# SolarHybrid Heating And Cooling – An Environmental Friendly And Economic HVAC Solution

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**Abstract**

Due to increasing living comfort and climate change, an increase in building air-conditioning demand can be expected. Solar air-conditioning is intuitively a good solution, because of the coincidence of solar irradiation and cooling demand. The Austrian research project SolarHybrid deals with the systematic processing and optimization of solar thermal and photovoltaic driven cooling technologies, especially in order to design promising (solar-) hybrid solutions. The project includes the construction and measurements of functional models of a single/half effect $NH_3/H_2O$ absorption chiller and a $NH_3$ vapour compression chiller. The system design and optimization is carried out by means of annual dynamic system simulations for three load profiles.

The combination of the absorption chiller and the vapour compression chiller represent an interesting initial position for effective and economic attractive solar driven HVAC solutions. Under the considered boundary conditions the solar thermal configurations appear to be more efficient and more cost competitive than the PV or PVT variations. Solar heating and cooling can become cost competitive when systems are designed clever with simplified HVAC layouts, advanced control strategies and high efficient components.

*Keywords: ammonia vapour compression chiller, absorption chiller, hybrid system, solar cooling*

## 1. Introduction

The supply of heat and cold in the building and industrial processes has a large potential for non-renewable primary energy savings. However, the substitution of non-renewable energy sources should always follow the basic premise of firstly reducing energy demands before applying renewable systems. High solar fraction and thus high non-renewable primary energy saving can be achieved with solar thermal (ST) or solar electric (PV) supported systems. However, there is a controversial discussion about these two technologies and their technical and economic aspects ongoing. Both solar technologies can offer energetic and/or economic advantages depending on the boundary conditions and the ratio of heating to hot water demand.

Within the project SolarHybrid different hybrid concepts for solar heating and cooling (SHC) are investigated. The major challenge for introduction of SHC systems to the market are the initial investment costs. Therefore, one focus of this project is the radical reduction of system components (e.g. no storage, etc.). The approach to achieve this goal is (i) to define possible system configurations, (ii) to examine them in software simulations, (iii) to test the real components in Hardware-in-the-Loop (HiL) tests, (iv) to validate the simulations with the measured data and run annually simulation studies and (v) to evaluate the results in terms of technical and economic performance.

Both technologies (ST, PV) but also the combined one (PVT) are investigated, their applications in HVAC system are optimized and new solar hybrid systems are developed. The aim is to improve the efficiency of individual components and the overall HVAC systems. The systems are optimized in order to increase primary energy efficiency and economic feasibility in respect to entire load profiles. The optimized components are integrated into system solutions resulting in innovative, radically minimized solar hybrid solutions.


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The economic and energetic optimization of the solar thermal concepts is achieved by means of adapted hydraulic layouts, radically minimized number of components and improved, more efficient control strategies. The technical as well as the economic potential is assessed by means of the T53E4 Tool (Neyer 2016). Extensive sensitivity analyses are used to determine the impact of changing technical and economic boundary conditions on key performance indicators (KPIs).

In the course of the project two components were built and optimized for the hybrid operation; (i) a vapour compression chiller (VCC) and (ii) an absorption chiller (ACM). The ammonia/water ACM, developed in the research project DAKTris (2013), has been further optimized in SolarHybrid, and was driven in hybrid operation with the ammonia VCC in Hardware-in-the-Loop laboratory measurements.

The project is based on detailed research of the components and systems, their modelling and the different load profiles. The systematic analysis of the simulation approaches and the individual models with their part load behaviour and application limits are an essential base for modelling. Through the appropriate choice of the models, dynamic system simulation models are created. Further investigations are carried out with suitable dynamic simulation models, which were adapted according to the laboratory measurements. Three profiles are investigated; an office building, a hotel and a potential study. The hybrid systems are optimized regarding non-renewable primary energy savings and economics. Figure 1 is showing the overall methodological approach of the project.

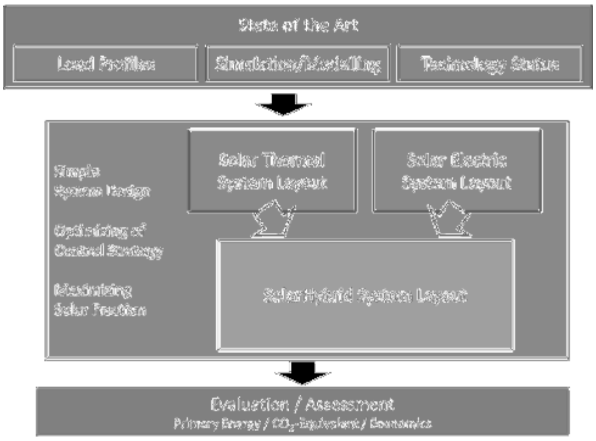


**Fig. 1: Methodological approach and workflow of SolarHybrid**

This paper is focusing on the simulation results of the hotel profile and the potential study of the hybrid ACM and VCC operation.

## 2. Methodology

Three main methods are described here. The assessment and its key performance indicators, the steady state and dynamic laboratory measurements and the simulation study for the hotel and the potential study. Table 1 summarizes the main nomenclature and subscripts.

