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Towards a Solar Hybrid Solution for Heating and Cooling

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Abstract

Solar electrical and thermal driven systems are promising for a sustainable supply of heat and cold demand. The target of the Austrian research project SolarHybrid is the optimization of both. Thermal and electrical systems are combined in solar hybrid systems to utilize advantages of each (sub-) system. The evaluation of the steps toward the hybrid solution including system design and control strategies, are performed with detailed component models. The dynamic simulation studies are supplemented with Hardware-in-the-Loop laboratory measurements.

Initial design optimization for photovoltaic and solar thermal driven system include dimensioning, control strategies and energy cascade usage. Cost savings will be achieved by common used components e.g. for the electrical and thermal driven chillers. A demonstration of such arrangements is realized with specifically designed and adapted chillers in the laboratory and investigated under dynamic tests. The controllers get further enhanced by mathematical methods and the implementation of predictive control.

The so far optimized solar thermal driven systems can achieve up to 60% non-renewable primary energy savings with a cost ratio of 1.1 compared to a simple reference system with natural gas boiler and vapor compression chiller. If an electrical driven system with a reversible heat pump is applied as benchmark, savings remain at 50%, but the cost ratio drops to 0.9. Further improvements and comparisons of the photovoltaic and solar thermal driven systems as well as the hybrid solutions will be performed in detail within the course of the project.

Keywords: *solar hybrid, solar heating and cooling, simulations, DHW preheating, Hardware-in-the-Loop*

1. Introduction

Solar technologies are promising for a sustainable heat and cold supply. An increase of solar cooling systems is observed by Mauthner et al (2016). Solar includes both technologies solar thermal and photovoltaic driven systems. Several IEA Tasks are related to this topic, one of them is IEA SHC Task 53. The objective is to assist PV and solar thermal driven systems to a sustainable market development (http://task53.iea-shc.org/).

Several studies of the previous year's show the competitiveness of solar thermal and solar electrical driven systems. The advantage of each system configuration depends on the boundary conditions. Different boundary conditions try to carve out these advantages. Magnitudes of efficiency or rather primary energy savings and costs can be equal for both technologies.

These controversial statements are taken up by the Austrian project SolarHybrid. The major target of SolarHybrid is to develop and evaluate economic, efficient and reliable solar hybrid systems. Hybrid includes solar electric (PV) and solar thermal supported heating and cooling systems in different combinations. Effective solar hybrid systems can be realized if the underlying base technologies are

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optimized and when they are adapted for a certain kind of applications.

In a first step both technologies were enhanced regarding the economical and energetic efficiency. The potentials have to be utilized before solar hybrid systems are examined. Only when optimized single technologies are used, solar hybrid systems lead to success. Both technologies are optimized by the mean of optimization tools. The optimization of the single systems is a basis for the hybrid ones. An automatic optimization of the control strategy supports the improvement of different parameters.

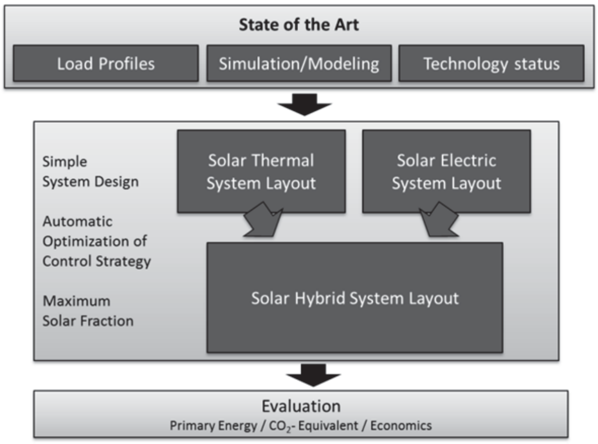

Fig. 1: Methodological approach and workflow of SolarHybrid

Optimization and development of the solar hybrid system is based upon the same methodology. Dynamic simulations are built up, taking a minimized number of components and enhanced schemes into account. The systems and their control strategies are designed using automated algorithms. The complex minimization problems lead to ideal efficiency. Applying these minimized schemes with combined used components, the solar hybrid systems allow interesting economic alternatives. The performance of this development will be proven in hardware-in-the-loop laboratory measurements.

2. Steps towards a solar hybrid solution

In this section all main steps carried out in the course of SolarHybrid are described including ideas, methodologies, highlights and results.

