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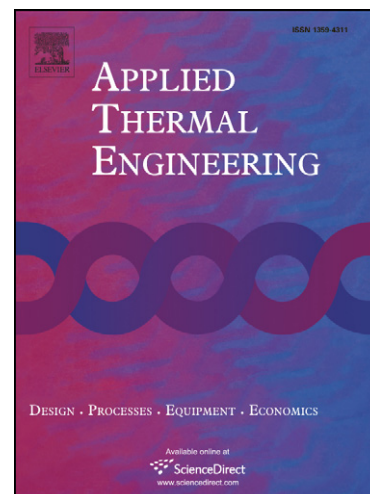
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SOLAR DRIVEN STEAM JET EJECTOR CHILLER

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Abstract

This paper presents a solar driven process to generate cold water for air-conditioning by parabolic trough collectors and a steam jet ejector chiller. The only working fluid in the system is water, which is used as refrigerant and working fluid. The operational behaviour of such a system has been investigated by a small test rig. The investigation shows that the cooling water temperature as well as the cold water temperature has a strong influence on the coefficient of performance of a steam jet ejector chiller. The coefficient of performance reaches high values in part load and at good re-cooling conditions, so that the mean efficiency is clearly higher than the nominal efficiency of the system. A first calculation of profitability leads to specific cold cost of 0.62 €/kWh in Germany and 0.15 €/kWh in Egypt.

Keywords

Solar energy, solar cooling, steam jet ejector chiller, parabolic trough collector, vacuum collector, cold water generation

1 Introduction

The rapid growth of energy demand and the changing situation in the energy sector with more

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awareness on the environmental impact have lead to a higher interest in the utilisation of renewable energies. Due to the fact, that solar insolation generally matches the cooling load profile of buildings solar cooling techniques have been in the focus of scientific interest in the last years. The development of solar cooling has mainly been focused on solar driven “sorption” cooling processes as, for example, absorption chillers, adsorption chillers and Dessorptive and Evaporative Cooling (DEC). These technical solutions are complicated in design and in construction and there is still a development demand to optimise these systems.

An interesting alternative are Steam Jet Ejector Chillers (SJEC), which can be driven by solar energy, too. A SJEC is simple in design and reliable in operation. It has a good part load behaviour, which leads to a high yearly mean Coefficient of Performance (COP). Water can be used as refrigerant and furthermore as only working fluid in the whole system. At present, solar driven SJECs have been only studied theoretically or practically at small test rigs. The investigations have mainly been focused on SJEC with conventional refrigerants as ‘Freon’ or hydrocarbon. Experimental investigations were carried out in refs. [1, 2, 3, 4, 5, 6, 7] while theoretical analyses and simulations of SJEC with conventional refrigerants were presented in refs. [8, 9, 10, 11, 12, 13, 14, 15, 16, 17, 18, 19, 20, 21]. The solar SJEC is assessed to be reliable in operation because of its simplicity in design and in construction but it only has a moderate nominal COP value. The use of conventional refrigerants enables the operation of the SJEC with heat at a low temperature level, which can be provided by, for example, flat plate collectors.

In refs. [22, 23, 24, 25, 26] water is considered as refrigerant and working fluid. The advantages of water as refrigerant are its completed harmlessness and very good thermal properties. The necessary relatively high temperatures for steam generation can be reached by vacuum tube or parabolic trough collectors. If water is also used as heat transfer fluid, an “open” system can be designed without hydraulic separation between solar collector, SJEC and chilled water supply system.

Fraunhofer UMSICHT has investigated [27] the concept of a solar driven SJEC at a small test rig with a cold capacity of 1 kW. The test rig was operated in summer 2005 outside and in winter 2005/06 inside at a solar collector test facility at the University of Bochum in Germany. The

objectives of the project are to deliver the necessary experiential knowledge for a future pilot plant. Main subject of the investigation are the operational characteristics while changing the operational conditions and an analysis of the economical aspects. This paper describes the operational behaviour of a solar SJEC and possible efficiencies of the system.

2 Experimental Apparatus and Procedures

2.1 Description of the test rig

The test rig consists of three parabolic trough collector (PTC) modules type Roof Mounted Trough (RMT), manufactured by Solar Industrial Technology Corporation, USA, and a small SJEC for cold water generation with a designed cold capacity \dot{Q}_0 of 1 kW_{th}.

The PTC type RMT is a special development for on-roof installation. It is smaller and more light-weight than the IST-PT, which is proposed in [24] as solar collector for a SJEC. This kind of solar collector is new and the performance data of the collector have not been ascertained by the manufacturer yet. The absorber tubes in the focussing line of the solar collector are connected with each other by screw fittings so that the three modules form one trough. Metal hoses connect the PTC with the SJEC, which is installed in a movable frame. Thus, the whole test rig is mobile and can be set up at different locations. Figure 1 shows a photograph of the test rig during an outdoor test in August 2005. For inside test runs, the PTC was replaced by a vacuum tube collector (VTC) type CPC-40 Allstar with a total absorber size of 4 m². Manufacturer of the VTC is Paradigma GmbH & Co. KG, Germany.

Figure 1 Test rig during an outside test run (August 2005): 1-solar collector, 2-steam jet ejector, 3-evaporator, 4-steam drum, 5-condenser and 6-convector.

The PTC is marked with (1) in the photograph. It has a total collector size of 10.5 m². The frame with the SJEC stands close to the PTC on the right hand side of the photograph. The steam jet compressor (2) is arranged above the evaporator (3). The steam drum (4) is delivering the motive steam for the SJEC. A plate heat exchanger (5) is used as condenser and a

convective cooler (6) is simulating the cold consumer for the test runs. Figure 2 shows the process flow scheme of the test rig.

