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SIMULATION BASED OPTIMIZATION OF DYNAMIC POWER CONTROL FOR SMALL CAPACITY CHILLERS

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Abstract

The control strategy is essential for successful implementation of solar thermal cooling plants. One possible solution is a dynamic power control of the chiller capacity - especially if part load conditions have to be considered.

When part load is included in results and should be obtained by simulations, the model of the chiller has to be able to give realistic results for mass flow and temperature variations of the external circuits. Such a model was introduced for an ammonia/water chiller and was used to set up a holistic simulation model in TRNSYS.

The system configuration is based on real cases, which show good performance during operation. First simple control approaches indicate promising results. Compared to simple ON/OFF- control strategy 40% primary energy can be saved by using the dynamic power control. This control strategy includes the variation of mass flow rates of generator and recooling circuits and also the cooling tower fans with a simple linear modulation. Further results using different strategies will be presented.

Keywords: solar cooling, control strategy, primary energy efficiency

Tab. 1: Nomenclature and subscripts

Quantity	Symbol	Quantity	Symbol
Absorption chiller	ACM	Medium temperature	MT
Building	room	Non-renewable energy	NRE
Cold backup	CB	Output of a system	out
Cold distribution	CD	Part Load Indicator	PLI
Cooling	C	Primary energy conversion factor	ε
Cooling tower	CT	Primary Energy Ratio	PER
Cooling tower ventilator	CTvent	Return	re
Domestic hot water	DHW	Savings	fsav
Dynamic power control	DPC	Seasonal Energy Efficiency Ratio	SEER
Electrical	el	Seasonal Performance Factor	SPF
Energy	Q	Secondary (cold) storage	SS
Fan coil	FC	Solar fraction	SF
Heat pump	HP	Space heating	SH
Heating, ventilation and cooling	HVAC	Setpoint	set
High temperature	HT	Supply	su
Input to a system	in	System	sys
Low temperature	LT	Thermal cooling	thC

1. Introduction

Following the building´s thermal energy demand, HVAC systems often have to operate under part load conditions. For an improvement in primary energy efficiency of these systems hardware components and control strategies must be designed accordingly. The control strategy accounting for part load behavior of certain components is a crucial point.

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Regarding the primary energy efficiency of solar thermal cooling plants both the type of backup and the part load efficiency have to be considered. While the influence of backup devices (hot or cold, e.g. natural gas boiler or vapor compression chiller) is well understood in its basics and furthermore part of ongoing research, different aspects of control strategies should be investigated more in depth. In this study only configurations with cold backups are analyzed. Effects on the primary energy demand of hot backups have been studied in (Henning et al. 2013) or (Neyer 2013).

There are two basic ways to consider part load operation. One is to use cold water storages and simple ON/OFF-strategies. In this case load and production are separated and more or less independent, but can lead to high standby losses and higher investments. Dynamic power control (DPC) is a second option, preferably in case the absorption chiller (ACM) is the main and dominant cooling device. Dynamic power control includes variable mass flow rates and/or variable temperatures at the external loops of the absorption chiller. Main objective is to regulate the actual cooling power of the chiller in order to match the actual cooling load of the building. The involved mass flow rates (hot and/or medium temperature (HT/MT) side and/or air of the cooling tower (CT)) are dynamically set depending on the current cooling load, which can be indicated through the return temperature of the chilled water circuit. Particular limits and eligibility criteria for the appropriate choice of one of these strategies, but also different possibilities of dynamic control for different profiles and system configurations should be investigated.

