Analysis of a solar hybrid cooling system for industrial applications

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Abstract

This paper presents the analysis of a new Fresnel solar thermal collector (FCSP) system with a Hybrid Heat Pump (HHP) for solar cooling purposes in industry using TRNSYS simulation. The HHP is a combination of a thermally driven adsorption chiller and electric compression chiller in a cascaded configuration, to achieve highly increased Energy Efficiency Ratios (EER) for the operation of the solar assisted HPP system. The benefits of the solar assisted HHP system is mainly based on constantly low heat rejection temperature of the compression chiller, compared to conventional ambient heat rejection systems, as the adsorption chiller is responsible for the heat rejection of the entire hybrid cooling system. The evaluation of a demo case in Barcelona, Spain showed, that EERs between 7-8 can be reached, hence, the electric power consumption of the cooling system can be reduced by 29 % in transitional seasons and by 44 % during summer. As a result of this approach, it is determined that full load operation conditions of minimum 3,800 h and ambient heat rejection conditions with above 20°C have to occur constantly through operation period, to enable an overall economic operation of the solar assisted HHP System. As an example, for the investigated industry process in Barcelona specific cooling costs of 0.031 EUR/kWh can be reached at operating hours of 4,748 h/a. Moreover, it was analysed that the solar assisted HHP system is likely feasible for the implementation into industrial process typologies with both, heat and cold demands.

Keywords: Solar cooling, solar process heat, hybrid heat pump, adsorption chiller, cascade system, heat rejection, TRNSYS simulation

1. Introduction

More than 20 % of direct green house gas (GHG) emissions can be traced back onto the industrial sector. By attributing further emissions from electricity and heat production this share increases to 31 % GHG emissions in total. (IPCC, 2014) Despite the general awareness on a positive trend of the renewable energy sector, the use of renewables in the global industry sector is devastatingly small >1.8 % (Mekhilef et al., 2011). About 60 % of the thermal heat demand in industries is required at temperature levels below 250 °C (Vannoni et al., 2008). At these temperature levels, solar thermal energy shows a great potential to be applied in the field of industrial applications. But due to high capital costs and low costs on fossil fuels, only limited deployment has taken place over the last decades. Furthermore, the application of solar energy into industrial processes places particularly high requirements on performance and reliability. Yet, the potential in CO_2 saving through the implementation of renewable energy sources is huge, for both, heat and cold supply. Therefore, the potential of solar thermal heat for direct use or generation of cold needs to be pushed by highly efficient innovations and technologies. So far, the market for solar cooling is still very young, with only few manufactures for sorption chillers, especially for solar cooling in industrial applications (Daßler and Mittelbach, 2012). Since new regulations for the use of refrigerant are passed (F-gas Regulation, 2014), the market urgently requests low Global Warming Potential (GWP) refrigeration systems for present and future cooling systems.

2. The investigated solar HHP system

The solar hybrid cooling system investigated consists of a Fresnel solar thermal collector (FCSP) together with a Hybrid Heat Pump (HPP). The system shall provide a wide range of design and operational configurations to increase the potential implementation of solar heat (heat/cold) in industrial systems. The linear FCSP (Figure 1) is designed to work with heat transfer fluid such as pressurised water, water-glycol or thermal oil. Typical heat transfer output temperatures range between 130°C - 250°C. The advantage of this linear Fresnel is the use of

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multiple small mirrors. These flat mirror strips are significantly less expensive in manufacturing than curved mirrors for a parabolic through collector (Fresnex, 2019).

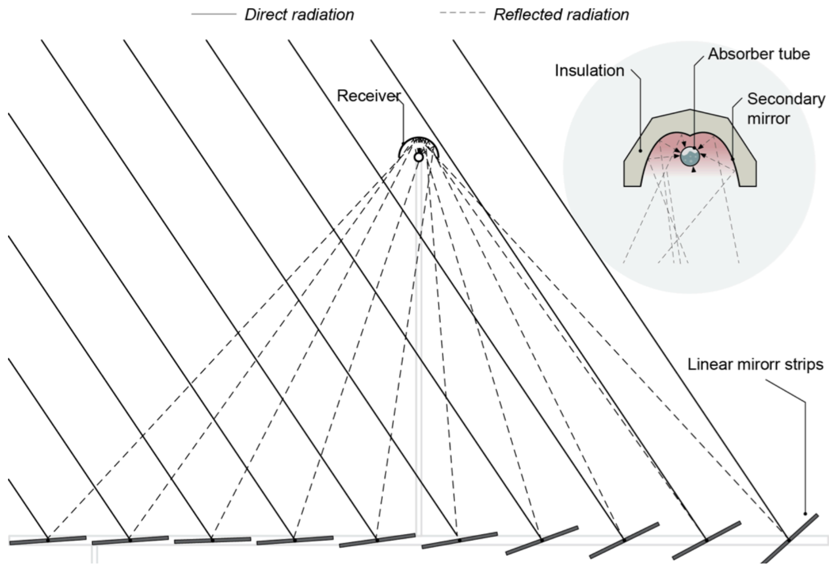

Fig. 1: Schematic scheme of a linear Fresnel solar thermal collector (Source: JER)