2.1. Assessment

The assessment follows the method of IEA SHC Task 48 and Task 53 (Neyer et al. 2015b). The main figures are non-renewable Primary Energy Ratio for the entire system (PER_{NRE}) and a pre-defined reference system (PER_{ref}). Derived from the comparison of these PER's, non-renewable Primary Energy Savings ($f_{sav\text{-}NRE}$) are calculated. The two main primary energy conversion factors used in this assessment are electricity (ε = 0.4 kWh_{el}/kWh_{prim}) and backup fuel (natural gas with ε=0.9 kWh_{end}/kWh_{prim}).

For the economic evaluation the annualized costs for investment, replacement & residual value, maintenance, energy and water costs are calculated with pre-defined values, representing cut off values derived in the Tasks. Levelized costs of energy (space heating-SH, domestic hot water-DHW, cooling-C and domestic electricity) are derived for both, the entire system and the reference system. The ratio of these levelized costs of energy (CostRatio) is used for assessing and presenting the effect of the simulation studies.

The standard reference system consists of a natural gas boiler as heat source and a vapor compression chiller as cold source. For further comparison a specific reference system was defined. The system boundary for technical and economic analysis includes all energy demands of the entire study: space heating, domestic hot water, pool heating, cooling and domestic electricity.

- Seasonal Performance Factor (SPF)

The SPF is the ratio of useful energy (out: energy supplied to the application) to energy effort from external sources. It is distinguished between thermal and electrical performance.

$$SPF_{th} = \frac{\sum Q_{out}}{\sum Q_{in}}, \; SPF_{el} = \frac{\sum Q_{out}}{\sum Q_{el,in}} \quad \text{(eq. 1)}$$

- Non-renewable Primary Energy Ratio (PER_{NRE})

The PER converts all energy inputs of the system into primary energy equivalents. This provides appropriately comparable quality ratings for energy derived from alternative electricity, solar and fossil fuel heat energy sources. The PER_{NRE} is also calculated for the entire reference system.

$$PER_{NRE} = \frac{\sum Q_{out}}{\sum\left(\frac{Q_{el}}{\varepsilon_{el}} + \frac{Q_{EC}}{\varepsilon_{EC}}\right)} \quad \text{(eq. 2)}$$

- Fractional Saving (f_{sav})

f_{sav} represents the non-renewable primary energy saving due to the entire SHC system compared with a reference system.

$$f_{sav-NRE} = 1 - \frac{PER_{ref}}{PER_{sys}} \quad \text{(eq. 3)}$$

- Cost Ratio (CR)

The cost ratio is determined with the levelized costs of energy for the SHC system ($C_{tot.SHC}$) and the levelized costs of the reference system ($C_{tot.REF}$). The levelized costs (EUR/$kWh_{useful\ energy}$) are calculated including the sum of the annualized costs (invest, replacement, maintenance, energy, etc.) and the delivered energy flows of the application.

$$CR = \frac{C_{tot.SHC}}{C_{tot.REF}} \quad \text{(eq. 4)}$$

2.2. Simulation model adaption

Crucial models in the system analysis are the component models for the vapor compression- and the absorption chiller. An ordinary effort was prosecuted to reach reliable and realistic results especially in part load conditions. Beside the chillers, other component models (e.g. cooling tower) were analyzed and adapted.

- Vapor compression chiller

Three types of compression chillers ((1) on/off screw, (2) on/off double scroll, (3) part load controlled turbo compression) are taken into account. For each compressor type a suitable simulation model was set up. Operating conditions, which remain uncovered by manufacturer data, were extrapolated using the Thin-Plate-Spline method. Fig. 2 shows this limitation due to the operation range and the limitation due to the available data.

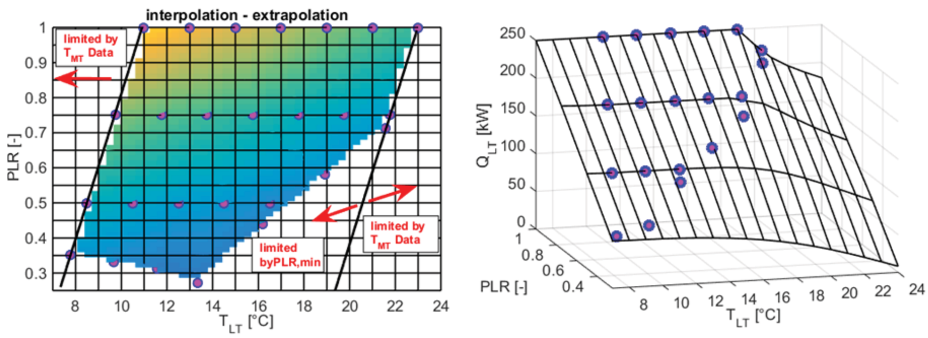

Fig. 2: Interpolation and extrapolation of boundaries for the vapor compression chiller for the turbo compressor (PLR…part load ratio, LT…chilled water, MT…cold water)

Several publications (e.g. Fillard 2009, Bettanini 2003) suggest modeling the part load energy efficiency ratio (EER) of compression chillers as function of the part load factor (PLF). While this approach correlates with the performance map for the chiller #1, major deviations from the part load behavior of the other models (especially the turbo-chiller) could be observed.