A few plants were built and are in operation using different strategies of dynamic power control. Two of these plants were monitored, analyzed and optimized during the course of the Austrian research project SolarCoolingOpt (SCopt 2010). Both configurations use an ammonia/water chiller (Pink chiller, type PC19). Main system components beside the chiller are hot temperature flat plate collectors, a heat rejection unit and a hot water tank. No cold water storage is used; the chilled water loop supplies the thermally activated ceilings directly. If there is a change in the building cooling load, the system reacts accordingly and the supply temperature and mass flow rate of the LT-circuit is changed respectively. Measurement data shows that since beginning of operation the plants achieve a monthly electrical energy efficiency ratio (Seasonal Performance Factors - SPF_{el}) of around 6. Based on further monitoring and data analyses several improvements were found and implemented. Now these solar thermal cooling plants work at daily SPF's of greater than 8 (Nocke 2014). In comparison average solar thermal cooling plants reach a SPF_{el} from 3 to 7 or even lower (Wiemken 2013).

Based on these practical experiences theoretical analyses are carried out in the course of the Austrian research project solarhybrid (solarhybrid 2014). Main focuses for the optimization of thermally driven systems are the different possibilities of control strategies using non-linear modulation and different combinations of mass flow and temperature control. This paper focuses on the difference of ON/OFF controlled to dynamic controlled system.

2. Modelling

To test the above mentioned ideas detailed system simulation models with different configurations regarding the system design as well as control strategies were set up in TRNSYS (TRNSYS 17.1). Mainly standard Types were used for the solar collector, hot/cold water storages, cooling tower, the building and controllers. For simplification no pipes were included, the vapor compression chiller is set up as function using an SEER of 2.8 (average value defined in IEA SHC Task 38 (Sparber et. al, 2009)) and iterative feedback controllers were used.

The crucial component in the model is the absorption chiller. For these applications an ammonia/water absorption chiller with small capacity (19kW cooling power) was applied. This chiller was surveyed in detailed steady state and dynamic laboratory test. A physical model was built up and was used to create the data base for a simplified lookup-table model (Hannl 2012). This model was implemented in TRNSYS and was extended by dynamic and start/stop behavior of the chiller. The TRNSYS type (Type 1002) can be used to simulate various control strategies using variable mass flow rates. The model provides the chiller's performance curves as a function of 6 input variables (3 temperatures and 3 mass flow rates). Including the dynamic effects and the wide range of data (temperature and mass flow rates) realistic results can be obtained.

Out of simulation and monitoring data of existing plants it is known that the chilled water supply temperature (T_{LTsu}) is often lower than the set temperature (T_{LTset}). That indicates that the chiller power under such conditions is higher than necessary. This leads consequently to frequent ON/OFF switching of the chiller. When using cold water storages the number of start and stops can be reduced using two temperature sensors in the tank. This is already an improvement of the standard ON/OFF strategy and was implemented in the simulations. A high frequency of ON/OFF switching actions can be avoided in systems with dynamic power control.

The control signal for the power modification follows different linear and non-linear approaches. If T_{LTsu} is less than T_{LTset} the generator (HT-) and the recooling (MT-) mass flow rates as well as the cooling tower ventilator are reduced accordingly to the new developed Part Load Indicator (PLI). The degree of reduction is determined by different equations, which were compared (equation 1 and 2).

$$PLI_{(1/x)} = \frac{1}{T_{LTset} - T_{LTsu} + 1} \quad \text{(eq. 1)}$$

$$PLI_{(q)} = \frac{T_{LTre} - T_{LTset}}{T_{LTset} - T_{LTsu}} \quad \text{(eq. 2)}$$

The resulting PLIs are shown in Fig. 1 for given boundary conditions (T_{LTset} = 6°C, T_{LTre} = 9°C). The PLI is decreasing the greater the difference of the set temperature and actual temperature is. The maximum of the PLI's is limited to 1. This signal can either be used directly to reduce the driving speed of the above mentioned components or in combination with an appropriate controller to reach the desired set temperature.