The HPP technology covers cooling loads for industrial processes at a minimum temperature T_{chill} of +5°C. In the future, depending on the chosen compressor technology, temperatures as low as T_{chill} of -10°C or even lower can be reached by the HPP. The operation of the HHP requires for cascaded mode (Figure 2) thermal energy for the adsorption chiller (AdC) and electrical power for the vapour compression chiller (VCC). Sorption systems have a highly reduced electrical energy consumption and a limited maintenance expenditure, whereas their efficiency with high temperature lifts between condenser and evaporator is limited and their response on fluctuating demand loads is low. Electric compression chillers, on the other hand offer high precision in temperature regulation and fluctuating load coverage but require a high demand on electrical input power especially at higher condensation temperatures (Vasta et al., 2018).

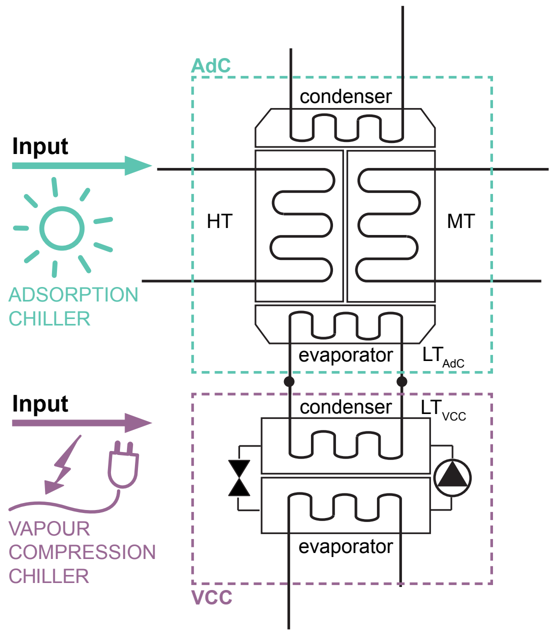

Fig. 2: Schematic layout of the Hybrid Heat Pump (Source: JER)

Therefore, the main idea of the HHP system layout is to significantly increase the efficiency of a VCC by reducing its condenser temperature. Applying the evaporation temperature of the AdC for heat rejection purposes does this in a cascade configuration. Driven by hot water in the range of 80°C - 90°C, the AdC represents the warm stage of the cascade. The adsorption process provides cold water at a low temperature level ($T_{chill,AdC}$ about 20°C). The second stage of the modular HPP is based on a vapour compression unit. For the HPP model both working principles are combined. The cold water from the AdC, with a set point temperature $T_{chill,AdC}$ of 20°C, will be used to dissipate heat from the condenser of the lower compression stage. In this way, the temperature lift between evaporator and condenser of the compression circuit is reduced. This leads to low electric input power for the VCC and therefore to highly increased Energy Efficiency Ratio (EER) values for the system.

3. Modelling solar HHP system

For the investigation of the solar HHP system a TRNSYS simulation deck was developed including solar concentrating collector circuit, process heat and hot water supply circuit, heat rejection circuit as well as heat demand side and cold demand side circuits. The different main components of the deck (Figure 3) are modelled using Type 1288 for the FCSP (concentrating collector) and the HPP as combination of an adsorption chiller and a compression chiller.