For the double compressor chiller (chiller #2) the PLF-model is valid at constant heat rejection temperature within an error interval of 1%. Thus the chillers have been modelled by a normalized lookup-table containing the performance data.

Dynamic effects were modeled by adding a thermal capacity to the output and taking the start-up time of the chillers into account. For the model of the turbo compression chiller (chiller #3), the minimum part load due to the operating range of the compressor have to be considered. Lower cooling capacities have to be provided by on/off switching.

- Absorption chiller

In case of hybrid systems, solar thermal cooling can represent a good alternative to electrical driven chillers. Therefore the influence of the usage of ammonia/water absorption chillers is investigated by simulations and assisted by real measurements of an appropriate chiller prototype. For simulations the physical models were developed for a single- and half-effect chiller (SE- and HE-chiller) and were used to create the data base for a simplified lookup-table model (Hannl 2012). Both lookup-tables are used by a newly developed TRNSYS Type (Type 1005). The data provided by the lookup-tables are interpolated accordingly. The model reflects the SE-chiller's performance curves as a function of 6 input variables (3 temperatures and 3 mass flow rates). The HE-chiller is currently implemented with 3 independent temperature input variables, volume flows are constant (Tab. 1). The mode (HE or SE) can be switched manually. Either single mode can be used in the simulations or a proper control strategy can be implemented according to the actual requirements of the entire heating and cooling system.

Tab. 1: Matrix for the characteristic curve diagrams for the SE- and HE-chiller

Parameter	Unit	SE	HE
T_{HT}	°C	75 – 80 – 85 – 90 – 95	75 – 80 – 85 – 90 – 95
T_{MT}	°C	24 – 28 – 32 – 36	25 – 28 – 31 – 34 – 37 –40
T_{LT}	°C	6 – 9 – 12 – 15	-6 – -3 – 0 – 3 – 6 – 9 – 12
$\dot{V}_{HT}$	m³/h	1 – 2 – 3 – 4 – 6	2 (GEN1 und GEN2)
$\dot{V}_{MT}$	m³/h	1 – 1,5 – 3 – 4.5 – 6 – 9	5/4/3 (ABS1, ABS2, CON)
$\dot{V}_{LT}$	m³/h	1 – 2 – 4.5 – 7	3.5

The model provides a very realistic start/stop-behavior and includes the electricity demand of the solvent pump. The chiller is scalable due to the integrated normalized data lookup-table. The Type is validated by steady state and dynamic measurements of the absorption chiller prototype at different operation modes.

2.3. System concepts

The simulations are set up in TRNSYS & MATLAB and are based on a load file concept. This concept enables the separation of building- and HVAC-simulation.

- Building profile

The building-simulation was set up in TRNSYS with a four-star Hotel (Gritzer 2016). This hotel has a capacity of 240 beds and an area of 10'080 m². Internal loads and geometry are based on different standards (e.g. SIA, etc.). The south oriented building has a high quality thermal envelope. Shading is achieved by construction and active shading elements. The hotel includes the following usage zones: accommodation, reception, lobby, bar / restaurant, kitchen and spa area with a pool. Geometry, design and control strategies determine the load profiles and energy demand for space heating, pool heating, domestic hot water and cooling accordingly. The profiles are defined on a daily basis and change monthly. Ventilation includes heat recovery. Air change rates and specified temperatures correlate with each usage zone. Depending on the boundary condition, heat and cooling loads arise for each usage zone. These assumptions result in the following details for the Innsbruck climate profile, which is investigated in detail here.

Tab. 2: Energy demand and loads for the hotel profile in Innsbruck

Application	Demand (MWh)	Load (kW)
Space heating	271.2	190
Pool heating	766.1	208
Domestic hot water	562.5	260
Air conditioning	85.7	80
Domestic electricity	517.5	

Demand and different system configurations are evaluated separately by focusing on the profile of interest and neglecting the others.

- Thermal driven systems

The scope of the simulation study with the solar thermal driven heating and cooling systems is to survey following points as listed in Table 3.