**Table 1: Nomenclature and Subscripts**

| | | | | | |
|---|---|---|---|---|---|
| ACM SE | Absorption chiller, Single effect | $f_{sav}$ | Savings | PV | Photovoltaic |
| C | Cooling | in | Input | Q | Energy [kWh] |
| CR | Cost ratio (-) | $\eta$ | Efficiency | ref | Reference system |
| DHW | Domestic hot water | HB | Hot Backup | SH | Space heating |
| $\varepsilon$ | Primary Energy Factor ($kWh/kWh_{PE}$) | loss | Losses | SPF | Seasonal Performance Factor (-) |
| EC | Energy Carrier | NRE | Non-renewable | sys | System |
| el | Electrical | out | Output | VCC | Vapour compression chiller |
| equ | Equivalent | PER | Primary Energy Ratio | WD | Water Distribution |

## 2.1 Assessment – T53E[4]-Tool

The performance assessment of SHC systems can be complex. Performance indicators can be determined in numerous ways by taking different system boundaries into account or using different energy quantities. Performance evaluation of integrated SHC systems distinguish in the distribution of collected solar heat and backup energy sources to space heating (SH), domestic hot water (DHW) and cooling (C) applications.

The assessment is performed with the techno-economic evaluation tool of IEA SHC Task 53. The T53E[4]-Tool offers a database of primary energy factors (ε), efficiencies and economic data for all main SHC components. The evaluation is based on non-renewable primary energy for electricity and all energy carriers. Electricity exported towards the grid does not reduce the primary energy effort of the system, but the economic benefits is considered. The most important key figures are briefly discussed, the according equations are shown in Table 2. The key figures are fully discussed and defined in the IEA SHC Task 53 (Neyer et al. 2016).

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

The evaluation is carried out based on non-renewable primary energy (NRE). The $PER_{NRE}$ quantifies the non-renewable primary energy efficiency of the system (eq. 1). The primary energy factors (ε) required for the conversion are depending on type of energy, the location of the plant and the season.

- Non-renewable primary energy savings ($f_{sav.NRE}$)

The primary energy of the entire SHC system is compared with a reference system. The standardized Task 53 reference system is an air-cooled compression chiller (VCC) and a natural gas boiler and its primary energy ratio $PER_{NRE.ref}$ is calculated according equation 2. Applying both PERs (SHC and ref) the savings can be calculated according to equation 3.

- Equivalent Seasonal Performance Factor ($SPF_{equ}$)

The $SPF_{equ}$ is the ratio of useful energy (supplied to satisfy the needs of the building) to consumed energy from external sources (eq. 4). The energy inputs are converted into electrical equivalents units according to the primary energy factor of electricity ($\varepsilon_{el}$).

- Cost Ratio (CR)

The cost ratio is determined with the levelized costs of energy for the SHC system ($C_{use.SHC}$) and the levelized costs of energy for the reference system ($C_{use.REF}$). These costs include investment, replacement, maintenance, electricity, energy, water as well as feed-in tariffs. The costs are discounted according to the annuity method and are summed up on a yearly basis. If the CR equals one cost equity is reached for the SHC system compared to the reference system.

**Table 1: Overview technical and economic key performance figures defined in IEA SHC Task 53**

| | | |
|---|---|---|
| Non-renewable Primary Energy Ratio | $PER_{NRE} = \frac{\sum Q_{out}}{\sum\left(\frac{Q_{el,in}}{\varepsilon_{el}} + \frac{Q_{in}}{\varepsilon_{in}}\right)}$ | (eq. 1) |
| | $PER_{NRE.ref} = \frac{\sum Q_{out}}{\sum\left(\frac{Q_{out.heat}+Q_{loss.ref}}{\varepsilon_{in}*\eta_{HB.ref}} + \frac{Q_{out.cold}}{SPF_{C.ref}*\varepsilon_{el}} + \frac{Q_{el,ref}}{\varepsilon_{el}}\right)}$ | (eq. 2) |
| Non-renewable primary energy savings | $f_{sav.NRE} = 1 - \frac{PER_{NRE.ref}}{PER_{NRE.sys}}$ | (eq. 3) |
| Equivalent Seasonal Performance Factor | $SPF_{equ.sys} = \frac{PER_{NRE.sys}}{\varepsilon_{el}}$ | (eq. 4) |
| Levelized Costs of energy | $C_{use} = \frac{C_{an}}{\sum Q_{out}}$ | (eq. 5) |
| Cost Ratio | $CR = \frac{C_{use,SHC}}{C_{use,REF}}$ | (eq. 6) |

## 2.2 Measurements & Hardware-in-the-Loop

Two chiller prototypes were tested in the laboratories; (i) an ammonia/water ($NH_3/H_2O$) absorption chiller (ACM) and (ii) a vapour compression chiller (VCC). The ACM was developed in the research project DAKTris (2013), it has a nominal cooling capacity of 20 kW and enables the possibility to switch from single- to half-effect (SE/HE). This permits the possibility of high heat rejection temperatures (up to 45°C), the use of a dry cooling tower and the possibility of high useful temperatures in heat pump mode. Within this project, the ACM is adapted and optimized concerning the findings in DAKTris and the requirements of SolarHybrid (e.g. automation of the solvent compensation, etc.). In consideration of the hybrid operation of both chillers the cooling capacity of the VCC is also 20 kW (at evaporation temperature 4°C, condensing temperature 50°C) and the chosen refrigerant is ammonia (R717). The main components of the vapour compression chiller are a frequency controlled piston compressor, a flooded evaporator and a hot gas bypass.

In a first step steady state performance of both chillers are measured and characteristic curve diagrams are generated. Further dynamic Hardware-in-the-loop (HiL) tests of the components integrated in the simulated overall systems and the evaluation of the results are carried out.