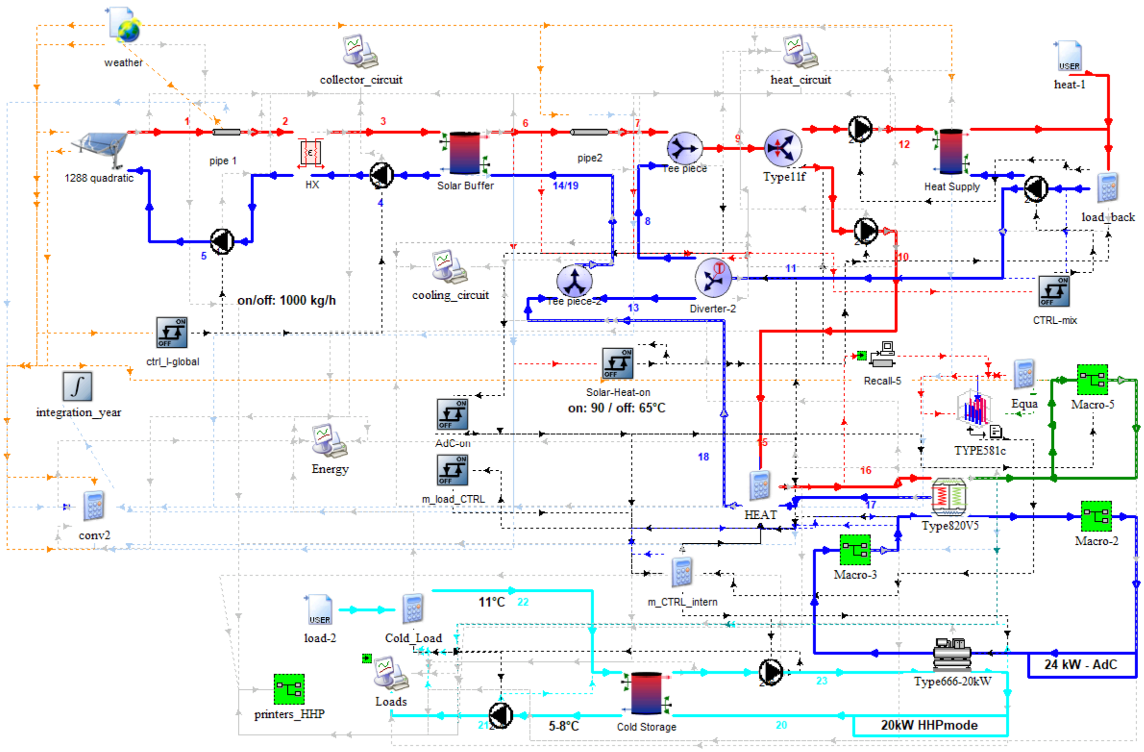

Fig. 3: TRNSYS deck of the solar Hybrid Heat Pump system (Source: JER)

Therefore, a novel HPP model was developed in TRNSYS (Kiedaisch, 2018) consisting of the silica-gel adsorption chiller Type 820, which is based on a lump parameter model (Dalibard, 2017) with dynamic heat and mass transfers equations for the ad-/desorption of the two beds (also other sorbents can be used, like zeolite), a simulation time step of 10s was chosen. As the adsorption type is the most dynamic and detailed component of the system, special attention has been paid on its operational behaviour. The simulation results of the adsorber showed quite good consistency on its sorption behaviour compared to nominal ranges from literature. Furthermore, for vapour compression unit the Type 666 was investigated, which is also based on a lookup table. The standard lookup table of Type 666 showed that the EER is limiting the overall EER increase of the system to 25.7 %, hence, a new lookup table was created. The new lookup approach based on a nominal capacity (EER of 4.4) for a R290 chiller and the deployment of an adapted Carnot-efficiency fit curve. To achieve a desired cooling load $Q_{chill,CC}$ of 20 kW at a chilled water set temperature of 7°C, the thermal adsorber chilling capacity $Q_{chill,AdC}$ is 24 kW at a heat rejection and heating temperature of 30°C and 90°C, respectively (Figure 4). Finally, the interconnection of both units was applied and a parametric simulation was carried out to define boundary conditions of the HHP-System.