Tab. 3: Matrix for the characteristic curve diagrams for the SE- and HE-chiller

Main Problem	Explanation / Examples
(1) System design issues	Optimization of the solar thermal heating and cooling system due to design and control strategies. Comparisons of PV driven systems simultaneously. E.g. variable collector area, hot /cold water storage tanks, ACM and VCC capacities depending on a variety of load files (pool off, SH off, cooling 100%, DHW 100% and 25%).
(2) Cascade energy use	The temperature level of re-cooling of the ACM and VCC enables possibilities of usable energy cascades. E.g. rejected heat for DHW pre-heating
(3) Common used components	Commonly used components can be forced for economic reasons, but also when refurbishing system common used components enables potential E.g. Serial or parallel use of ACM and VCC in MT- and LT-circuits
(4) Multiple use of components	Solar parts, but also the thermal driven chillers are main investment cost drivers. The multiple usage of these and other components should increase the economic competitiveness. E.g. ACM used as heat pump, combined with low temperature heating/cooling

Selected results for the (1) step are e.g. the optimization of the DHW only solar thermal systems (ST) and the comparison with PV driven heat pump (HP) systems. The solar thermal driven system is shown in Figure 3. The solar collectors feed the hot storage, back upped with a natural gas boiler. The applications for the first studies are reduced to DHW and cooling. The solar energy driven absorption chiller is running in parallel to a vapor compression chiller. Both are keeping the cold storage tank at a certain temperature to fulfill the needs of the load profile.

For the design of the ST system two different hot storage sizes are considered: 30 and 50 l/m² solar thermal collector area. The size of the solar collector is designed that 10 h of stagnation are observed. The DHW consumption is scaled from 100% down to 25% in four steps. The smaller tank volume combined with a moderate collector size (0.5-1.5 m²/person, depending on DHW demand) was chosen to be best for further applications. Solar fractions of 20-25% are reached at cost ratios of 1.25-1.1 for the standard reference system.

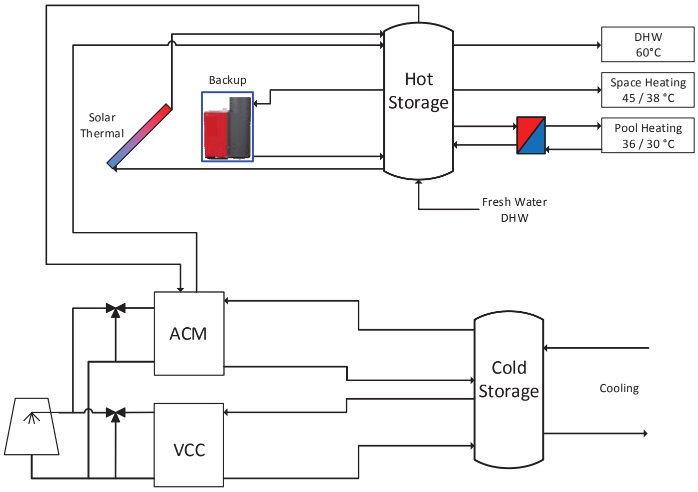

Fig. 3: Solar thermal system layout with hot and cold backups

Compared to a simple heat pump, a cost ratio of 0.82 can be reached. Because of the low solar fraction the non-renewable primary energy savings reach 17%. Hence, the solar collector area was increased to reach a cost ratio of 1. Finally, a collector field of 720 m² was determined with a f_{sav} of 0.39 compared to the heat pump. If the heat pump is equipped with a PV, 140 kW_p would be necessary to reach the same primary energy savings as the thermal driven system. Simultaneously the cost ratio drops down to 0.7. Main reason is the DHW application at its 60°C set temperature that enforces a low EER of the HP system.

With these findings a second simulation study, including the cooling demand, was started. Result of this study is an optimized control strategy and a system design. Solar fractions of 55% can be obtained with this configuration. Compared to the standard Task 53 reference primary energy savings of 49% and a cost ratio of 1.11 can be achieved. The optimization is e.g. reflected in the auxiliary electricity demand of the system.

In Figure 4 the carpet plots of the electrical efficiency SPF_{el} are shown and include the annual trend (x-axis: days of the year, y-axis: hours per day, color of each pixel: SPF_{el}). The maximum value of thermal cooling has a SPF_{el} of 21.5 and the vapor compression chiller of 5.7. The maximum for the thermal cooling efficiency appears simultaneously with small loads whereby the chiller is driven with hot water stored in the tank (mainly at night). The maximum for the vapor compression chiller appears during minimal loads. On an average, the electrical efficiency of absorption and electrical cooling results in a value of 4.