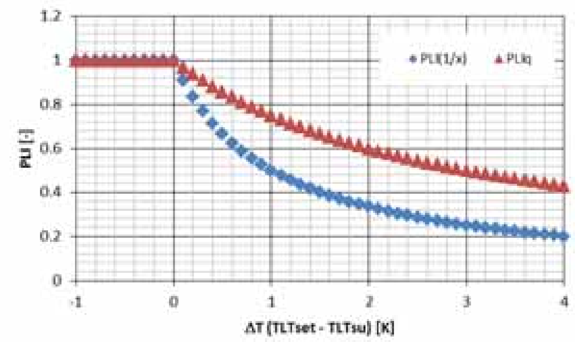

Fig. 1: Part Load Indicators (@ T_{LTset} = 6°C, T_{LTre} = 9°C)

The main components and the layout of the model are based on the existing solar cooling plant in Gleisdorf/Austria (Vukits et al 2012). The reference building is a three-story office building with an area of 900 m². The simulations also include the heating period, but the focus is on cooling mode. There is no hot water demand in this profile. The cold and hot delivery systems in the building are fan coils operating at (for solar heating and cooling (SHC) systems) unfavorable cooling design conditions of 6/9°C. The nominal power of the chiller matches the maximum cooling load of the building (approx. 21 W/m²). Main components of the simulated plant are:

- 19 kW ammonia/water chiller
- 65 m² flat plate collector
- 2000 l hot water storage
- 500 l cold water storage (LT-storage) is equipped with 2 temperature sensors atop and bottom for the ON/OFF control of the absorption chiller.
- Profile: office building, 900 m², climate of Vienna (Meteonorm data), approx.40 kWh/m².a heating demand and 10 kWh/m².a cooling demand
- The building cooling load was chosen in order to reach roughly 60% solar fraction in the cooling season.
- Cold backup (vapor compression chiller with an SEER of 2.8) in series, assures that the T_{LTset} is always reached.
- Hot backup is only used for space heating in winter season.
- Free floating fan coils / dehumidification is possible, but uncontrolled.

The system configuration is summarized in Fig. 2.

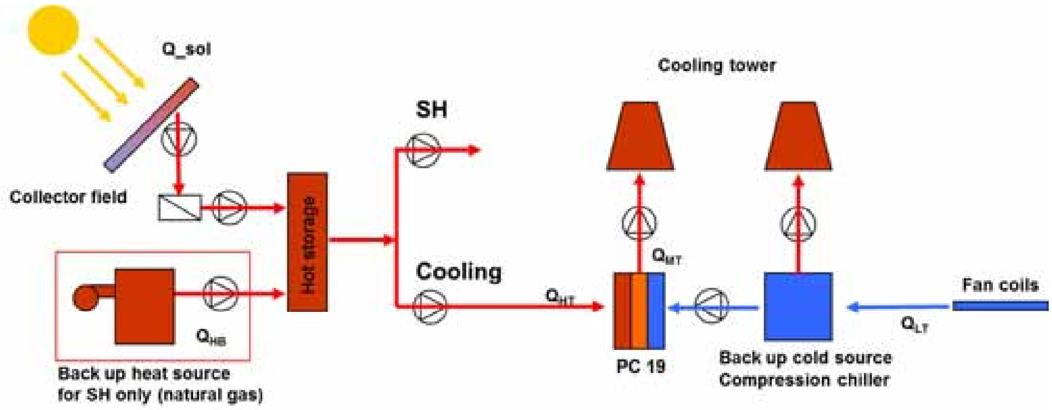

Fig. 2: System configuration for space heating (SH) and cooling of the office building in Vienna, without cold water storage (PLI controlled); Thermal backup is only used for space heating.

The system model is designed as coupled building and HVAC simulation. The interaction between building and refrigeration / heating is considered and a standard controller is used to operate the fan coils. The fan coils turn ON/OFF, when the min./max. temperature in the building is exceeded. This procedure follows a predefined hysteresis (24/26°C). Fan coils are fed primarily by the hot water storage in winter and cold water storage in summer.

In the optimized version the absorption chiller is directly fed by the returning water mass flow from the fan coil and a feedback controller regulates the air volume flow within the fan coils. The usage of the hot backup to drive the cooling process is prohibited. The solar primary and secondary loop are speed controlled to reach the required generator temperature in the storage as fast as possible. The chiller switches on, when the hot water temperature in the storage is above 85°C. The considered controller strategies are summarized in Table 2 below.