- Laboratory infrastructure

The Unit for Energy Efficient Buildings at the University of Innsbruck provides the lab infrastructure for various tests of HVAC components. The infrastructure contains a heat-sink and -source facility. The sink is cooled by a vapour compression chiller (50 kW cooling power) and offers, depending on climatic conditions, the possibility of free-cooling. Three immersion heaters supply the heat for the source, each with a capacity of 15 kW. The temperature ranges up to 95°C for the source and down to -10°C for the sink. Four test benches with a capacity of 10 kW and 30 kW respectively are affiliated.

A multifunctional software is developed with the software platform LabView (National Instruments – NI) to provide the control of the laboratory. It is working on compactRIO (NI) real time devices and enables flexible and easy configurable tests. The communication is based on a dynamic link library, which offers the possibility to connect arbitrary software (TRNSYS, Matlab, Excel, BCVTB, etc.) to the lab.

- Steady state measurements

Steady state measurements are used to identify the performance at several conditions. The settings are adjusted accordingly considering the settling time and the measurement period to calculate a representative mean value and the corresponding deviation.

The measurements of the ACM and the VCC are executed at a wide range of operating conditions. The absorption chiller is tested in SE and HE mode, various volume flows (LT: 1.5–3, MT: 4.25–6, HT: 3–4.5 m³/h) and inlet temperatures (LT: 6–22, HT: 80–90, MT: SE 20–35, HE 20–45°C). The compression chiller capacity is varied through the compressor speed and with the hot gas bypass as well as variable volume flow rates (LT: 2–3.5, MT: 3.5–6 m³/h) and inlet temperatures (LT: 12–22, MT: 25–45°C).

- Dynamic measurements

Overall system performance and components interaction can be evaluated by dynamic measurements. Building up complex systems is time and cost intensive, therefore Hardware-in-the-Loop (HiL) tests are a promising alternative. HiL tests combine system simulations and real component testing. The overall system is designed in an adequate accuracy with a software (here TRNSYS) and the investigated components are connected to the lab. This coupling of simulation and lab in real time enables the determination of the overall system performance.

Based on the steady state results (limiting boundary conditions) and defined system configurations (see next paragraph) a simulation model is set up in TRNSYS. Each configuration is measured under the boundary conditions of two significant days (one sunny, one cloudy) at the chosen location in different variations (different temperatures, mass flows, control strategies, machine modes, etc.).

Following description is one variation of the combined chiller and heat pump mode of ACM and VCC (configuration c). The main purpose of this system is the supply of heat and simultaneously the optional allocation of cooling energy. The defined temperature levels are 12/6°C for LT circuit, 14/40°C for MT circuit and variable HT input (60~95°C). HT (4.5 m³/h) and LT (3.5 m³/h) volume flows are fixed, MT (min. 0.5 m³/h) volume flow is controlled to reach set outlet temperature (40°C). The ACM delivers as much as possible (depends on solar irradiation) and the VCC is complementing to deliver the set LT outlet temperature.

Subsequent to the measurements the evaluated data and energy balances are used to fit the simulation model of the ACM and VCC. Annual simulation studies are carried out accordingly.

- System configurations for Hil tests

As a consequence to the idea of minimized and reduces number of components the measured configurations are defined accordingly as direct solar thermal driven systems. The climatic conditions are simulated for different locations (Innsbruck, Sevilla, etc.). The solar collector field, the cooling tower and the other components are designed accordingly. The grid provides the required electrical energy.

a) Chiller-mode ACM and VCC: The generator (HT) circuit of the ACM is directly connected to a solar collector field (ST), re-cooling (MT) circuits of both machines are connected in parallel with a dry cooling tower and the chilled water (LT) outlet of the ACM is connected in serial to the inlet of the VCC.

b) Heat pump-mode ACM (SE and HE): HT circuit of ACM is directly connected to the collector, MT mass flow is controlled to reach the desired temperature at the MT outlet of the ACM and the chilled water could be used optionally.

c) Combined chiller- and heat pump-mode ACM and VCC: HT circuit of ACM is directly connected to the ST, MT and LT circuits of the ACM are connected in serial to the VCC, MT mass flow is controlled to deliver the defined temperature and VCC capacity is controlled to reach set LT temperature.

## *2.3 Simulation studies*

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

- HVAC simulation models

The individual models (ACM, VCC, cooling tower, etc.) have been investigated and improved on their partial load behaviour. The ACM and VCC are built up as characteristic curve models and are adjusted to the measured steady state and dynamic data.

The ACM model (Type 1005, Neyer et al., (2015b)) is based on semi-physical models of a single-effect (SE) ACM (Hannl et al., 2012) and a half-effect (HE) ACM (Zotter et al., 2015). The model can switch between the two operating modes (SE/HE) and its nominal capacity is scalable. A new characteristic curve model has been developed for the VCC. It’s also based on a semi-physical simulation model (Luger 2017) that was fitted to the according measurements. The deviation between the simulation model and the daily dynamic measurements is smaller than 3% of the total energy balance.

- Load profile

The building simulation of a four-star hotel was set up in TRNSYS. 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 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 ventilation includes heat recovery in winter and the air change rates and specified temperatures correlate with each usage.

In table 3 the annual energy demands and maximum loads for the locations Innsbruck and Sevilla are summed up.