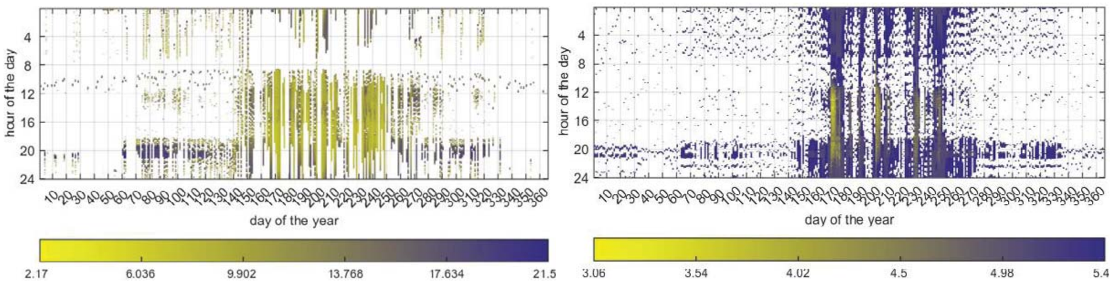

Fig. 4: Carpet plots of the electrical efficiency for the thermal cooling- (left) and vapor compression chiller (right)

The so far optimized solar heating and cooling system was used to integrate ideas of cascade energy usage (2). The heat rejection of the chillers (absorption and electrical) with temperatures around 25-40°C enables the use of this energy flux for DHW pre-heating (Fig. 5). The heat exchangers for the fresh water (DHW) pre-heating are situated in the cooling tower circuits. Thus, the hydraulic of each rejection circuit has to be extended with one mixing valves (V_{MT}) and one bypass (V_{CT}).

The first valve is used to keep a certain set point for the chiller re-cooling inlet temperature. This set point can change according to the different control strategies of the system (e.g. chilled water set temp., etc.). If the outlet temperature of the cooling water after the heat exchanger (HX) drops below the set point, the cooling tower will be bypassed.

In Table 4 all results for DHW only and DHW&C are presented for the total DHW demand (1) and for the smallest demand (0.25). Accordingly, the ratio of C/DHW changes and affects the results.

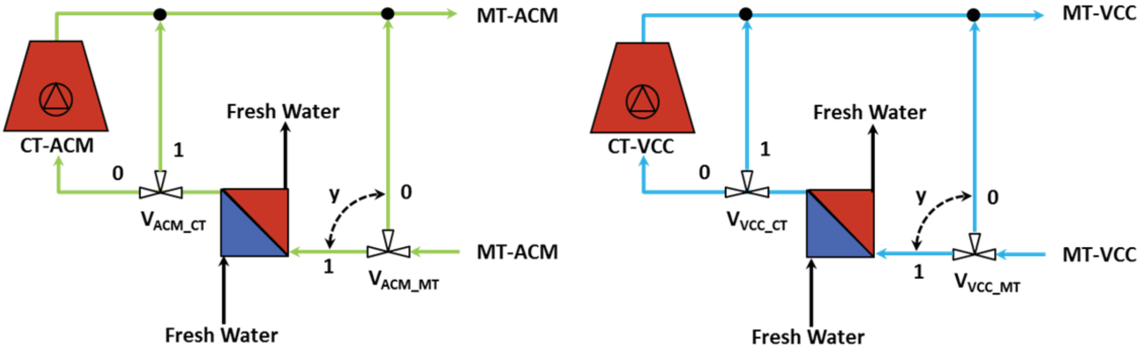

Fig. 5: DHW-pre-heating in the heat rejection circuits of the ACM (left) and VCC (right)

When DHW pre-heating is activated, the fractional savings increase from 49% to a level of 53% (DWH=1) and from 55% to 57% at low DHW demands respectively. At the same time the Cost Ratio decreases from 1.11 to 1.09 (DHW=1) and from 1.44 to 1.43 (DHW=0.25). This can be explained when the energy balance for the hot storage is disposed. All energy used for preheating, as it is arranged in continuous flow mode, reduces the affordable energy input. When the system is compared to the specific reference system (HP), the savings arise to 50% with a corresponding CR of 0.91 (65%/1.4 for DHW 0.25).