Tab. 2: Control strategies and parameters

<table>
<tr><th></th><th>HT</th><th>CT_{vent}</th><th>MT</th><th>LT</th><th>FC</th></tr>
<tr><td>ON/OFF</td><td colspan="4">constant mass flows / speed, ON/OFF @ T hot storage top 85/65°C AND T cold storage (2 sensors) 8/9°C,</td><td rowspan="2">Const. mass flow T_{room} ON/OFF 26/24°C</td></tr>
<tr><td>ON/OFF – $DPC_{(MT, CTvent)}$</td><td>same as ON/OFF</td><td colspan="2">Variable speed, feedback controller T_{MTset} 25°C</td><td>same as ON/OFF</td></tr>
<tr><td>$DPC_{(q)}$</td><td colspan="3">Variable flow, $PLI_{(q)}$*mass flow or fan speed On if T_{room} >25.5°C</td><td rowspan="4">Variable speed ΔT_{LT} = T_{LTre}-T_{LTsu} = 5K</td><td rowspan="4">Variable flow $T_{room\text{-}set}$ 26°C</td></tr>
<tr><td>$DPC_{(q-cf)}$</td><td colspan="3">Variable flow, feedback controller, $PLI_{(q)}$ set = 1, On if T_{room} >25.5°C</td></tr>
<tr><td>$DPC_{(1/x)}$</td><td colspan="3">Variable flow, $PLI_{(1/x)}$*mass flow or fan speed On if T_{room} >25.5°C</td></tr>
<tr><td>$DPC_{(1/x-cf)}$</td><td colspan="3">Variable flow, feedback controller, $PLI_{(1/x)}$ set = 1, On if T_{room} >25.5°C</td></tr>
</table>

3. Assessment

Following the IEA SHC Task 48 (Task 48/B7) technical and economic key figures were defined to analyze and compare different solar heating and cooling system among each other and with reference systems. A number of standards were analyzed and a coherent nomenclature and definition of performance figures was developed. In this paper the focus is on the technical figures concerning the performance of the thermal cooling and its backup systems. A clear definition of the boundaries for the calculation has to be drawn. As there is no parallel usage of cooling, domestic hot water or space heating in this study, the boundaries can be defined in a simple way. For cooling there is only a differentiation of thermal cooling (thC) and overall cooling (C) including the cold backup. The focus is on following key figures.

- **Seasonal Performance Factor** (SPF) for the assessment of the (sub-)system performance is including all auxiliary components under defined, time dependent rating conditions over a certain period of time. The SPF can generally be defined as the ratio of useful energy output to energy input with respect to a given system boundary (thC or C). As there is no thermal backup the focus is on the electrical efficiency and the SPF_{el} is calculated as follows:

$$SPF_{el} = \frac{\sum Q_{i.out}}{\sum W_{el.i.in}} \qquad \text{(eq. 3)}$$

- **Primary Energy Ratio** (PER) gives more in-depth information under the economic or environmental point of view. It is defined as the ratio of the useful energy output to the primary energy input to the system boundary. To be able to calculate it, certain primary energy conversion factors for every type of energy input have to be provided (eq. 2). Here only non-renewable energy is accounted.

$$PER_i = \frac{\sum Q_{i,out}}{\sum\left(\frac{Q_{el,i,in}}{\varepsilon_{el}} + \frac{Q_{i,in}}{\varepsilon_{in}}\right)} \qquad \text{(eq. 4)}$$

  The primary energy ratio is also calculated for a reference system ($PER_{ref.i}$). All primary energy factors and the proceeding to calculate the reference follows the Task48 procedure (Task48/B7).