**Table 3: Energy demand and loads for the hotel profile in Innsbruck and Sevilla**

| | Innsbruck | | Sevilla | |
|---|---|---|---|---|
| Application | Demand [MWh] | Load [kW] | Demand [MWh] | Load [kW] |
| Space heating (SH) | 271.2 | 190 | 7.1 | 38 |
| Pool heating (pool) | 274.8 | 208 | 227.8 | 205 |

| | | | | |
|---|---|---|---|---|
| Domestic hot water (DHW) | 562.5 | 260 | 562.5 | 260 |
| Air–conditioning (C) | 85.7 | 80 | 315.5 | 135 |

The load profiles are used individual and in several combinations: DHW only, DHW+C, DHW+C+SH+pool.

- HVAC layouts for the hotel

Six HVAC layouts are compared in detail for the hotel profile and are briefly described here.

*REF:* The standard Task 53 reference system consists of a modulating natural gas boiler and an air-cooled vapour compression chiller.

*ST:* The solar thermal collector field (ST) feeds a hot water storage tank, which is used to ensure the heat supply and operation of the ACM. A natural gas backup boiler is used for DHW and SH only. The ACM is used to cover the base load (19 kW) and the conventional VCC (70 kW) covers the remaining demand using grid electricity. Both chillers operate with a dry cooling tower.

*HP:* A heat pump operates reversible and feeds the hot water storage tank for domestic hot water usage and a cold water storage. Groundwater is used as the source for the heat pump.

*HP+PV:* A PV area, which is dimensioned for the operation of the reversible heat pump, complements the system. The HP is not controlled specifically, thus the coincidence of HP operation and PV electricity results from the simultaneousness from load and irradiation. The demand of electricity from the grid and the excess of PV electricity result correspondingly.

*PVT:* The system consists of a solar thermal collector field for domestic hot water purpose with a natural gas backup. The PVT collector is used to preheat the fresh water for the pool heating and to cool the PV accordingly. The cooling is supplied by a water-cooled conventional VCC (dry cooler).

*PVT(P2H)*: in this variation the surplus PV electricity is used to heat up the hot water storage (power to heat).

*PV(P2H)*: The reference system is equipped with PV. The PV is only used to heat up the hot water storage (power to heat).

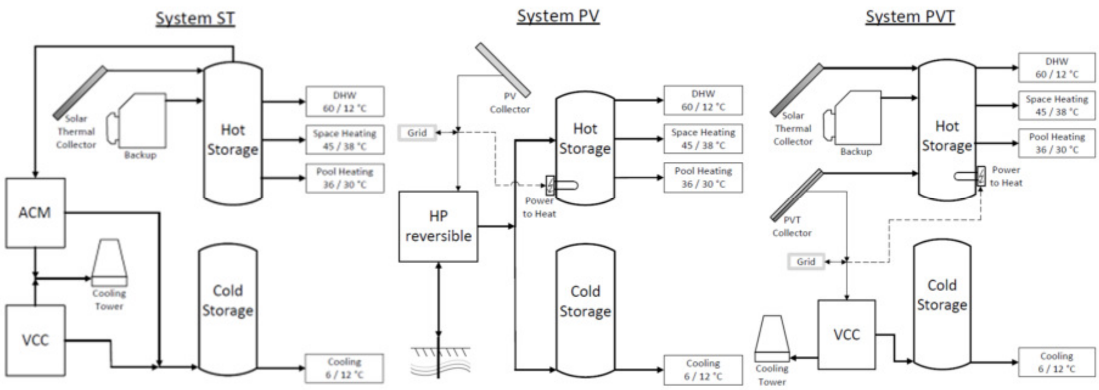


**Fig. 2: Principle schemes of three solar hybrid HVAC solutions**

The collector area for the entire system was varied during the simulation study: solar thermal (ST: 220, 420, 720, 820, 920, 1'120 m²) and photovoltaic (PV: 26, 49.5, 73, 84.5, 94, 130 kWp), PVT (420m² ST + 5, 12.5, 25, 50, 75, 100 kWp) and PV(P2H) (25, 46, 100, 150, 100, 200, 300 kWp).

- Potential study

The same configurations that are measured in the dynamic Hardware-in-the-Loop test are investigated in annual simulations. The system configurations are described above (cf. configurations a, b, c for HiL tests). The absorption chiller (in different modes) is directly solar driven, no hot water storage is included. Due to the design of the system the collector can deliver sufficient generator power to run the ACM if the solar irradiation is greater than 200 W/m². The maximum potential is investigated by means of constant return temperatures of LT- and MT-circuit and the operation through the entire year as soon as there is sufficient radiation available.

Selected boundaries for configuration (c) are presented in table 4. Three operation modes are analysed in order to reach different solar fractions: the ACM in single operation (without LT outlet temperature set point), the ACM with controlled VCC support (to reach the set LT temperature) and the ACM with maximum capacity VCC. The MT inlet temperature (returned to the chiller) is kept constant at 14°C and controlled to reach 40°C outlet (useful) temperature. The LT inlet is kept constant at 12°C or at ambient temperature plus 3 K to simulate an air source heat pump. The measurements are executed for the location of Innsbruck and Sevilla and are distinguished between the use of MT only and the simultaneous MT and LT usage.

**Table 4: matrix of boundary conditions of the annual simulation for configuration (c)**

| Use | Location | LT return | MT temp | LT set temp. | Operation mode |
|---|---|---|---|---|---|
| MT<br>MT+LT | Innsbruck | Amb. +3 K | 14/40°C | Free floating | ACM only |
| | Innsbruck<br>Sevilla | 12°C | | 6°C | ACM + VCC controlled |
| | | | | Free floating | ACM + max VCC |

## 3. Results

This paper only shows a short selected survey of measured and simulated results. More details can be found in the final report of the project SolarHybrid (Neyer 2017).