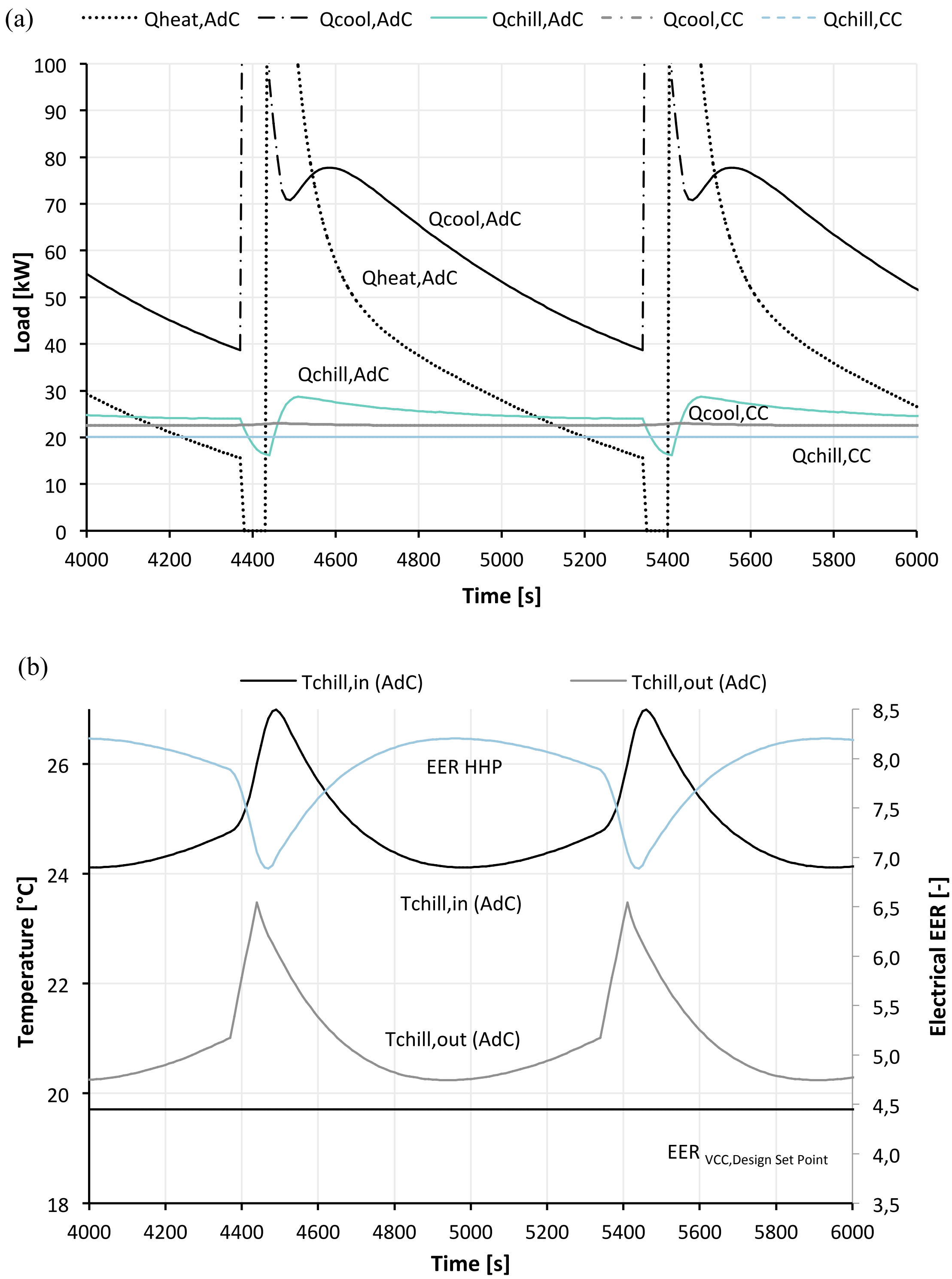

Fig. 4: Simulation results of the HHP modelling: (a) heating, cooling and chilling loads of AdC and VCC; (b) AdC chilled water in-/outlet temperatures and electrical EER of the HPP

The variation of heat input (60 to 95°C) and heat rejection temperatures (25 to 40°C) for the system pointed out that a set point control (adapting mass flows) for the HHP system is necessary to provide a reasonable operation. Furthermore, a capacity ratio for the design of adsorption and compression chiller was detected with a factor of CR = 1.2 for the design of the adsorption chiller. The electrical EER for the HHP operation varies between 6.9 and 8.4 as shown in Figure 4 with the corresponding chilling temperatures of the AdC with ΔT_{chill} of 2.5 to 4 K. $T_{chill,out}$ is slightly above the set point temperature of 20°C. It is evident that chilled water outlet temperature is exceeding its set point of 20°C, when the sorption cycle is switching between the two adsorption chambers and the new sorption cycle begins. The simulation results of the cascaded operation are showing a successful interaction of the thermal and electrical unit.

4. System simulation

The overall HPP system including concentrating FCSP field, the HPP and appropriate demand profiles for industrial application was simulated for two seasonal periods: spring and summer. The aim of the investigated solar assisted HHP system is to achieve an EER close to 10.0 for the operation of the compression chiller, hence, heat rejection set point temperatures at the condenser side of 20°C shall be obtained.

4.1. System data and KPIs

The simulation of the HPP system is done on a five days approach. Therefore, a summer week and a week in spring were analysed in detail because of the dynamic operational behaviour of the AdC. Table 1 shows the design parameters of the simulated solar HPP system.