- In order to compare the renewable SHC system with a reference the **fractional savings** (f_{sav}) can be used. The non-renewable primary energy savings ($f_{sav\text{-}NRE.PER}$) in comparison to a reference system then can be calculated as follows.

$$f_{sav.NRE.PER} = 1 - \frac{PER_{NRE.ref}}{PER_{NRE.i}} \qquad \text{(eq.5)}$$

- The **equivalent Seasonal Performance Factor** (SPF_{equ}) can be used to compare the investigated solar heating and cooling system with a reference vapour compression chiller or a reversible heat pump based on the electrical seasonal performance factors SPF_{el_i} and SPF_{ref} respectively. The SPF_{equ} (eq.4) can be calculated following the unit conversion:

$$\mathrm{SPF}_{equ.i} = \frac{\mathrm{PER}_{\mathrm{NRE,i}}}{\varepsilon_{\mathrm{el}}} \qquad \text{(eq.6)}$$

  Same SPF_{equ}'s indicates finally an equal primary demand of different systems. In this case the $SPF_{equ.i}$ and $SPF_{el.i}$ are the same because only electricity and no thermal backup energy are inputs to the thermal cooling system.

4. Results

All results and key figures are discussed only for the cooling season from mid of May to mid of September. Due to direct coupling of the solar cooling plant to the building and the room control, the delivered cooling energy differs. The amount of delivered energy ranges from ca. 7'000 kWh to almost 9'000 kWh per year depending on the set points and types of controllers. The implemented ON/OFF- controller in the building allows a wide range of temperatures (24/26°C) in which the maximum comfort conditions are still respected. When changing to a direct coupled PLI controlled cooling system a continuous control for the building is necessary. For all variations no comfort problems are occurring (room temperature is always less than 26°C).

Table 3 shows the most important energy flows in the systems with respect to different control strategies. The table includes the delivered cold to the building ($Q_{CD.sys}$), the energy delivered to the cold storage (SS) by the cold backup ($Q_{SS.CB}$) and by the absorption chiller ($Q_{SS.HP}$). The electricity demand of all auxiliary components (pumps, fans, standby, etc.) for thermal cooling is summarized in $Q_{el.thC}$. The overall electricity demand includes the cold backup.

Analyzing the data in Tab. 3 indicates that the ratio of usage of the cold backup is also strongly depending on the control strategy. Under the boundary conditions of this study the ratio ($Q_{SS.HP}/(Q_{SS.HP}+Q_{SS.CB})$) can be interpreted as solar fraction (SF_C). The solar fraction affects the primary energy ratio of the overall system (PER_C) and its savings accordingly. The fractions vary from around 40% up to 65%.

Tab. 3: Main energy flows for the different control strategies

	$Q_{CD.SYS}$ (kWh)	$Q_{SS.CB}$ (kWh)	$Q_{SS.HP}$ (kWh)	SF_C (-)	$Q_{el.thC}$ (kWh)	$SPF_{el.thC}$ (-)	$Q_{el.C}$ (kWh)	$SPF_{el.thC}$ (-)
ON/OFF	7'195	3'123	4'072	0.57	562	7.24	1'788	4.02
ON/OFF – $DPC_{(LT, CTvent)}$	7'034	2'947	4'087	0.58	514	7.95	1'604	4.39
$DPC_{(q)}$	7'337	2'656	4'681	0.64	606	7.72	1'549	4.74
$DPC_{(q-cf)}$	8'688	5'098	3'590	0.41	353	10.15	2'155	4.03
$DPC_{(1/x)}$	7'461	2'997	4'464	0.60	610	7.31	1'683	4.43
$DPC_{(1/x-cf)}$	8'374	4'526	3'848	0.46	415	9.27	2'036	4.11

The first configuration "ON/OFF" shows a $SPF_{el.thC}$ of 7.2 and an overall $SPF_{el.C}$ (including the backup) of 4. 30% of primary energy would be saved in this configuration. Changing to the variable recooling circuit (mass flow rate and cooling tower ventilator – $DPC_{(LT,MT)}$) increases the primary energy saving for 20% although the $SPF_{el.thC}$ only increases by 10%.