Figure 2 Process flow scheme of the test rig.

A PTC is a concentrating solar collector, which focuses the solar radiation on an absorber pipe by a parabolic mirror. The parabolic mirror is only able to focus direct normal radiation and consequently must be tracked according to the sun movement by a tracking system. The tracking system of the test rig consists of a small microcontroller to steer a direct-current motor, which is used to move the PTC around a horizontal axis via a chain drive. The absorber pipe is coated with a selectively blackened nickel surface and surrounded by tough Pyrex® glass. It transforms the radiation into thermal energy and transfers the heat energy to a heat transfer fluid, which flows through the absorber pipe. According to [28] the absorber has an absorptance of 0.96 – 0.98 and an emittance of 0.15 – 0.25. The transmittance of the glass tube is 0.95 – 0.965. Water is used as heat transfer fluid and circulates between the PTC and the steam drum, which forms the collector cycle. The collector cycle is designed as a pressurised water cycle. This means that the water does not boil in the absorber pipes of the collector but in the steam drum at lower pressure. The steam drum has a total volume of 75 l and serves as steam generator as well as heat energy buffer to ensure constant steam supply conditions for the SJEC. The maximum pressure of the solar cycle is 0,8 MPa.

The steam jet ejector compressor is the main device of the SJEC. It pumps the vaporous refrigerant from the evaporator into the condenser against a higher pressure and reduces the pressure in the evaporator. The pressure reduction causes boiling of the remaining liquid water in the evaporator and generates cold water, which is fed to the convective cooler. The steam jet ejector compressor is purchased from GEA Jet Pumps in Ettlingen, Germany, and designed for an absolute motive steam pressure p_1 of 0.3 MPa (abs), an absolute suction steam pressure p_0 of 10 hPa (abs.) and an absolute condensation pressure p_c of 50 hPa (abs.). The suction mass flow \dot{m}_0 is 1.6 kg/h and the motive steam flow \dot{m}_1 is 3.8 kg/h for the designed operation point. The evaporator has a volume of 150 l and is equipped with a sight glass. Fresh water is used as cooling water for the condenser. The motive and suction steam are liquefied together in

the condenser and fed back into the steam drum and into the evaporator respectively. A water-ring pump removes the leaked-in air from the system.

Measuring instrumentations and their locations in the test rig are also shown in the process flow diagram of the test rig. The volume flow rate over the PTC and the volume flow rate of the refrigerant from the condenser to the evaporator are measured by rotameters, while the cold water flow rate and cooling water flow rate are measured by electromagnetic flow meters. The temperatures of the solar collector cycle number 1 and 2 are measured with PT-100s which have a measuring range of 0-175 °C. The temperature sensors 3 to 6 are digital with a resolution of 0.1 °C. The incident total solar radiation on horizontal surface is measured at the solar collector level by a pyranometer with a measuring range of 0-1100 W/m² with an error of 0.1%. The measurement data acquisition system DAVID, more details in [29], is used for the data recording.

2.2 Experimental runs

The test rig was operated in summer 2005 outside in Oberhausen, Germany (51° 28'N lat. and 6° 51'E long.) and in winter 2005/06 inside in a solar collector test facility at the Ruhr University of Bochum, Germany. Figure 3 and figure 4 show the horizontal insolation I_{hor} on the solar collector and the steam drum temperature T_{drum} , as well as the suction steam pressure p_0 , the motive steam pressure p_1 and the condenser pressure p_c of the steam jet ejector for the outside test run on 17 August, 2005. The PTC was orientated in southern direction, so that its rotary axis is aligned from east to west.

Figure 3 Insolation I_{hor} and temperature curve T_{drum} of the steam drum during the outdoor test run on 17 August, 2005.

Figure 4 Motive steam pressure p_1 , suction steam pressure p_0 and condenser pressure p_c during the outdoor test run on 17 August, 2005.

The test run started at 9:00 h a.m.. The direct horizontal solar radiation I_{hor} reached

maximum values of 550 - 600 W/m². The temperature in the steam drum T_{drum} rose up above 140 °C. The SJEC was put into operation at about 1:00 p.m., which prevented a further rising of the temperature of the steam drum. The SJEC attained the aimed operation conditions. The suction steam pressure p_0 reached 10 hPa(abs.), which corresponds to an evaporator temperature T_0 of 7 °C. During the test run the suction steam pressure p_0 did not rise over 12 hPa(abs.), which corresponds to a saturated steam temperature of 10 °C. The motive steam pressure p_1 was adjusted at 0.3 MhPa(abs.) at the beginning of the test run and was then reduced down to about 0.24 MPa(abs.) due to a low cooling water temperature, which led to a low condensation pressure p_c . The refrigeration process was operated for more than 3 hours at this day.

2.3 Operational characteristics of the test rig

Main components of the solar SJEC are the solar collector and the steam jet ejector chiller, so that the operational behaviour of the system can be discussed by analysing the characteristics of the solar collector efficiency η and the COP of the steam jet ejector chiller under changing operational conditions.

The efficiency η of a solar collector can be described by the correlation according to equation (1) taken from [30], which describes the efficiency η as a function of the solar radiation I and the temperature difference between the solar collector T_{col} and the ambient air T_a .

$$\eta = \eta_0 - k1 \frac{(T_{col} - T_a)}{I} - k2 \frac{(T_{col} - T_a)^2}{I} \quad (1)$$

The parameters η_0 , $k1$ and $k2$ are related to the solar collector. The optical efficiency η_0 describes the capability of the absorber to transfer solar radiation into thermal energy. The parameters $k1$ and $k2$ are empirical values and represent the heat loss of the solar collector, which reduces the collector efficiency η .