Table 1: Design parameters of the solar HPP system

Solar concentrating collector circuit	FCSP field size (mirror area)	400 m^2
	Heat storage volume	6.5 m^3
Process heat and hot water supply circuit	Hot water storage volume for cleaning and sanitary	4 m^3
Cold demand side including heat rejection circuit	HPP cooling capacity	20 kW
	Cold water storage volume	6 m^3
	Heat rejection capacity (dry recooler)	61 kW

Furthermore, heat and cold load demand for a food processing plant in Barcelona, Spain (Table 2) were applied to the HPP system to estimate a possible coverage of it. The fluctuation of the analyzed heat load profile for cleaning and sanitary purposes shows peak loads of almost 35 kW, the minimum demand is 8.3 kW. Base load is 16.5 kW even through non-working days (weekends). The heat load occurrence and frequency shows that more than 80% of the heat load occurs in the range of 10-20 kW. It is assumed that for a constant heat demand through the year the annual heat demand rates to 156 MWh/a for the investigated case. The investigated annual cold demand for the food storage shows peak loads of 27.6 kW, the minimum demand is 4 kW. By analysing the load curve, it is seen that with a design load capacity of the HPP of 20kW almost 90% of the load is covered. It is assumed that for a constant cold demand through the year the annual demand rates to 136 MWh/a for the investigated case.

Table 2: Annual heat and cold demand for a food processing plant in Barcelona, Spain

Heat demand for cleaning and sanitary purposes	Cold demand for food storage
156 MWh/a	136 MWh/a

For the evaluation of the solar HPP system the following Key Performance Indicators (KPIs) were used to determine the performance improvement (Table 3). EERs and COPs evaluate the chiller´s performance at full load operation at nominal conditions. The EER is generated by the division of output and input. Therefore, it needs to be distinguished between the EER_{el} (Equation 1) of a compression chiller and EER_{th} (Equation 2) of an adsorption chiller. The improvement in efficiency of the HHP (EER_{HHP}) is only based on an increasing EER_{el}. Neither is the actual technical performance of the adsorption nor of the compression unit improved. The significant increase of the EER_{HHP} is only a result of reduced heat rejection temperatures at condensation side of the compression unit.

Table 3: Chiller performance (HyCool D2.5, 2018)

Energy Efficiency Ratio (EER)	Definition
electrical	$\mathrm{EER}_{el} = \frac{Q_{chill}}{P_{el}}$ (1)
thermal	$\mathrm{EER}_{th} = \frac{Q_{chill}}{Q_{heat}}$ (2)

4.2. EER results

The simulation results for the FCSP field shows solar annual yields of 228 MWh/a for the location of Barcelona, where the solar heat is used for the HPP and the remaining heat for cleaning and sanitary purposes. The heat storage size of 6.5 m^3 allows a maximum time shift of load supply of 4 hours.

The evaluation of the EER is the key point in the analysis of the solar HHP system. The following Figures 5 and 6 present the dependency of the EER in HHP operation mode for both cases (spring and summer) compared to a conventional compression chiller, being re-cooled by ambient air. Figure 5a is displaying the ambient temperature $T_{cool,in,amb}$ as well as the output temperature of the AdC ($T_{chill,out}$). The heat rejection temperature difference is actually the operation potential, which is used to achieve increased EER of the HHP. A temperature delta is only achieved when HHP is operating.