Switching to dynamic controlled systems (DPC) increases the $SPF_{el.thC}$. Maximum values of roughly 10 can be reached. This equates to a plus of 40%. If $PLI_{(q-cf)}$ is used, the highest sub system performance can be achieved. Maximum primary savings occur with a $PLI_{(q)}$ controller. Again the influence of the solar fraction appears with a low SPF_C. The results of the key figures are summarized in Figure 3.

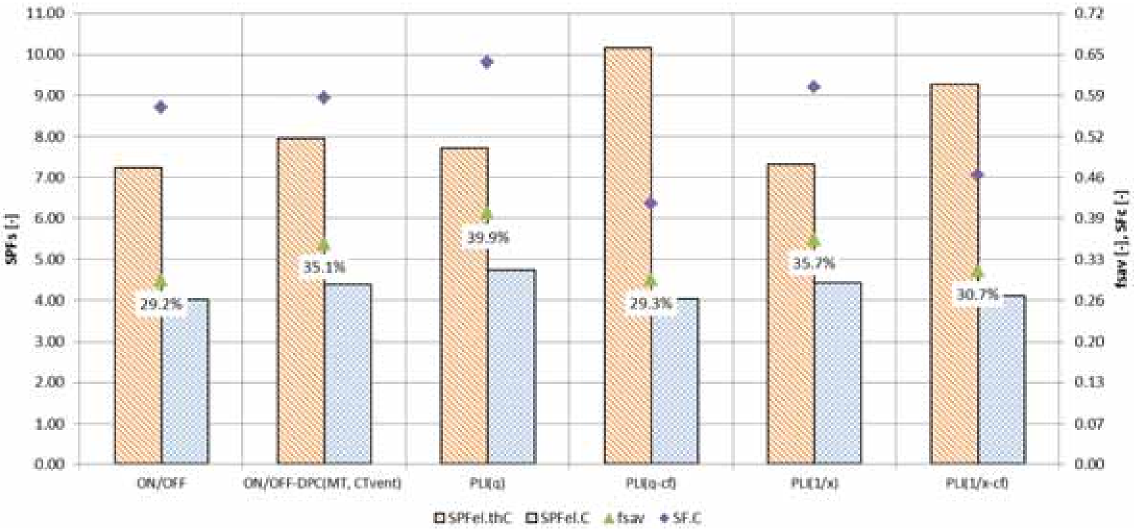

Fig. 3: System configuration for space heating (SH) and cooling of the office building in Vienna

Due to the PLI controller and its continuous adaption the running hours of the systems increase and the electricity demand for some pumps are rising. Main advantage is the reduction of the major electricity consumers of the recooling circuit including both the pump and the fan of the cooling tower. The chilled water pump is the distribution pump in case of dynamic coupled systems. The auxiliaries for distribution are not counted.

Figure 4 indicates the relative electricity demand (left axis) of each single pump/fan/chiller and the absolute electricity demand (right axis) for the thermal cooling including pumps/fans/chiller ($Q_{el.thC}$) of the cold backup ($Q_{el.CB}$) and overall electricity of the system ($Q_{el.sys}$).

The recooling circuit is responsible for ca. 50-60% of the electricity demand. A relative huge part of 30% or higher refers to the ammonia/water chiller. The solar, hot and chilled water pumps account for the rest. The absolute demands reflect the electrical seasonal performance factors in Figure 3. The lower the $Q_{el.thC}$, the higher the sub system performance is. But if then the cold backup is needed more often the overall performance can get lower.