The COP can be defined as ratio of cold capacity \dot{Q}_0 to motive heat \dot{Q}_1 according to equation (2).

$$COP = \frac{\dot{Q}_0}{\dot{Q}_1} \quad (2)$$

The product of COP of the refrigeration cycle and efficiency η of the solar collector form the total efficiency η_{tot} of the system. Based on the measurement data of the test runs the efficiency η and the COP are determined for different operational conditions. In the diagram of figure 5 the measured efficiency η of the PTC is given depending on the temperature difference between solar collector and ambient air $(T_{col} - T_a)$ for the solar radiation I of 300, 500 and 700 W/m². Furthermore the efficiency curves $\eta = f(I, (T_{col} - T_a))$ according to equation (1) are calculated and also plotted into diagram.

Figure 5 Measured collector efficiencies η for different insolation and temperature difference $(T_{col} - T_a)$.

The efficiency η of the PTC decreases at rising temperature difference $(T_{col} - T_a)$ and vice versa. A higher solar radiation I leads to a higher collector efficiency η . The parameters η_0 , k_1 and k_2 of equation (1) have been determined by a successive adjustment so that the efficiency curves $\eta = f(I, (T_{col} - T_a))$ correspond to the measured efficiencies η . The investigated parameters of the PTC are given in table 1 and compared to the parameters of the VTC, which were provided by the manufacturer.

Table 1 Parameters of the efficiency-equation.

The optical efficiency η_0 of the PTC is worse than the optical efficiency η_0 of the VTC. This means that the absorber of the VTC converts the solar radiation better into heat energy than the absorber of PTC does. On the other hand it must be mentioned that the heat loss of the PTC is lower than the heat loss of the VTC because of the reduced absorber surface of the PTC. The

heat capacity c of the VTC is higher than the heat capacity c of the PTC which means that the thermal power output of the solar collector reacts less sensitive to changes in the solar irradiation.

The operational behaviour of the SJEC is characterised by the operational behaviour of the steam jet ejector. Two important process parameters of the steam jet ejector are the suction mass flow \dot{m}_0 and the motive steam mass flow \dot{m}_1 . The suction mass flow \dot{m}_0 is direct proportional to the cold capacity of the SJEC and the motive steam mass flow \dot{m}_1 is direct proportional to the necessary motive heat power. With the objective to describe the operational behaviour of the steam jet ejector, both parameters have been determined under different operational conditions. The diagram in figure 6 shows the suction mass flow \dot{m}_0 of the steam-jet compressor depending on the suction pressure. The suction mass flow \dot{m}_0 and the motive steam mass flow \dot{m}_1 are not measured directly but derived by an energy balance over the evaporator and an energy balance over the steam drum based on the measurement data. More details about test runs can be found in [27].

Figure 6 Suction mass flow \dot{m}_0 depending on suction pressure p_0 .

The suction mass flow \dot{m}_0 rises with rising suction pressure p_0 . This is a typical behaviour of all compressors. A typical behaviour of steam-jet compressors is an almost constant suction mass flow \dot{m}_0 while raising the motive steam pressure p_1 as shown in the diagram of figure 7. The suction pressure p_0 amounts 10 to 12 hPa(abs.) for the indicated values.

Figure 7 Motive steam mass flow \dot{m}_1 and suction steam mass flow \dot{m}_0 in dependence of motive steam pressure p_1 at a suction pressure p_0 of 10 – 12 hPa(abs).

The typical flow condition in a steam jet ejector is supersonic. Consequently, the mass flow through the steam jet ejector is limited by the narrowest diameter of the steam jet ejector which is located at the end of the mixing chamber and at the beginning of the diffuser and the steam

condition of the motive steam. A higher motive steam pressure p_1 leads to a higher motive steam consumption and improves the ability of the steam jet ejector compressor to feed against a higher back-pressure p_c . But the suction steam mass flow \dot{m}_0 remains constant as long as the suction steam pressure p_0 is constant. The back-pressure p_c is almost equal to the condenser pressure of the SJEC and is related to the temperature of the re-cooling water. It has no influence on the pumping capacity of the steam jet ejector compressor. This coherence permits to reduce the motive steam pressure p_1 and consequently the motive steam consumption when the temperature of the re-cooling water decreases while the suction mass flow \dot{m}_0 remains constant. But the possibility to reduce the steam pressure p_1 is limited. There is a minimum of motive steam pressure p_1 which is necessary for the steam jet ejector to feed against a given condenser pressure p_c . If the motive steam pressure p_1 is too low to feed against the respective condenser pressure p_c the suction steam mass flow \dot{m}_0 suddenly breaks down and the refrigeration process stops. These critical operating points were identified during the test runs and the necessary motive steam flow \dot{m}_1 , due to the required minimum motive steam pressure p_1 , were determined. The necessary motive steam flow \dot{m}_1 for the tested SJEC is shown in the diagram of figure 8 related to the temperature of the evaporator T_0 and temperature of the condenser T_c .

Figure 8 Necessary motive steam mass flow \dot{m}_1 depending on the condenser temperature T_c and evaporator temperature T_0 .

The test runs show that a decrease of the condenser temperature T_c of 5 K permits to reduce the motive steam consumption by more than 30 %. A higher evaporator temperature T_0 , which is related to the suction pressure p_0 , also permits to save motive steam. With the determination of the motive steam mass flow \dot{m}_1 and the suction steam mass flow \dot{m}_0 , the COP can be calculated according to equation (2). Figure 9 shows the COP of

the investigated SJEC related to the evaporator temperature T_0 and the condenser temperature T_C .

Figure 9 COP depending on the condenser temperature T_C at different evaporator temperatures T_0 .