Tab. 4: Comparison of achievable cost ratio (CR) and primary energy savings (f_{sav}) for different applications of solar thermal systems

Results	Additional information	DHW only		DHW&C		DHW&C preheating	
Collector area	m²	420	131	720	431	720	431
DHW load	-	1	0.25	1	0.25	1	0.25
Solar fraction	-	0.25	0.19	0.55	0.75	0.55	0.74
f_{sav} – T53	Ref: natural gas boiler & VCC	0.25	0.19	0.49	0.55	0.53	0.57
CR – T53		1.10	1.25	1.11	1.44	1.09	1.43
f_{sav} spec.	Ref: heat pump	0.17	0.11	0.46	0.63	0.50	0.65
CR spec.		0.82	0.88	0.93	1.41	0.91	1.40

The differences in the savings and the cost ratio between highest and lowest DHW demand, derive from the specific collector area and the ratio of investment to overall costs. The 25% DHW is equipped with a greater collector area. Thus solar fraction, primary energy savings, costs and the cost ratio increase accordingly.

- Electrical driven systems

Considering the high electric loads, the integration of PV-panels or PVT-panels can lead to a decrease of primary energy consumption. Depending on the amount of low temperature thermal energy demand and on the internal heat transfer coefficient of the panel PVT or PV are used. Especially the internal heat transfer is critical for the low temperature thermal efficiency and improvement of the electric efficiency of the PVT compared to a PV panel. The PVT panels are simulated by using the model of Bertram (2012). This model takes the effective solar cell characteristic for the electrical part and the thermal model from EN 12975 (now EN ISO 9806) into account.

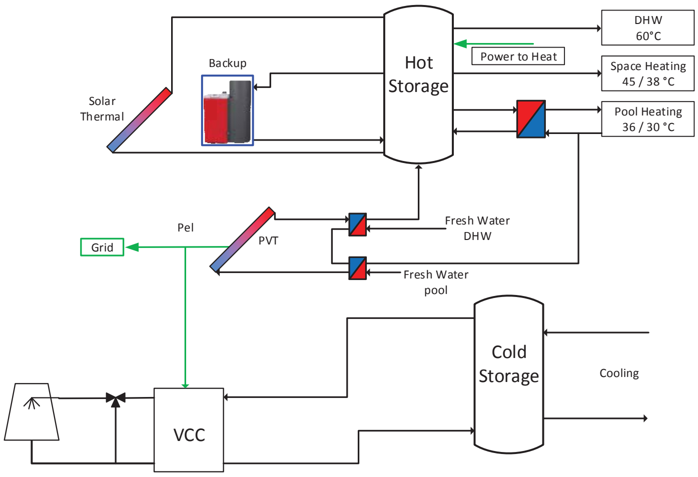

Fig. 6: Integration of PVT panels for DHW pre-heating or pool heating

The available load profile of the hotel contains a high demand of freshwater for DHW and the pool at low temperatures (around 12°C). Thus different feed in points for PVT are studied (Fig. 6). Regardless of the decision for PV or PVT, the produced electric energy is a dimensioning factor for the amortization (ecological or economical). Power to heat can only be an option if excess electric energy is produced and variable feed-in tariffs are considered. In this case it can be better from an economical point of view to store the energy in the system.

Other components, such as chillers, could also benefit from this low temperature energy demand as sink for rejection heat. This energy exchange is addressed at first in the design process. Secondly, the optimization process, which is part of the control strategy, determines the energy flow at runtime with respect to the above mentioned optimization problem amongst others (compare chapter 2.2).

The achieved results are used to show the impact of PVT collectors on the seasonal performance factor, the primary energy ratio and the Cost Ratio.

2.4 Optimization

The optimization problem described and used in the project is used to find the optimal strategy to operate the introduced systems (solar thermal and PV driven systems). Improvements of the system itself (e.g. size of the hot storage, etc.) are not considered here.

Solar hybrid systems consist of several components to fulfill specific targets. A typical configuration is a solar thermal collector and a hot backup for domestic hot water preparation, an absorption chiller and a vapor compression chiller to meet the cooling demand. To run such a system, several control loops have to be designed. The satisfaction of the energy demand is given high priority and thus complicating the above described system with three options to provide hot water, and two options for chilled water. Numerical optimization routines can help to find the operating strategy of the system, which leads to a specific minimum of a quantity. The design could e.g. aim at minimizing the primary energy demands.

The strategy of the control system is as follows: If a (course, control oriented) mathematical model of each component is given, the overall system can be described as $\dot{x} = f(x, u)$, $y = g(x, u)$ where x is the system state, u are the inputs (control signals, e.g. pump speed) and y are the control outputs. At any given time instance k, the model is used to make a *prediction* into the near future (e.g. oncoming 24 hours), which depends on u. An optimization algorithm can be used to find the optimal control signal u^* for the following 24 hours. Therefore, the primary energy usage is minimized. After implementing the first element of this optimal time series, the horizon is shifted, $k \rightarrow k + 1$ and the optimization problem is formulated and solved again.