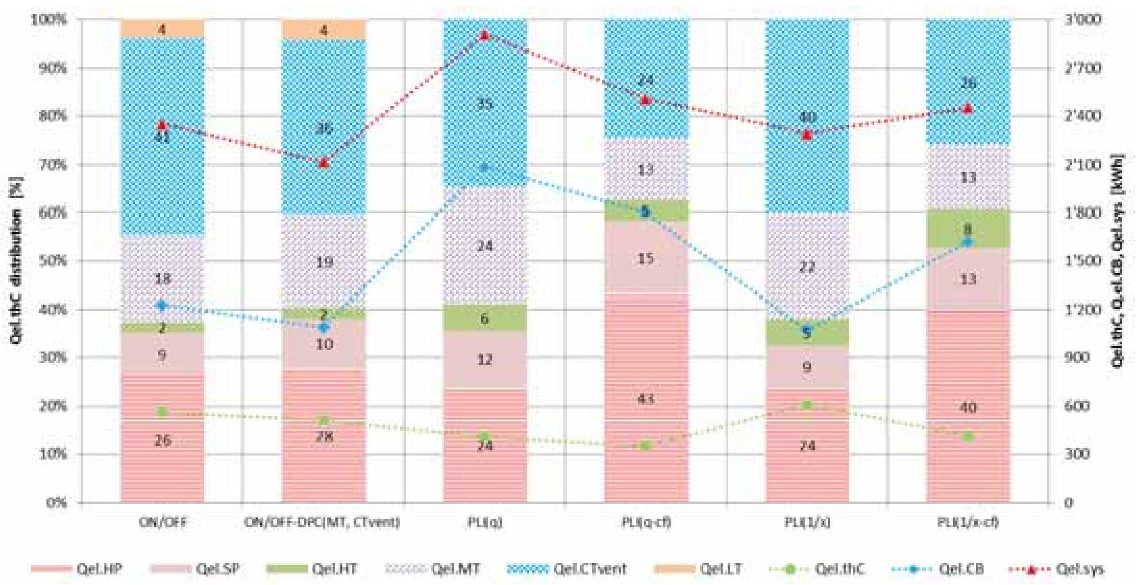

Fig. 4: Resulting relative electricity demands for thermal cooling (left axis) and absolute consumptions for thermal cooling ($Q_{el.thC}$), cold backup ($Q_{el.CB}$) and sum ($Q_{el.sys}$) (all right axis)

5. Summary / conclusions

Different control strategies have been tested for one case of application (office) with a moderate solar fraction (ca. 60%) related to cold production. The studies have been carried out with a holistic HVAC simulation model in TRNSYS. The building is directly coupled to the solar heating and cooling system.

The results show that there is a huge influence in the amount of cold energy delivered and on the overall performance of the system including the cold backup. As shown in Figure 3 the dependency of building controller is rigorous. On the one hand this shows the potential of the building and its controller respectively. The building controller can be optimized and the absolute energy can be reduced. On the other hand it can be seen how important it is to approach optimization on the overall system level. Even when the sub (solar) system is efficient the overall result can be inferior. The approach does not call in question when focusing on the system performance.

The integration of the cold backup (vapor compression chiller) in series to the absorption chiller and as simplified equation in the simulation does not allow any deviation from the set point. Any deviation from T_{LTset} is accounted as immediately needed backup and the solar fraction gets lower. A more realistic model should be set up and include the part load behavior of the backup chiller and an appropriate controller.

The overall primary energy savings need a proper backup strategy even with a cold backup. The system performance of the backup is assumed at a level of SEER 2.8. Consequently high solar fractions are necessary to get towards high savings. Either way efficient sub systems are crucial and originate from proper design and control strategy but are no guarantee for high overall performance.

Comparing the sub system efficiency ($SPF_{el.thC}$) of ON/OFF and dynamic control shows efficiency potentials up to 40%. This difference is depending on the control strategy and its adaptivity to the building load. But in almost the same manner the advantage of dynamic controllers is depending on the integration of the cold backup and its model. Small variances in the annual electricity demands (100 kWh) account for the differences and all boundaries. Input variables should bear in mind.

Finally the advantage of dynamic power control in sub system performance could be shown even if the overall savings were not conclusive. Clear limits of the usage of either ON/OFF or dynamic controllers need to be elaborated. PLI and other dynamic controller will be implemented, analyzed and optimized in the ongoing Austrian research project solarhybrid (solarhybrid 2014).

Acknowledgement

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