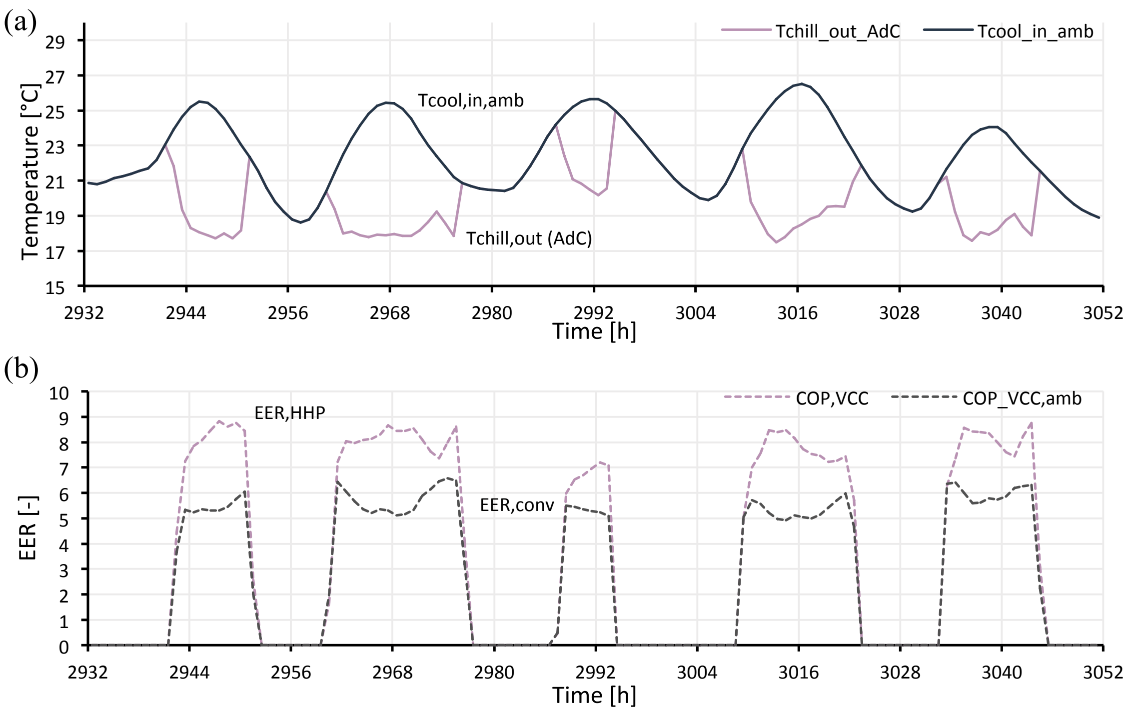

Fig. 5: Simulation results of heat rejection temperatures and efficiency (EER_{HHP} and EER_{conv}) for spring period

The Figure 5b presents the specific EER for both heat rejection options. The increase of the EER_{HHP} is significant compared to the regular EER_{conv}. The nominal (rated) EER of the system is 4.4, hence, the heat rejection conditions in spring are quiet good, low $T_{cool,in,amb}$, also the EER for the conventional compression chiller operation is remarkably high (EER_{conv} = 5 – 6.6). A maximum EER_{HHP} for the HHP operation is achieved with 8.8. The total electric power consumption of the two chiller operation modes is 181 kWh for ambient heat rejection and 129 kWh for HHP operation, this leads to electrical energy savings of 29 %. Figure 6a is showing the same graphs as described above, but for the summer period. The driving temperature difference between $T_{cool,in,amb}$ and $T_{chill,out}$ (AdC) is much more of distinctive, hence, the increase of the EER_{HHP} compared to EER_{conv} is also more significant. For hours of peak ambient temperature, EER_{HHP} is even more than double the EER_{conv}.

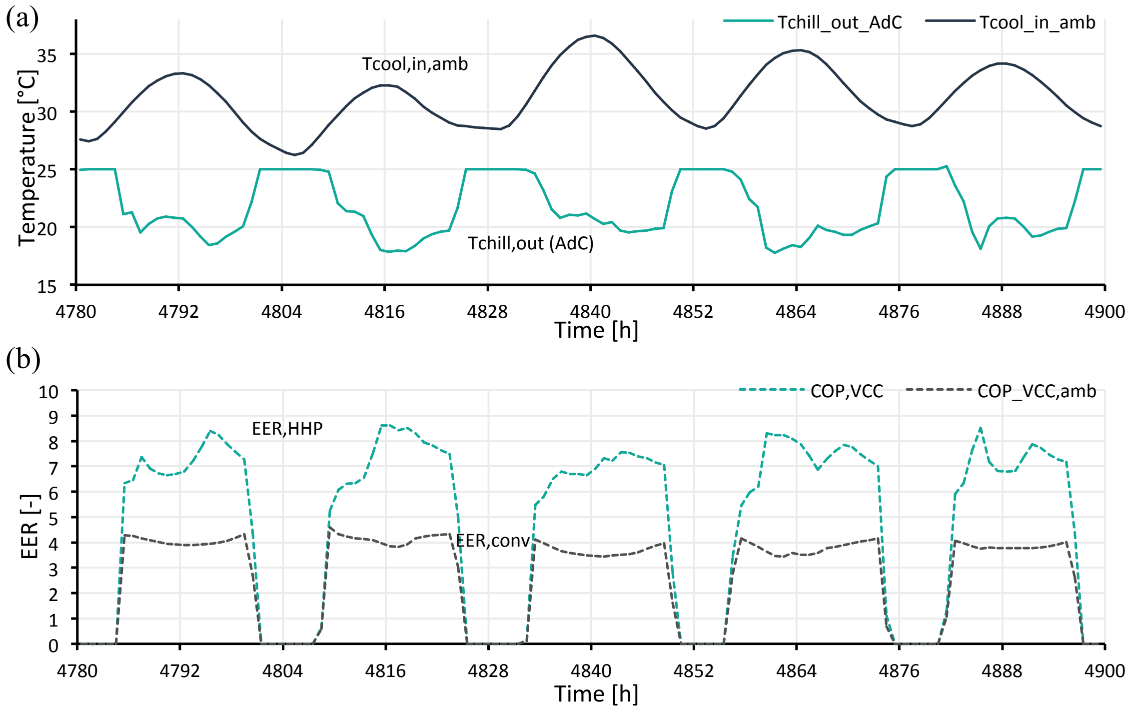

Fig. 6: Simulation results of heat rejection temperatures and efficiency (EER_{HHP} and EER_{conv}) for summer period