With decreasing condenser temperature T_C , the COP rises. Below a condenser temperature T_C of about 20 °C, the COP reaches values above 1. At an evaporator temperature of 13 °C, a COP of above 1 can still be reached at a condenser temperature T_C of 25 °C. Assuming a suitable control concept of the motive steam pressure p_1 the SJEC is able to reach a high COP value at good re-cooling conditions. Furthermore, the COP can be improved by raising the cold water temperature. Thus the SJEC can reach high COP-values at part load. In view of a typical demand of a cold supply, with low operating time at full load and governing operating time in part load and good re-cooling conditions over the year, a high annual mean COP can be attained.

3 Mean efficiencies and economical aspects

3.1 Calculated annual mean efficiencies and annual mean COP values

To appraise the operational characteristics of a solar driven SJEC ascertained before the annual mean collector efficiency, the annual mean COP of the SJEC and the annual mean total efficiency of a solar driven SJEC was calculated by an annual calculation based on weather data from the Meteonorm [31] data base as input values of the calculation. The solar SJEC was modelled and quasi dynamic simulations with an hourly time step were made over one year for the locations: Essen in Germany, Toulouse in France, Genova in Italy, Safi in Morocco and St. Katrine in Egypt. The simulations were made with the PTC type RMT discussed above and the VTC type Allstar. Additionally, also PTC type PT of the manufacture IST was simulated. The type PT is the standard PTC of the company IST and has got a bigger parabolic mirror than the type RMT. According to the manufacturer, the optical efficiency η_0 is 0.6931, the parameter for

the heat loss k_1 is 0.4755 and k_2 is 0.003125. The annual specific heat gains and the mean efficiencies of the solar collectors are given in table 2.

Table 2 Specific heat gain and mean efficiency of the solar collectors.

Apart from St. Katrine with a solar radiation ratio of direct to global of nearly 0.7, the VTC have a better specific gain of heat per square meter and year than the PTC. The PTC type PT has got the best annual mean efficiency of all three solar collectors. The smaller PTC type RMT can not compete with the PTC types PT or VTC. Its advantage is its light construction which enables an on-roof installation.

The SJEC was simulated in analogy to the solar collectors with the operational behaviour described above. The operational conditions were simulated hourly and the COP value and the cold capacity were calculated for each time step. For the calculation the cold capacity was classified into three load conditions (50, 75 and 100% of cold capacity), similar to the concept of integrated part load values (IPLV) according to ARI Standard 550/560/590-98. A defined cooling start temperature served as switch-on condition for the SJEC. The mean COP is evaluated for each load condition and afterwards the annual mean COP is determined. In table 3 the results of the simulation are given.

Table 3 Simulated mean COP values of the SJEC.

The COP at part load is higher than the COP at full load under nominal operational condition. This leads to a higher annual mean COP than under full load condition. Furthermore, the SJEC reaches good annual mean COP values at locations with a low maximal wet-bulb temperature, especially in St. Katrine in Egypt.

3.2 Economic aspects of a solar SJEC

Finally the economic aspects of a solar SJEC for cold water generation for air conditioning are calculated based on the simulation results. At first the heat price is calculated by using a PTC of the type PT and a VTC of the type Allstar. The heat price is calculated for a collector area of 100, 500 and 5000 m² for all five locations. In table 4 the results of the economical calculation are given.

Table 4 Calculation of the heat costs induced by PTC and VTC.

The calculation shows that PTCs are more economic for large-area collector fields. Furthermore, the location of an installation, especially the ratio I_{dir} / I_{global} , has influence on the solar yield and its economy. At locations with a lower ratio I_{dir} / I_{global} , the VTC could be more suitable than the PTC.

Last but not least, the costs of the cold generation by SJEC are calculated for the five locations. The cold capacity is assumed to be 200 kW. The COP as well as the operating time are based on the same simulation as before. The results of this calculation are given in table 5.

Table 5 Calculation of the cold costs with SJEC.

The specific cold costs range from 0.619 €/kWh in Essen, Germany, to 0.147 €/kWh in St. Katrine, Egypt (the specific cold costs of electrical driven refrigerators with the assumption above: 0.11 €/kWh in Germany and 0.07 €/kWh in Egypt). The investment costs have the main relevance on the cold costs. A future standardisation of the system, especially of the SJEC, would lead to a further reduction of the investment costs and the related cold costs.

4 Summary and Prospects

The solar driven SJEC is an interesting device for solar cooling. Previously, solar driven SJECs with water as refrigerant and working fluid have only been investigated theoretically. At the moment, there are neither demonstration plants nor prototypes and the operational behaviour of such a system is unknown. Due to the fact that SEJC are simple in construction they have a high cost reduction potential. Furthermore, water can serve as only working fluid in the whole system, which will guarantee an ecologically friendly concept of chilled water supply.

Within the research work presented, a small test rig was used to investigate the concept of a solar driven SJEC with parabolic trough collectors. The operational behaviour of the test rig is described by the efficiency η of the solar collector and the COP of the SJEC. The SEJC shows high COP-values at low condenser pressure p_c related to good re-cooling conditions and in part load when the cold water temperature can be increased. In view of a typical demand of a cold supply, with low operating time at full load and governing part load with good re-cooling

conditions over the year, a high yearly mean COP is expected.

Based on the ascertained operational behaviour, a quasi dynamic simulation was executed to calculate the annual mean efficiency of the solar collector, the annual mean COP of the SJEC and the annual mean total efficiency of the system. The simulations were made for the locations Essen in Germany, Toulouse in France, Genova in Italy, Safi in Morocco and St. Katrine in Egypt. The calculation proved the expected higher mean COP compared to the nominal COP, which leads to a higher mean total efficiency η_{tot} of the whole system.

