimum operation time of the collector pump is set to two minutes. According to the system design the cold supply temperature was set to 16℃.To simplify the control, the cases with variable generator inlet temperature are without temperature control at generator inlet and constant evaporator mass flow rate. The ACM is turned off if the generator inlet temperature drops below 65℃ or the evaporator outlet temperature decreases below 6℃.

Figure 3: **Analysed control options of the absorption cooling system**

Analysed cases	Control options										
	Cooling tower			$t_{a,in}$			$t_{g,in}$			Cold dist. pump	
	Typ	3-way-valve	fan speed	27℃	24℃	21℃	90℃	70-90℃ variable	70-95℃ variable	ΔT-control yes	ΔT-control no
Case 1	wet	X		X			X				X
Case 2	wet		X	X			X			X	
Case 3	wet		X	X				X		X	
Case 3.1	wet		X		X			X		X	
Case 3.2	wet		X			X		X		X	
Case 4	dry	X		X					X	X	
Case 5	dry		X	X					X	X	

$t_{a,in}$ Absorber inlet temperature, either controlled by a 3-way-valve or by fan speed control of the cooling tower. Values below 27°C (30°C for dry cooling tower) are only provided as long as reachable at the given ambient conditions.

$t_{g,in}$ Generator inlet temperature, either constant or variable according to the temperature in the hot and cold storage tank.

An additional Case 6 has been defined and analysed as reference system for a compression chiller with a quite high electrical COP of 4.0 at 27 °C heat rejection temperature and high cold water supply temperatures. The electricity consumption for heat rejection and cold distribution is considered separately. The compression chiller is combined with a dry heat rejection system with constant fan speed.

Annual simulations were carried out to analyse the effect of the described control options on the overall system performance of the installed solar cooling system. For the meteorological conditions Meteonorm weather data of the location Rimsting in Germany was used with an hourly time step. For a correct consideration of the inertia in the solar thermal system and the storage capacity of the hot and cold water storage tanks, the internal simulation time step used was 1 min. A linear interpolation was used for the hourly values of the meteorological conditions and the cooling load of the building.

Simulation Results and Discussion

To compare the efficiency of the system with varied control strategies three different COP are used:

1. The standard thermal COP_{th};

2. The electrical COP_{el} which considers the electricity consumption of the ACM, the cooling tower and all pumps including the cold distribution pump;

3. The total primary energy ratio (PER) which is defined as the provided cooling energy divided by the sum of consumed electricity and additional thermal energy multiplied by the PEF factors of 2.7 for electricity in Germany (GEMIS) and 1.1 for the gas boiler:

$$PER = \frac{Q_{cool}}{Q_{el} \cdot PEF_{el} + Q_{h,add} \cdot PEF_{gas}}$$

The electrical performance of the system strongly depends on the fact whether the cold water distribution pump of the building and the ventilator of the cooling tower are controlled according to part load conditions or not. The electrical COPs vary between 6 and 11.5 for the cases with wet cooling tower and between 4 and 8 for the cases with dry heat rejection. The compression chiller system reaches an overall electrical COP of 3.2.