### 3.1 Measurements & Hardware-in-the-Loop

- Characteristic curves vapour compression & absorption chiller

The steady state measurements of ACM and VCC are executed according to the defined measurement matrix. Figure 3 shows the performance maps of ACM (left) and VCC (right).

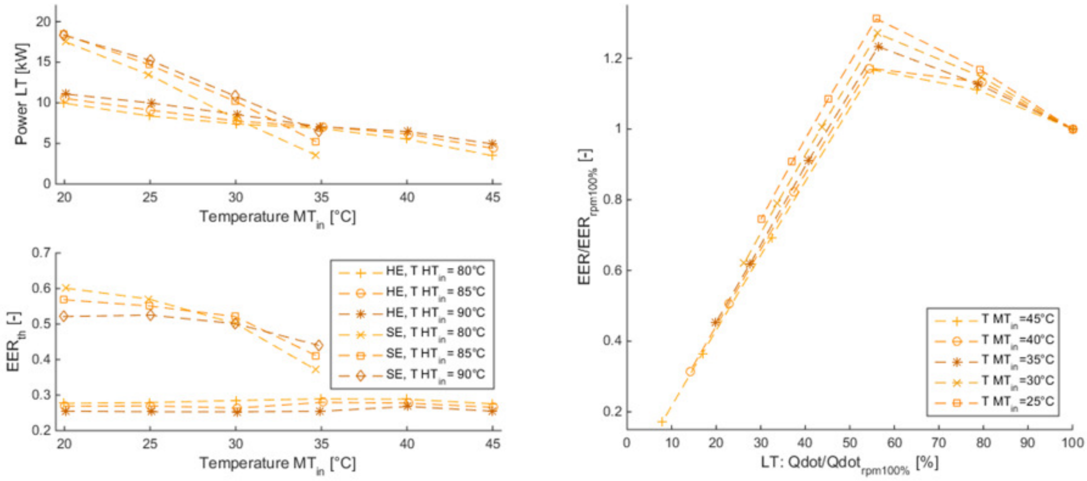


**Fig. 3: Performance maps of ACM (left) and VCC (right), $LT_{out}$=12°C, $\dot{v}_{MT}$=6.5m³/h, $\dot{v}_{LT.ACM}$=3m³/h and $\dot{v}_{LT.VCC}$=3.5m³/h**

The LT inlet temperature is kept at 12°C for both cases (left ACM, right VCC). The MT volume flow is 6.5 m³/h, LT volume flow is 3 m³/h for the ACM and 3.5 m³/h for the VCC measurements. The upper diagram of the ACM map shows the LT capacity and the lower diagram the Energy Efficiency Ratio ($EER_{th}$) in dependence on the $MT_{in}$ inlet temperature. The different lines represent different modes (SE/HE) and different HT inlet temperatures. In SE mode recooling temperatures up to 35°C are possible; HE modes enables recooling temperatures up to 45°C, but at nearly half EER (caused by double generator power). In terms of energy production, switching from SE to HE could be reasonable at a MT inlet temperature higher 32°C (cf. upper diagram).

The VCC map shows values in relation to the nominal capacity, x-axis indicates the relative LT capacity and y-axis the relative EER. The lines represent different MT inlet temperatures, the capacity is first reduced by controlling rpm (100, 75, 50% – right three points of each line) and second on the basis of opening hot gas bypass (left three points of each line). The EER of the VCC is maximized at minimal rpm with closed hot gas bypass, caused by better heat transfer in condenser and evaporator.

Both machines show a good performance, especially under consideration of the prototype status. The adaptations of the ACM show the expected impact and improve the performance significant. The VCC show EERs far above average small-scaled chillers and enables a capacity control for a large range.

- Dynamic measurements

Following the results of the described configuration (c) – combined chiller and heat pump mode of ACM and VCC – are discussed. The measurements are analysed visually, energies are summed up and characteristic numbers (e.g. electrical/thermal seasonal performance factor – SPF) are calculated.

Figure 4 shows the course of the cloudy day (Innsbruck); the upper diagram shows the capacities (HT, MT, LT, electrical) of ACM and VCC and the irradiation (multiplied with factor 10) on the collector field; the lower diagram shows the corresponding temperature (MT, LT in- and outlet; HT inlet temperatures multiplied with factor 0.1; ambient temperature)