Based on the results of the simulations, a first economical calculation leads to a specific cold price of 0.62 €/kWh in Germany to 0.15 €/kWh in Egypt.

Currently, Fraunhofer UMSICHT together with partners is planning to develop a small SJEC with a chiller capacity of about 10 kW as a standard device for solar cooling.

5 Nomenclature

Symbol	Quantity	SI-Unit
Abs.	Absorption Chiller	
abs.	absolute	
COP	Coefficient of Performance	
DEC	Desiccative Evaporation Cooling	
ILPV	Integrated Part Load Value	
PT	Parabolic Trough	
PTC	Parabolic Trough Collector	
RMT	Roof Mounted Trough	
SJEC	Steam Jet Ejector Chiller	
VTC	Vacuum Tube Collector	
c	heat capacity	kJ/m ² /K
I_{hor}	insolation horizontal	W/m ²
I_{global}	insolation global	W/m ²

I_{dir}	insolation direct	W/m ²
I	insolation collector	W/m ²
k_1	linear heat losses	W/m ² /K
k_2	quadratic heat losses	W/m ² /K ²
\dot{m}_0	suction mass flow	kg/h
\dot{m}_t	motive mass flow	kg/h
\dot{Q}_0	cold capacity	kW
\dot{Q}_1	motive heat	kW
$Q_{PTC,N-S}$	heat energy of PTC, north-south orientated	kWh/m ² /a
p_1	motive pressure	Pa
p_0	suction pressure	Pa
p_c	condenser pressure	Pa
T_0	temperature evaporator	°C
T_{drum}	temperature drum	°C
T_c	temperature condenser	°C
T_{col}	temperature collector	°C
T_a	temperature ambient	°C
t	time	h
η	efficiency collector	
η_0	optical efficiency of collector	
η_t	total efficiency	
$\eta_{PTC,N-S}$	annual mean efficiency, north-south orientated	
η_{tot}	total efficiency	

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Figures of the Manuscript SOLAR DRIVEN STEAM JET EJECTOR CHILLER

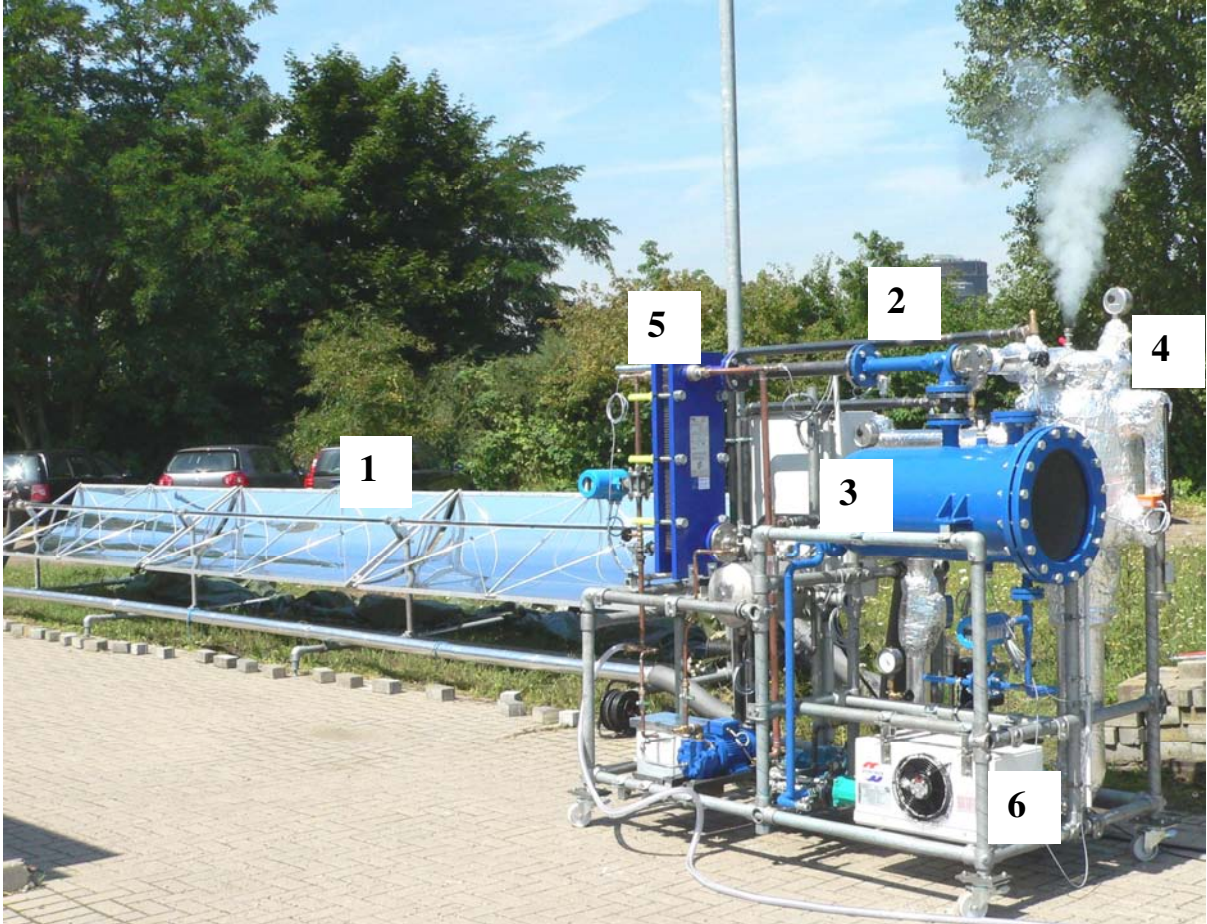


Figure 1 Test rig during an outside test run (August 2005): 1-solar collector, 2-steam jet ejector, 3-evaporator, 4-steam drum, 5-condenser and 6-convective.