The optimization problem can be solved efficiently by using standard tools. Depending on the structure of the models, nonlinear optimization techniques are a particular solution. However, their solution takes usually a long time and the result might be only a local minima. If the system can be approximated with local linear models with reasonable accuracy, the optimization problem boils down to a linear one for which the global optimum can be found within a fraction of the time-constants of such systems.

The predictive nature of the control system requires a prediction of the energy production of the PV and solar collector, which in turn requires a prediction of the irradiation and temperature. Such values are easy to access and quite accurate over a 24 hour time window.

2.5. Chiller prototypes

According to the original idea of designing a hybrid (absorption/vapor compression) ammonia/water (NH_3/H_2O) chiller comprehensive studies were undertaken. The main difficulty in realization represents the separation and purity of the different substances (ammonia, water and oil) needed for the diverse circuits. Therefore the decision was made to combine the absorption and vapor compression chiller externally via hydraulic connections of the water circuits. Both chillers are based on ammonia and have the same capacity.

- Absorption chiller

In another research project at University of Innsbruck (DAKtris 2012) a new generation of absorption chiller was designed and adapted to the use of dry cooling tower and the resulting high re-cooling temperatures. The development is based on the existing single-effect chiller (PC19), which got improved by analyzation further investigations. This new absorption chiller concept includes a switchable half- and single-effect (HE, SE) combination, flat plate heat exchangers for the evaporator, absorber and generator, enforced throughput of the generator, a new injection nozzle and an adapted internal control strategy.

Correspondingly a prototype of this chiller was built and investigated by measurements at different modes and boundary conditions. This includes the variation of volume flows and chiller inlet temperatures at single- and half-effect, shown in Tab.5.

Tab.5: Set-point variations for measurements of the absorption chiller

Circuit	**ACM-Mode**	**T_{in} (°C)**	**$\dot{V}_{min}$ (m^3/hr)**	**$\dot{V}_{mid}$ (m^3/hr)**	**$\dot{V}_{max}$ (m^3/hr)**
LT / evaporator	SE	12,18	1.5	2	3
	HE				
MT / condenser & absorber	SE	20,25,30,35	4.25	5	6.5
	HE	20,25,30,35,40,45			
HT / generator	SE	80,85,90	3	3,5	4.5
	HE				

The prototype runs stable and delivers satisfactory capacities and EER's. Recording of the chiller performance map was performed by steady state measurements executed with equivalent settling time and a moving average exercised on data.

One characteristic diagram of the absorption chiller at HT temperature of 85°C, LT temperature of 18°C on the left and 12°C on the right side is shown in Fig. 7. Reached capacities are printed in the upper charts with related energy efficiency ratios, beneath corresponding to the MT inlet temperature. Especially the course of capacity shows the potential of the HE chiller mode at higher MT inlet temperatures. Regarding the costs SE has to be preferred due to the higher thermal efficiency. Depending on the overall system other operation modes can have advantages.