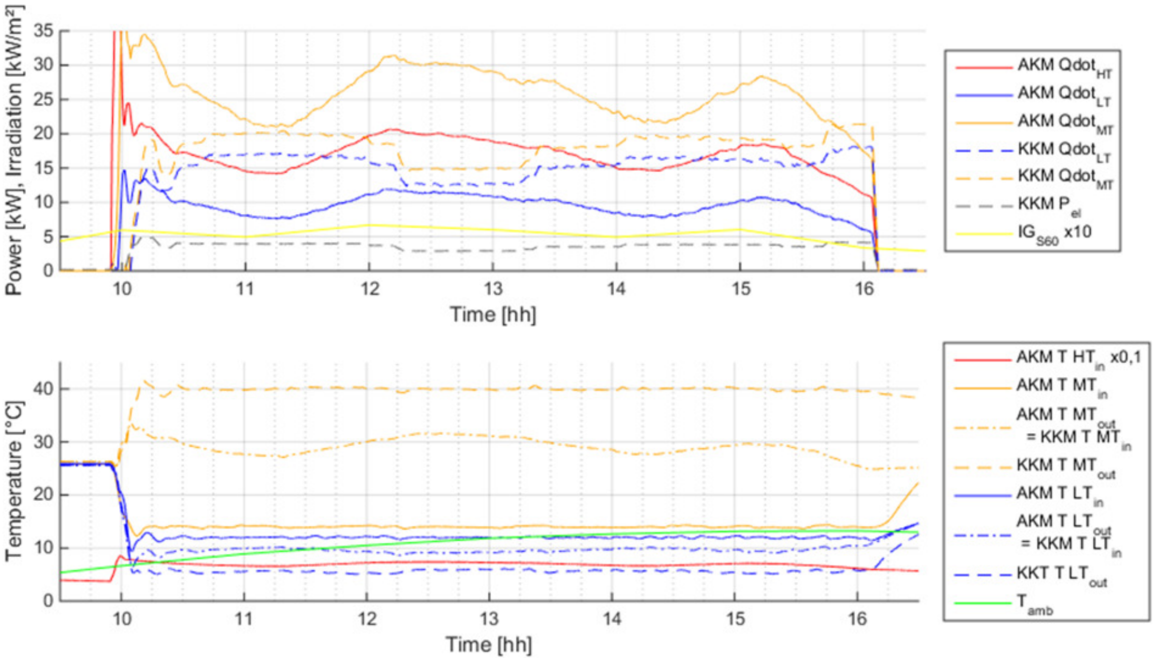


**Fig. 4: Combined chiller and heat pump mode of ACM and VCC – trend over the cloudy day (upper diagram: powers and irradiation; lower diagram: temperatures)**

The power devolution shows the correlation between ACM and VCC, as soon as the ACM delivers less power the VCC increases the capacity to reach the set point. This results in a constant LT outlet temperature (cf. Figure 4 lower diagram), constant MT outlet temperature is reached by means of controlled MT mass flow.

Table 5 shows the energy balances and system SPFs over the daily courses (sunny and cloudy day). The SPFs are distinguished in used energy (MT or MT+LT); the thermal SPF ($SPF_{th}$) represents the ratio of delivered energy (MT, MT+LT) to provided thermal energy (HT); $SPF_{el}$ (electrical) shows the ratio of delivered energy to electrical energy (all circuit pumps, ACM solvent pump, VCC compressor).

**Table 5: Energy Balances and SPFs of combined chiller and heat pump mode of ACM and VCC**

| | **Energy ACM** [kWh] | | | **Energy VCC** [kWh] | | | **ACM+VCC** [kWh] | | **System SPF [-]** | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | | | **MT+LT** | | **MT** | |
| | $Q_{HT}$ | $Q_{LT}$ | $Q_{MT}$ | $Q_{MT}$ | $Q_{LT}$ | $Q_{el}$ | $Q_{MT}$ | $Q_{LT}$ | $SPF_{th}$ | $SPF_{el}$ | $SPF_{th}$ | $SPF_{el}$ |
| Sunny day | 233 | 125 | 349 | 96 | 80 | 21 | 445 | 205 | 2,03 | 20,19 | 1,50 | 13,82 |
| Cloudy day | 102 | 57 | 152 | 102 | 86 | 21 | 254 | 143 | 2,04 | 12,33 | 1,49 | 7,89 |

The configuration with the ACM and VCC used as heat pump reaches in all cases electrical efficiencies of >8. The difference between sunny and cloudy day get obvious but still provide optimization potential as the pumps are all calculated with constant power, including further speed-controlled auxiliaries enables systems with electrical performance of > 10.

### 3.2 Simulation studies results

- Hotel profile

Table 6 shows the most important results of all six simulated HVAC systems for the entire building load (DHW+C+SH) and the location of Innsbruck.

The solar yield is highest for ST (423 MWh), the PVT variation consist of ST+PVT$_{th}$ (220+46 MWh) and PVT$_{el}$ (61 MWh). The PV variation yield is 112 MWh. The demand for backup (natural gas) and/or grid electricity is reduced according to the yield. The surplus PV electricity is fed into the grid for the PV/PVT variations. A corresponding non-renewable primary energy input (PE$_{NRE}$) is resulting and expressed as savings ($f_{sav.NRE}$) in relation to the reference system.

**Table 6: Comparison energy balances, costs and savings for maximum building load (DHW+C+SH/562+82+542 MWh/a) for each system**

| | Unit | REF | ST | HP | HP+PV | PVT | PV(P2H) |
|---|---|---|---|---|---|---|---|
| Collector area | m² | - | 720 | - | 720 | 420 (ST)+ 326 (PVT) | 652 |
| Solar yield | MWh/a | - | 423 | - | 112 | ST:220, PVT$_{th}$:46, $_{el}$:61 | 113 |
| Gas | MWh/a | 1210 | 795 | - | - | 880 | 1071 |
| Electricity supply | MWh/a | 48 | 32 | 518 | 408 | 33 | 30 |
| Electricity feed in | MWh/a | - | - | - | 2 | 39 | - |
| Sum Primary Energy | MWh/a | 1498 | 963 | 1295 | 1020 | 1060 | 1265 |
| Spec. Investment | €/kW | 330 | 1200 | 500 | 890 | 1397 | 890 |
| Investment ratio SHC/REF | - | - | 3.6 | 1.5 | 2.7 | 4.2 | 2.7 |
| $f_{sav.NRE}$ | - | - | 0.35 | 0.12 | 0.30 | 0.295 | 0.16 |
| Cost Ratio (CR) | - | - | 0.36 | 1.04 | 1.05 | 1.05 | 1.04 |

To analyse the costs, the specific investment costs are listed in Table 6. These were calculated with the standardized values of the T53E[4]-tool. The ratio of the investment costs of the entire system compared to the reference ranges from 1.5 for the heat pump up to 4 for the PVT variation. In addition to the investment costs, the costs for replacement, consumption-related costs (electricity, gas, water), maintenance costs and feed-in tariffs are reported. Figure 5 shows a comparison of the distribution of these cost categories in relation to the reference costs.