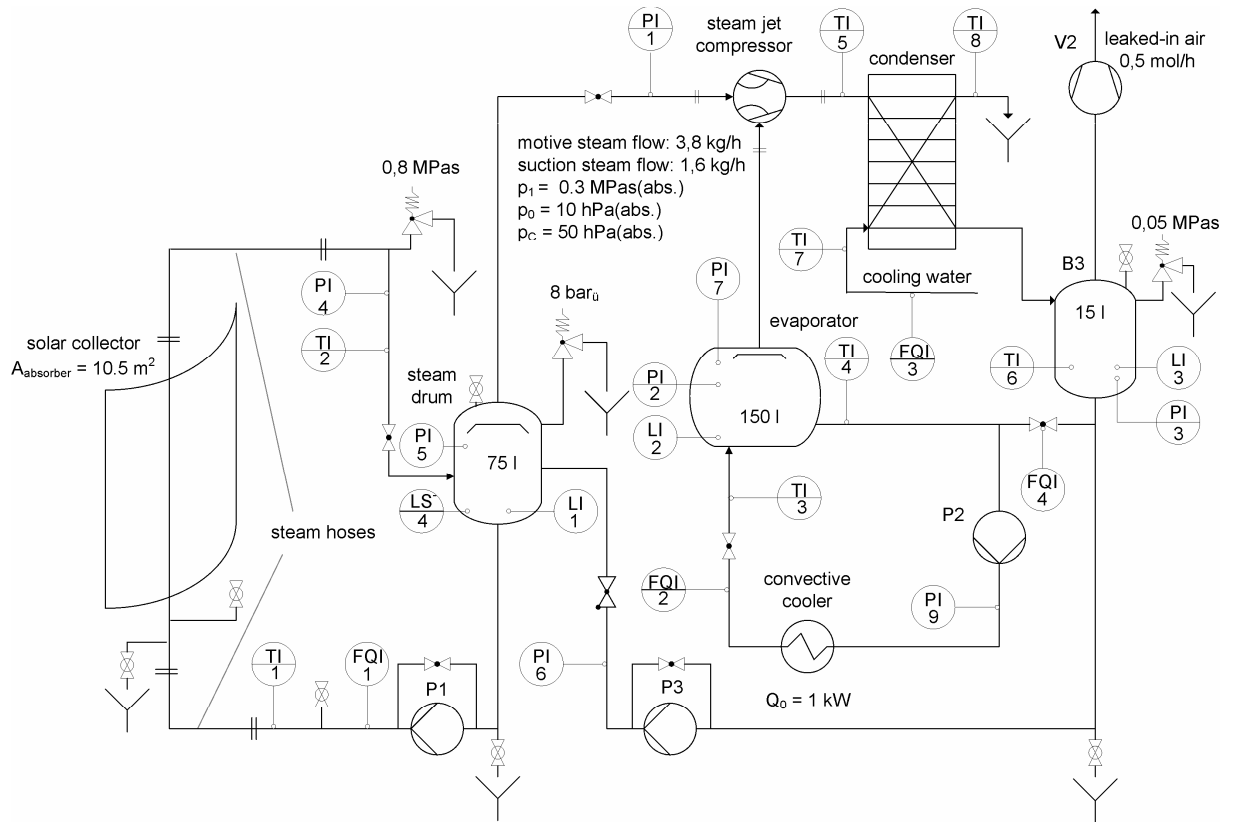


Figure 2 Process flow scheme of the test rig.

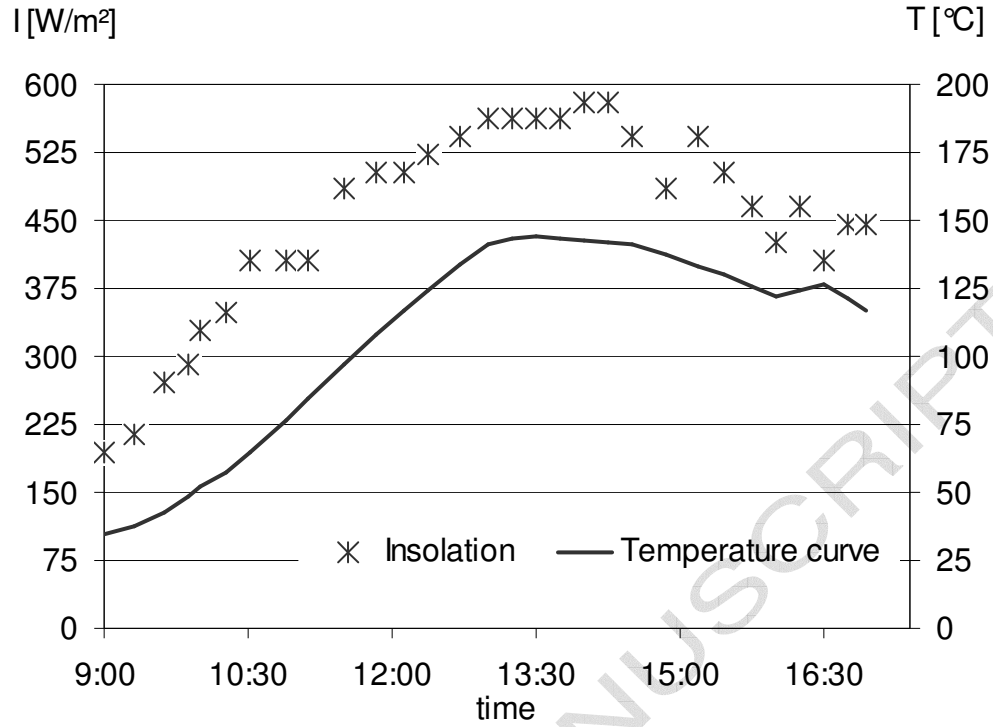


Figure 3 Insolation I_{hor} and temperature curve T_{drum} of the steam drum during the outdoor test run on 17 August, 2005.