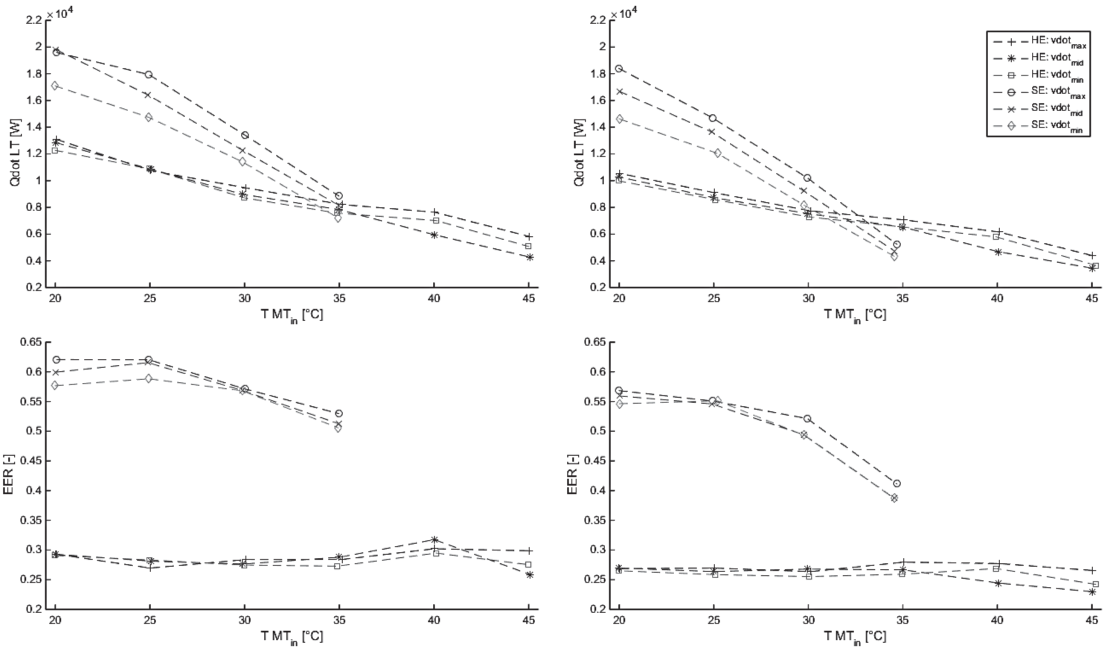

Fig. 7: Measured performance of the new SE/HE ammonia/water chiller with flat plate heat exchangers. LT temperature of 18°C (left) and 12°C (right)

Additionally to the steady state measurements, dynamic hardware-in-the-loop tests take place to evaluate the influence of fluctuating operation and different system setups especially in combination with the vapor compression chiller prototype.

- Vapor compression chiller

The vapor compression chiller L7KW 2T PP, produced by Engie Kältetechnik GmbH, is a water cooled liquid chiller and uses the natural refrigerant R717 (ammonia). The system is built up on the principle of a flooded evaporator in thermosiphon. Nominal cooling capacity is 20 kW (ϑeva = 4°C, ϑcond = 50 °C). The compressor can be continuously controlled by an inverter from 100 to 50%. The range of capacity control is enlarged by a hot-gas by-pass. The chilled water temperature at the evaporator outlet side can vary between 5 °C and 15 °C. On the condenser side the inlet temperature can diversify from 20 °C to 42 °C. These limits are mainly based on the compressor limits.

The theoretical EER varies between 14.59 at the highest evaporator outlet temperature and the minimum condensing temperature and 4.02 at the lowest evaporating and the highest condensing temperature. The chiller will be measured extensively in steady state and dynamic test in the Hardware-in-the-Loop environment of University of Innsbruck in individual operation but also in combination with the absorption chiller.

3. Conclusions

The project SolarHybrid aims to show how solar thermal and solar electrical driven heating and cooling systems can get more efficient and economical viable. This idea implies several optimization steps and comprehensive studies. It becomes obvious that a hybrid solution will benefit from single optimization of the different systems and components.

Detailed models for the adapted absorption and vapor compression chillers are used in the complex HVAC simulations. Huge effort was taken to realize models with data lookup tables, which enable realistic performance even under dynamic conditions, especially in part load. The system simulation model built up in TRNSYS & Matlab, is able to handle the multitude of the different and switchable energy cascades. Several sensitivity studies and the control strategies of the simulation variants were optimized.

The solar thermal systems show promising results for DHW but also for solar thermal cooling. The ST driven system can achieve up to 60% non-renewable primary energy savings with a cost ratio of 1.1 compared to a simple reference system with natural gas boiler and vapor compression chiller. If an electrical driven system with a reversible heat pump is applied as benchmark, savings remain at 50% but the cost ratio drops to 0.9. This magnitude of savings and costs can only be reached with domestic hot water pre-heating in the re-cooling circuit of the chillers (energy cascade).

The integration of PVT collectors benefit from the low temperature levels of the used load profile (hotel with pool). First results show an increased electrical output of roughly 10% compared to ordinary PV yield. Different hydraulic schemes with various temperature levels are under investigation. The effect on non-renewable primary energy savings and the cost ratio will be detailed in future developments.

The optimization procedure is arranged and first tests with solar thermal systems are successful. Additional solar thermal cooling will be set up and tested with integrated models for the chillers (ACM, VCC), hot and cold storage tanks as well as collectors (PV and ST). Finally, the hybrid solution will be optimized accordingly.

Two prototypes, an absorption chiller and a vapor compression chiller, were adopted to the boundary conditions due to the solar hybrid application. The absorption chiller is able to handle re-cooling temperatures up to 45°C under various generator temperatures. The new concept shows promising energy efficiency ratios under the investigated conditions. The vapor compression chiller is designed for the combined operation and especially adapted for large part loads. The laboratory measurements will show this operation mode and the advantages of the new developed chiller concept.

In the following course of the project the steps towards the hybrid solution and the final results will be discussed in detail. A high non-renewable primary energy saving and cost effective solution is expected after the successful first steps.

4. Acknowledgement

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