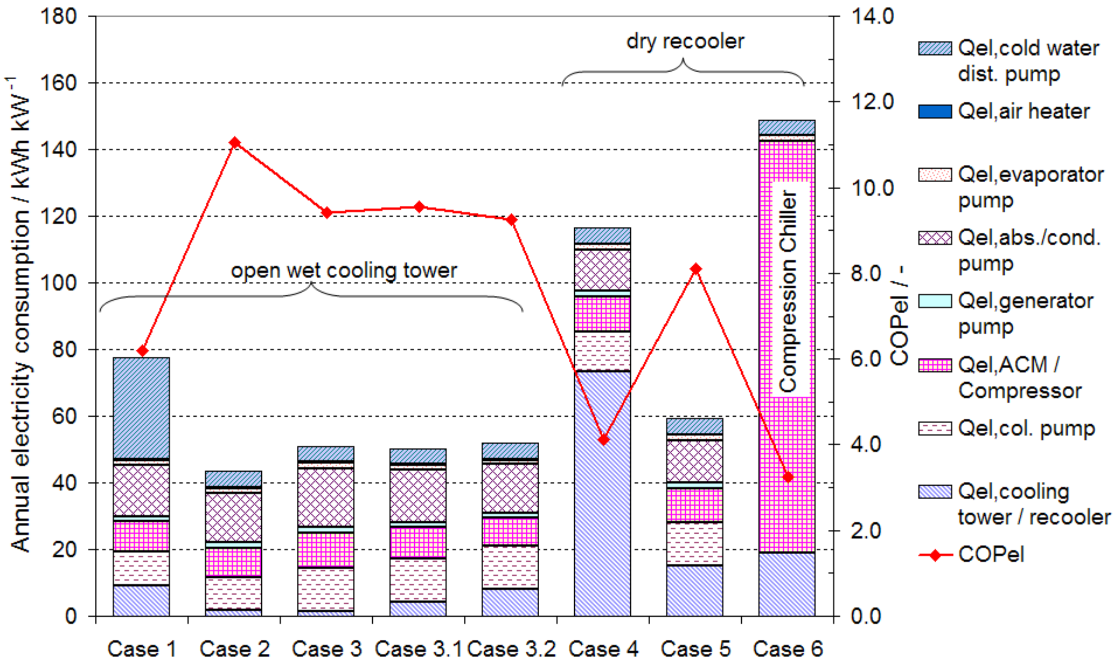

Figure 4: Annual electricity consumption and electrical COP

The lowest electricity consumption of 42 kWh per kW of cooling power and therefore the highest electrical COP of slightly above 11 is obtained for Case 2 with a controlled cold water distribution pump and cooling tower fan but with the absorption chiller operated at constant generator inlet temperature. If the generator inlet temperature is allowed to vary between 70℃ and 90℃ according to the temperature in the hot and cold storage tank(Case 3), the lower generator temperatures lead to longer operating hours of the solar system and of the chiller to provide the same cooling energy and therefore increases the electricity consumption. The thermal COP decreases very slightly from 0.75 in case 2 to 0.74.

However, at the same time the solar fraction is significantly increased from around 70% in cases 1 and 2 to 83% in case 3. A further increase of the solar fraction up to 88% can be achieved, if the heat rejection temperature set point is decreased from 27℃ to 24℃ in case 3.1 and 21℃ in case 3.2 (see Figure 5). The reduced heat rejection temperature set points lead to an increase in the electricity demand of the cooling tower due to higher fan speeds but at the same time reduce the operating hours of the whole cooling system due to the increased thermal COP and cooling capacity, which balances the additional electricity demand.