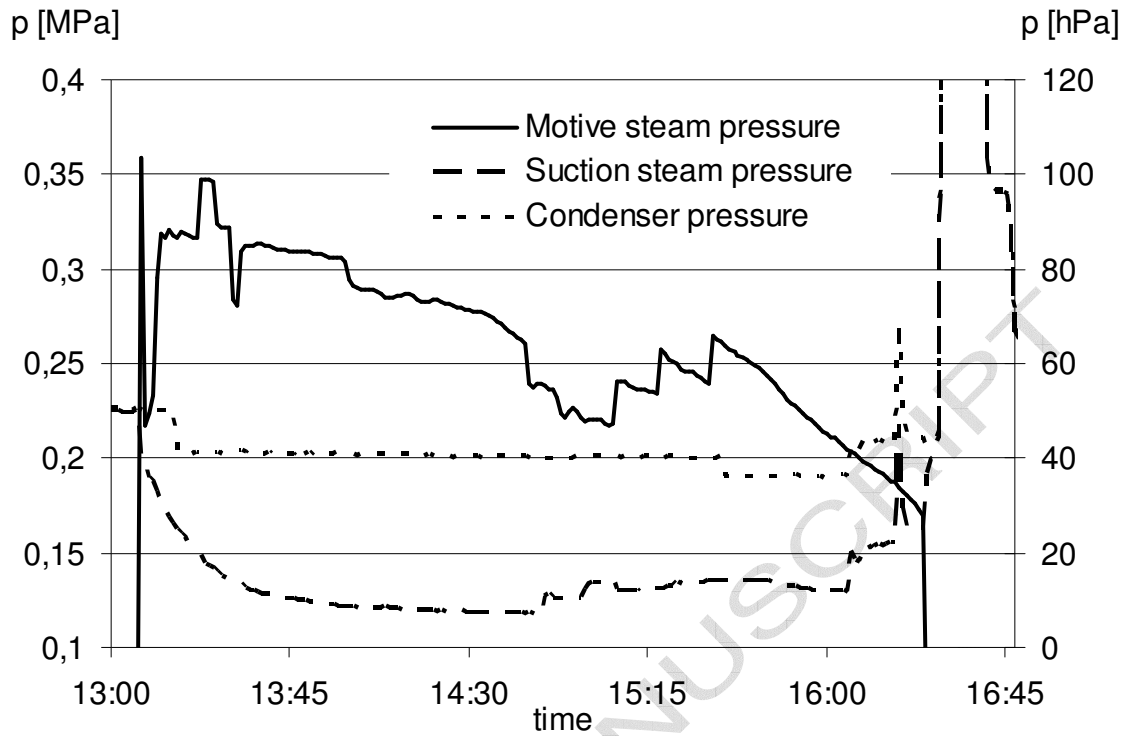


Figure 4 Motive steam pressure p_1 , suction steam pressure p_0 and condenser pressure p_c during the outdoor test run on 17 August, 2005.

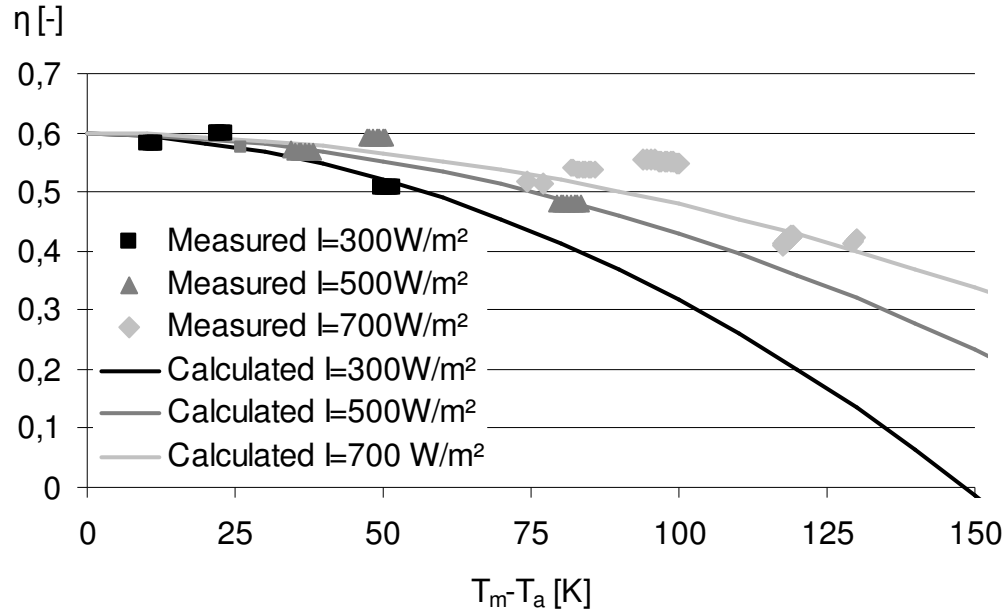


Figure 5 Measured collector efficiencies η for different solar radiation and temperature difference ($T_{col} - T_a$).