In case of the reference system, the energy costs for natural gas (73%) are dominant, while the proportion of investment is increasing significantly in the case of all renewable variations. In case of the solar thermal systems, the costs for natural gas; for the heat pump systems, the cost of electricity are the largest cost drivers.

All results of the techno-economic analysis are summarized in Figure 6 (left: Innsbruck, right: Sevilla). The cost ratio (CR) and the non-renewable primary energy savings ($f_{sav.NRE}$) are mapped. The markers represent the simulated variations, which are distinguished by collector size, by application (DHW only, DHW+C, DHW+C+SH) and by the system configuration (ST/PV/PVT). In general, it is obvious that the larger the collector size the larger the non-renewable primary energy savings can get but simultaneously rising cost ratios (CR) are observed.

Figure 6 shows the difference between the individual profiles as well as the system variations and the two locations. With the pure DHW systems, the highest savings ($f_{sav.NRE}$) are achieved (lowest energy quantity, highest solar coverage for the same area). The profile DHW+C+SH leads to the flattest curves (maximum energy quantity).

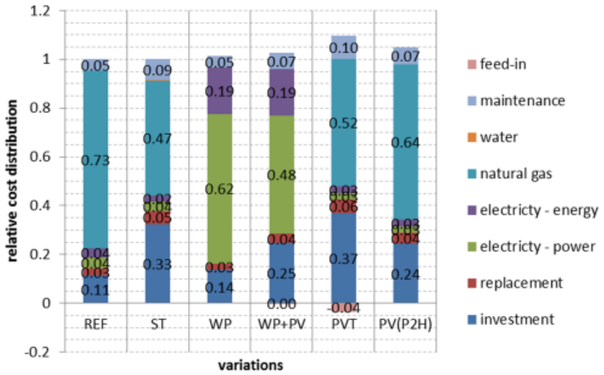


**Figure 5: Relative cost allocation for the total building load (DHW+C+SH/562+82+542 MWh/a) relative to the reference (REF) for each system**

With increasing collector size, the PVT quickly reaches a maximum of savings ($f_{sav.NRE}$) despite further rising of costs. This is due to the fact that, from a certain size, the user's self-consumption is no longer increased as the electricity consumption and the coincidence is low. On the other hand, increased PVT areas lead to larger ratios to preheat the DHW and thus the average collector temperature increases. Since uncovered PVT collectors usually have comparatively low stagnation temperatures, rising temperature can lead to a further reduced thermal yield.

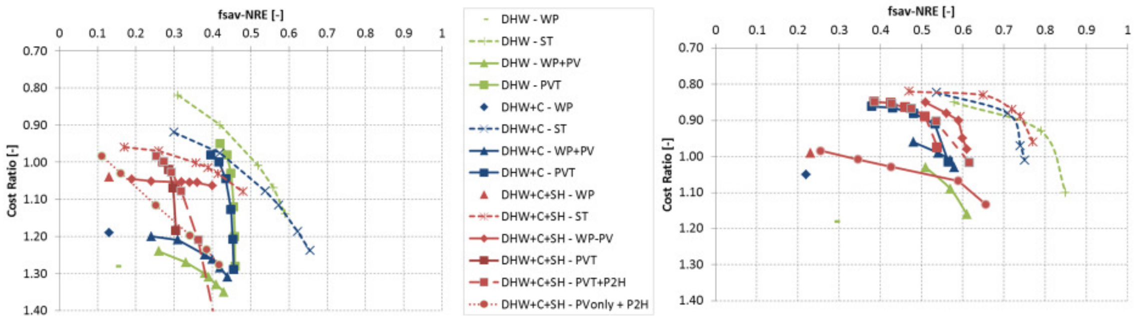


**Figure 6: Technical and economical comparison of plant configurations: ST, HP, HP+PV, PVT, PV (P2H) for Innsbruck (left) and Sevilla (right)**

Compared to the reference, the heat pump achieves $f_{sav.NRE}$ up to 15% (20% for SEV) with additional costs up to 28% (only DHW, IBK). The higher the energy demand the more favourable the HP. A CR of 1.04 (1 for SEV) can be achieved with the complete profile. The ratio of the investment costs to the energy costs, which account for 80% (IBK, cf. Fig.5) of the total profile, is decisive here. The cost can be reduced with high efficient solution accordingly.

If the heat pump is expanded by the PV, small increases in the CR will result for Innsbruck, but pays off in case of SEV immediately. If the PV area is increased, the costs increase accordingly. This result can essentially be attributed to the uncontrolled self-consumption and the trend of the profile.

The ST variations show the largest differences between the profiles as well as the different collector sizes. In the case of small areas, the savings in non-renewable primary energy are lower than in PV and PVT variations, but at lower costs (IBK). In Sevilla all variations of the ST show higher savings than the other solutions. The results (CR, $f_{sav.NRE}$) coincide with analyses of other measured and simulated plants in SHC Task 53 (Neyer 2016).