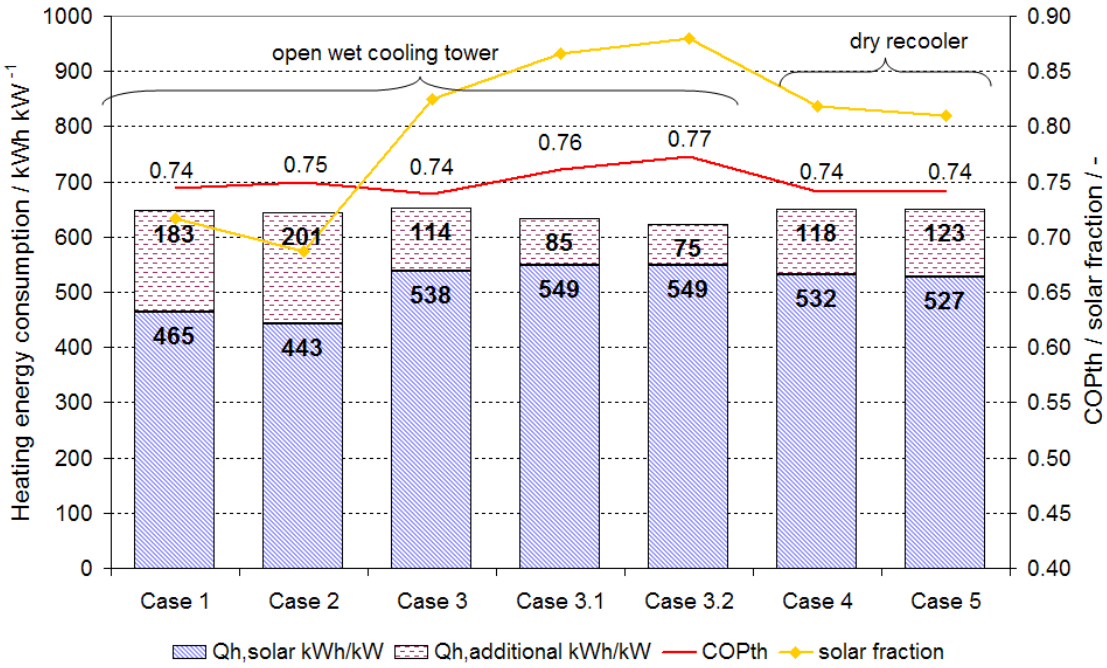

Figure 5: Heating energy consumption and solar fraction

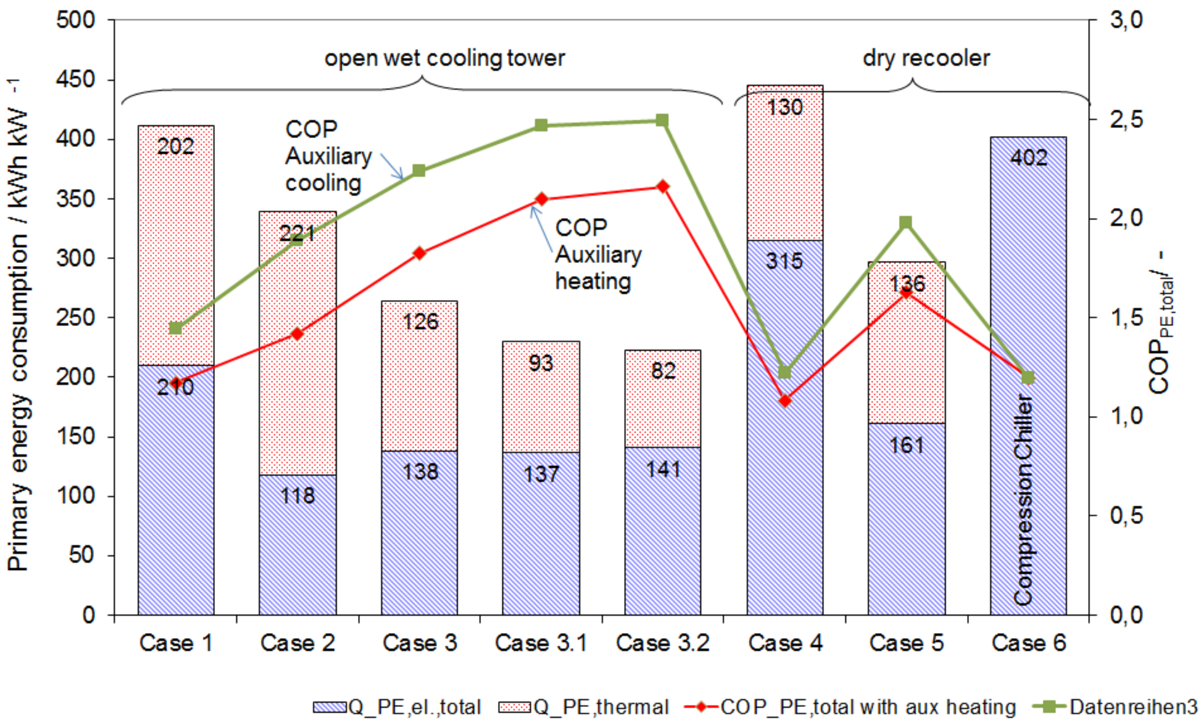

Figure 6: Primary energy consumption and primary energy ratio

The values of the primary energy ratio, which consider the electricity and additional heating energy consumption, vary between 1.1 and 2.2, with the lowest value for case 4 with dry heat rejection and without fan speed control and the highest value for case 3.2 with the lowest set point for the heat rejection temperature including fan speed control and a variable generator inlet temperature. The reference system with the compression chiller reaches a primary energy ratio of 1.1 which is nearly half of the value of the best absorption chiller case (Case 3.2) but already better than the worst absorption chiller case with dry heat rejection with constant fan speed (Case 4). This clearly indicates the importance of an energy efficient control and design of solar cooling systems and the requirement of further optimised hydraulic systems with reduced pressure drops and the utilisation of highly energy efficient pumps.

As visible from Figure 6 the additional heating energy consumption significantly increases the primary energy consumption although a high solar fraction of 70% and above is reached. This is due to the relatively low thermal COP of single effect absorption chillers. If the additional heating energy would be replaced by additional cooling provided by a highly efficient electrically driven compression chiller with a primary energy ratio of 1.1 (including electricity energy consumption of pumps and heat rejection system) 20 to 30% higher primary energy ratios can be reached with a maximum of 2.5. In this case the absorption chiller only provides cooling energy as long as sufficient heating energy is provided by the solar system and the remaining cooling energy is provided by the compression chiller.