The results in Figure 6b show that the effect of the HHP operation is more effective for increased ambient temperatures, as the temperature difference between $T_{cool,in,amb}$ and $T_{chill,out}$ (AdC) is the driving force for the ΔEER. This effect underlines a seasonal operation strategy of the solar HHP system, due to the fact that the HHP does not have advantages in the EER for $T_{cool,in,amb}$ below 20°C, as long as the set point control for the adsorption chiller output temperature is set to 20°C. The total electric power consumption of the two chiller operation modes is 326 kWh for ambient heat rejection and 182 kWh for HHP operation, this leads to electrical energy savings of 44 %. The total energy amount does contain implicate auxiliary demands for the adsorption chiller or heat rejection units.

The evaluation of the electrical EER showed a significant increase of the EERs for the HHP operation compared to a conventional heat rejection of the compression chiller by reaching EERs between 7 to 8 for the demo case (food industry) in Barcelona, Spain. The average efficiency increase for the summer period is 183 %, whereas transitional season operation (spring period) only improved by 138 %. One should note that for this preliminary comparison of the system, the same chiller with a relatively high nominal EER was used and only the method of heat rejection was changed (HHP or ambient heat rejection). Therefore, the increase of efficiency is expected to be significantly higher in real conditions, when compared to common refrigeration systems with rather poor EERs.

4.3 Coverage load results

A main issue in the analysis of the solar HHP system is the question of load coverage for cooling and process heat by the investigated system. The load coverage analysis of the system shows, that the coverage by the HHP system for cooling purposes is higher in the summer period than it is for spring. The relative coverage in summer is about 61.8 % against 47.2 % in the spring period.

As the simulation approach focuses on the operation of the HHP system to generate cold, thermal load surpluses were not directly used on heat demand side. Therefore, surplus solar energy can either directly supply the heat demand load or can be transferred into the heat supply storage. A thermal heat storage of about 18 m^3, at a temperature delta of 40°C, would be needed to utilize (store) the overall surpluses. In fact, the lack of solar thermal supply amounts to maximum 10-14 hours depending on the seasonality. Therefore, a reasonable storage design to cover 10 h of average heat load demand of 18 kW would take around 4 m^3. This actually matches the system layout, regarding the demo case (food industry) in Barcelona, Spain. The solar assisted HHP system is able to substitute both, heat and cold. For the improvement of the overall system storage layout needs to be improved, since an overnight storage design could extend the use of the solar assisted HHP system for summer period from mid-May till end of September.

5. Tool

Based on the simulation results an online tool was developed, the HyCool Pre-Feasibility Simulator (PFS). The PFS tool allows analysing the most relevant boundary conditions, which influence the techno-economic feasibility of the solar HHP system for a given industrial cooling process. The five main types of boundary conditions identified, investigated and analysed at European level, which influence the attractiveness of the HPP system, are listed in the following Table 4.

Table 4: Boundary conditions under which the solar HPP system is techno-economic feasible (HyCool D2.1, 2018)

Boundary conditions	Definition
Solar irradiation	The solar HHP system in the project relies on concentrating solar collectors, therefore, only the Direct Normal Irradiation can be exploited.
Ambient temperature	Any chiller needs to reject heat to the environment. In general, ambient air is the heat sink. The lower the ambient temperature, the higher is the heat rejection efficiency
Industrial process typologies	The required cooling temperature of the industrial process affects the feasibility of the solar HPP system.
Electricity prices	The solar HPP system relies on a compressor chiller, which is operated by electricity
Load profile / duration	The duration of an annual load profile for cooling demands in an industrial process typology influences the economic feasibility of a solar HPP system.

As the heat rejection temperatures are the driving factor for the increase of the EER, and therefore a crucial design parameter, which affects a reasonable implementation of the solar HHP system, the annual distribution of the dry ambient temperature and the wet bulb temperature need to be analysed in terms of an appropriate heat rejection system and the potential heat rejection hours of the HHP. Potential heat rejection hours, are describing the period of time, where the HPP is operating in a beneficial state. A simplified approach is assumed which correlates the average ambient temperature of the year with the number of heat rejection hours above 20°C.