In sum, depending on the design and boundary conditions, ST, PV or even PVT could be preferred. Under the chosen boundary conditions, the ST systems can achieve the best results regarding costs and non-renewable primary energy savings due to the existing buffer storage and the time resolution and evolution of the profile.

- Potential study

The results of all five case studies of the potential study are shown in Figure 7. Each curve represents the three possibilities in operation mode (c.f. table 4). The ACM only operation mode can thus be found on the right hand side showing the highest savings. They are almost 100%, only a small electricity demand is lowering the $f_{sav.NRE}$ accordingly. The more the VCC is supporting the lower the solar fraction und consequently the lower the savings. At the same time the CostRatio drops rapidly as more energy can be delivered and the domination of investment is reduces. Worst performance is presented by the air-heat pump mode (AMB-14-40, only calculated for IBK). The lower the source temperature (especially in winter) the lower the efficiency and thus lower the savings and the CostRatio. If both LT and MT can be used the performance increases accordingly. Due to the higher solar yield the results for Sevilla are improved compared to Innsbruck.

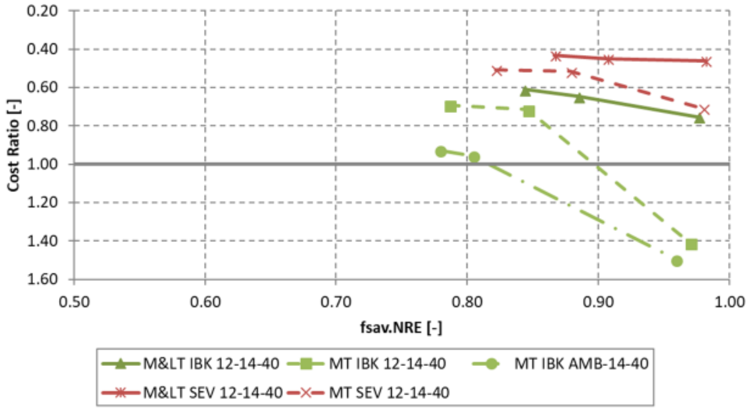


**Figure 7: Comparison of the simulation results for Seville and Innsbruck for the heat pump cases with 12/6°C cold water (evaporator) and 14/40°C hot water (condenser) temperature; nomenclature: use(MT, MT+LT) Location LT return, MT in/out**

Summing up the potential study illustrates that non-renewable primary energy savings >80% and CostRatio's mostly below 0.8 and as far as 0.4 can be reached. As the economic performance is still investment cost dominated a sensitivity analysis on different parameters was performed.

- Sensitivity analysis

The effect of the sensitivity analysis is shown by means of the shift of the trend lines. The parameters that are varied are the total investment costs, the electricity price, the electrical and thermal efficiency of the systems, as well as the reference performance.

Figure 8 is showing the summary of the sensitivity analysis (left; MT IBK 12-14-40, ACM+VCC) and for one curve of Figure 7 (MT IBK 12-14-40). The investment costs show the largest gradient and thus present the highest influence on the cost performance followed by the performance of the reference system and the entire electrical efficiency.

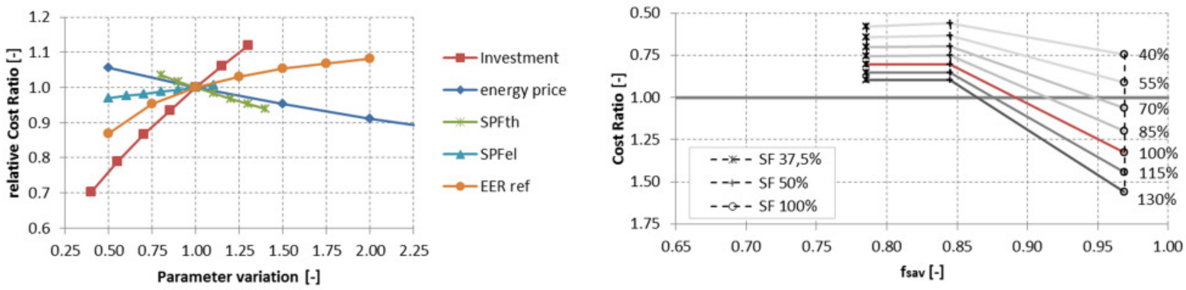


**Figure 8: Sensitivity analysis for the heat pump case (MT IBK 12-14-40) as overview (left) and on investment costs variation from 40% to 130% of the Task 53 standard values (right); SF: solar fraction**

Figure 8 (right) shows that if the invest costs could be reduced by roughly 35% the 100% solar driven ACM would reach a CostRatio below one and thus represent a highly primary energy savings solution that is cost competitive.

## 4. Summary & outlook

In SolarHybrid solar heating and cooling systems were investigated intensively by means of simulations, measurements of prototypes and the assessment & benchmarking against conventional system and other solar solutions.

- The combination of the single/half-effect ammonia/water absorption chiller and the ammonia vapour compression chiller represent an interesting initial position for effective and attractive solar driven HVAC solution
- Under the considered boundary conditions, the solar thermal configurations appear to be more efficient and more cost competitive than the PV or PVT variations.
- Solar heating and cooling can become cost competitive when systems are designed clever with simple HVAC layouts, advanced control strategies and high efficient components.

## 5. Acknowledgment

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