Optimisation Potentials

The remaining main electricity consumers for case 3 with optimised control are the absorber and condenser pump 37%, the collector pumps 28%, and the absorption machine itself 23% (mainly solution pump). In order to further improve the overall efficiency of the solar cooling system the electricity consumption of these three components needs to be reduced significantly. For the cooling circuit of the absorber and condenser attempts need to be made to further reduce the pressure drop of the heat exchangers and of the spray nozzles (alternative distribution system) of the open wet cooling tower. A reduction in electricity consumption of at least 30% could be reached by these measures.

For the solar system the electricity consumption could be significantly reduced by up to 50% and more, if the solar system would be operated with pure water. Such a system is e.g. provided by Paradigma in Germany. In this case the pressure drop and heat losses of the heat exchanger and the secondary collector pump could be avoided completely. However, in regions with danger of frost a special frost protection control and additional temperature sensors need to be implemented in the system. If danger of freezing is detected by the control system from the temperature sensors, the control system switches the collector pump

ON for a short time in order to pump warm water from hot storage bottom into the collector field. According to recent analyses on a large solar cooling installation at FESTO in Esslingen the heating energy losses caused by frost protection of the annual solar energy production of the collector field are only about 3% in the best case (Dalibard et al, 2009). However, these additional losses are partly equalised by the higher efficiency of the pure water system, which does not require a heat exchanger between collector circuit and hot water storage tank.