The seasonal operation is highly dependent on the climatic aspects as shown in Figure 7 for Barcelona. There is a seasonal potential operation use for the solar HPP system, yet year-around operation is not reasonable due to low ambient heat rejection temperatures. It also clearly visible that the operation of the HHP at night times is only reasonable for the time period of mid of May till end of September, whereas heat rejection temperatures are constantly above 20°C. The operation during night would significantly increase the hours of full load operation.

The potential operation hours of the HHP (Figure 7), for the site of Barcelona, compared to conventional dry and wet heat rejection systems amount to 4,748 h (dry) and 3,558 h (wet). For warmer climates the potential operation hours will increase.

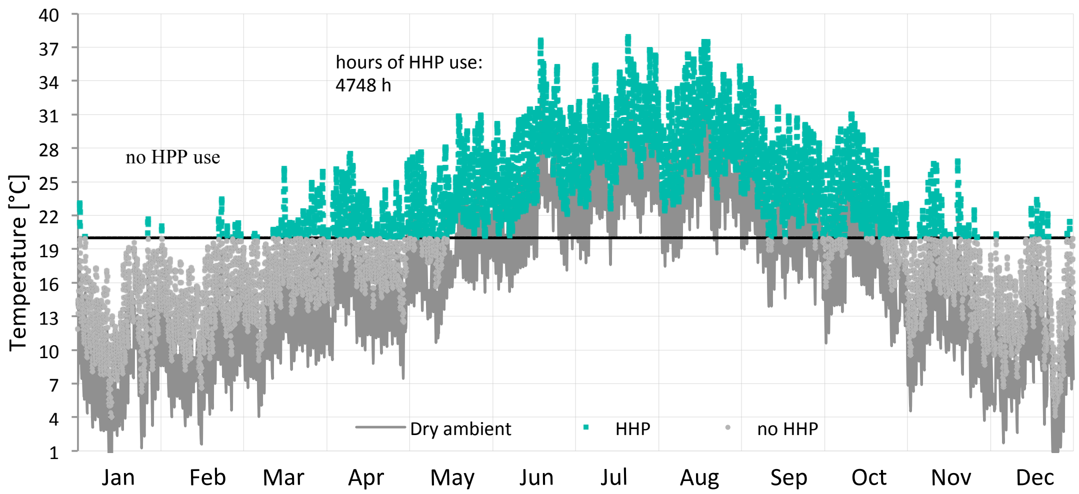

Fig. 7: Ambient temperature level and potential heat rejection temperatures for the site of Barcelona (Data from Meteonorm)

In order to provide the online user with a useful and simple tool for pre-assessing the feasibility of the solar HHP system for a given industrial cooling process, a score-based approach has been chosen (HyCool PFS, 2019). For each of the above-mentioned conditions a ranking has been developed, which quantifies how suitable the technology is under that specific perspective. Knowing the type of cooling process, the locally available solar irradiation, the average electricity price and having access to basic data about external ambient temperature, one can therefore quantify a score for each of those and have a general picture of how suitable the HPP system is for that specific site. As an example, for the investigated industry process in Barcelona specific cooling costs of 0.031 EUR/kWh can be reached at operating hours of 4,748 h/a.

6. Conclusions

The benefits of the solar assisted HHP system are mainly based on constantly low heat rejection temperature, therefore an optimized operation is based on high heat rejection temperature difference, compared to conventional ambient heat rejection systems. The simulation results and conclusion are based on the approach of a constant adsorption set point temperature of 20°C. The evaluation of the electrical EER_{HHP} showed significantly increased EERs for the HHP operation compared to a conventional heat rejection of the compression chiller. The average efficiency increase for the summer period was 183 %, whereas transitional season operation only improved by 138 %. For the use cases the electrical demand of the compression chiller was reduced by 29 % (spring) and 44 % (summer). By reducing the set point temperature of the adsorption chiller in transitional season, it can be expected that the operation through the year becomes more effective, in the number of potential operation hours as well as for higher EERs. As a result of this approach, it is determined that full load operation conditions of minimum 3,800 h and ambient heat rejection conditions with above 20°C have to occur constantly through operation period, to enable an overall economic operation of the solar assisted HHP system. Moreover, it was analysed that the solar assisted HHP system is likely feasible for the implementation into industrial process typologies with both, heat and cold demands.

7. Acknowledgments

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