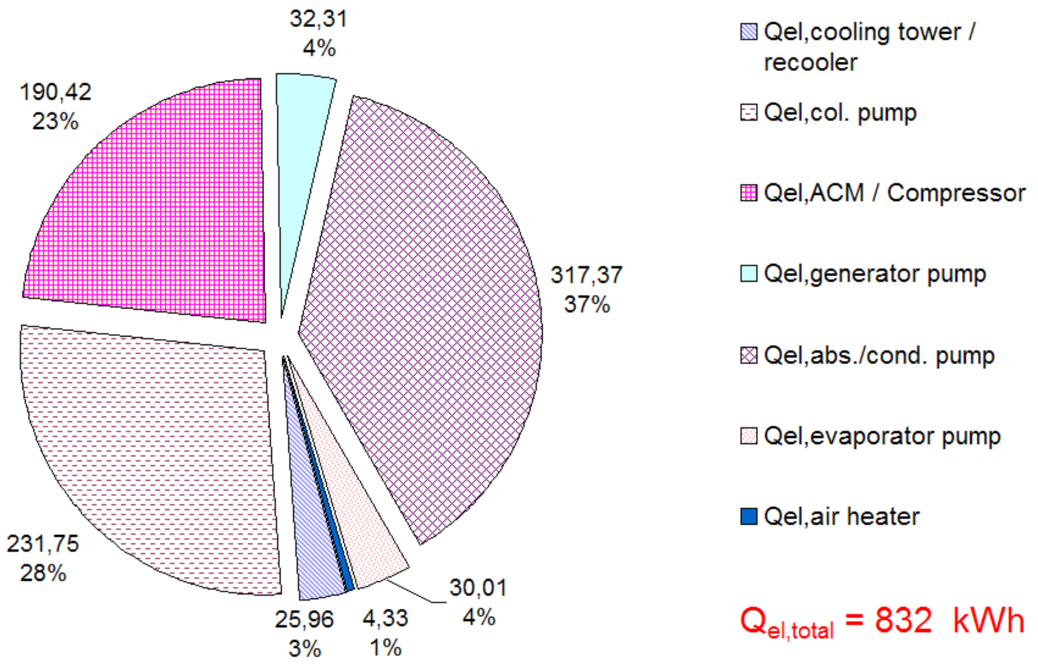

Figure 7: Electricity consumption of Case 3

For the absorption chiller, the electricity consumption is mainly caused by the integrated water and solution pumps. Due to the actual system design of the EAW WERACAL SE 15 with two containers, the absorber and generator are mounted at the same level. Therefore, two solution pumps are required, one for the concentrated and one for the diluted solution. Currently a new system design is developed by EAW in cooperation with the ILK in Dresden which integrates all components in one container with the generator and absorber mounted on different height levels (Weidner, G., 2009). For this new design only one solution pump is required, which reduces the electricity consumption of the absorption chiller by at least 30%. Considering all reduction potentials the electrical COP of case 3 could be improved to a value of 13. This would result in an increase of the primary energy ratio of over 20% to a value of 2.6 for auxiliary heating and to 3.1 for auxiliary cooling.

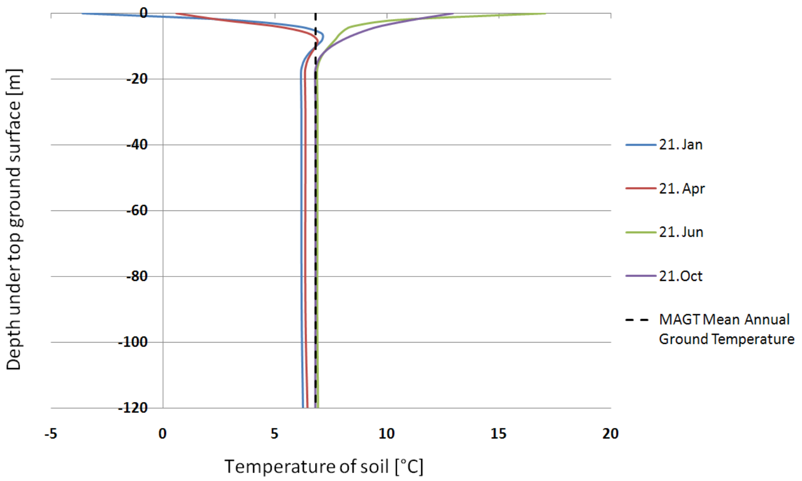

Figure 1: Development of seasonal ground temperature levels after five years of operation in summer heat rejection and winter heat pump mode. The maximum loads for heating and cooling of the office building are equal, the full load hours are higher in winter operation.

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