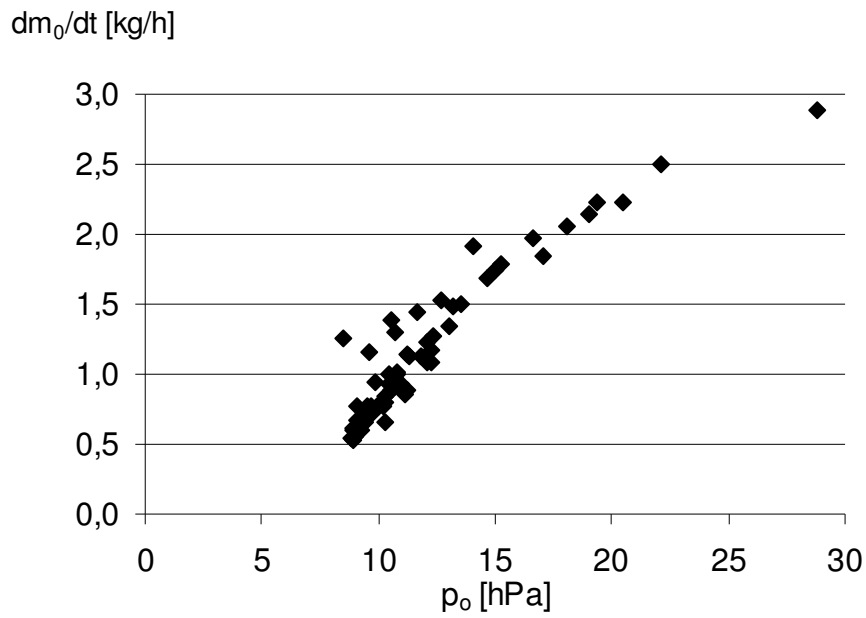


Figure 6 Suction mass flow \dot{m}_0 depending on suction pressure p_0 .

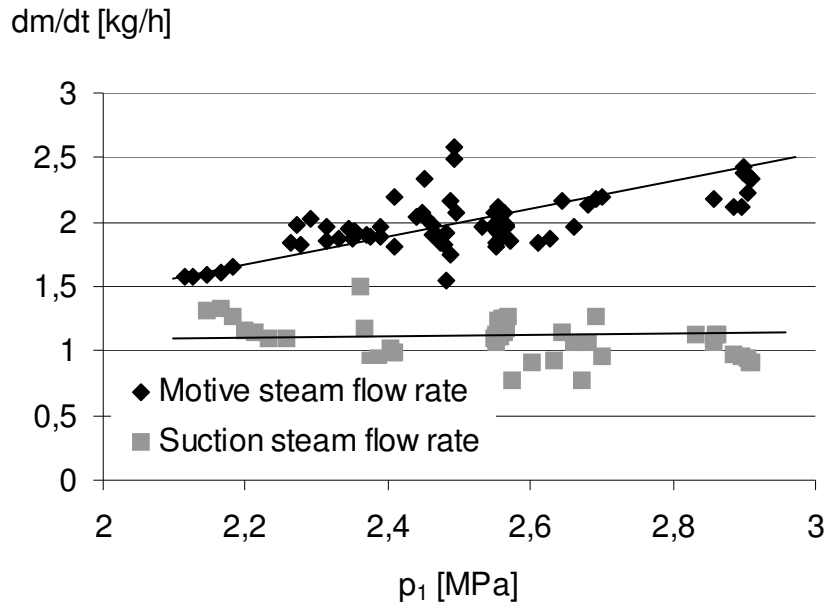


Figure 7 Motive steam mass flow \dot{m}_1 and suction steam mass flow \dot{m}_0 in dependence of motive steam pressure p_1 at a suction pressure p_0 of 10 – 12 hPa(abs.).

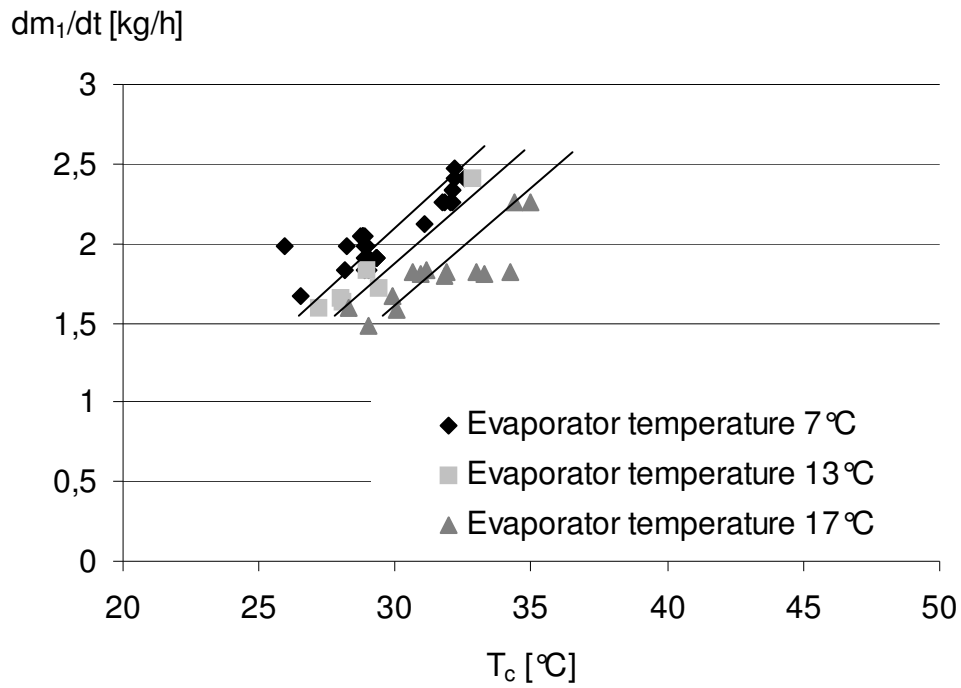


Figure 8 Necessary motive steam mass flow dm_1/dt depending on the condenser temperature T_c and evaporator temperature T_0 .

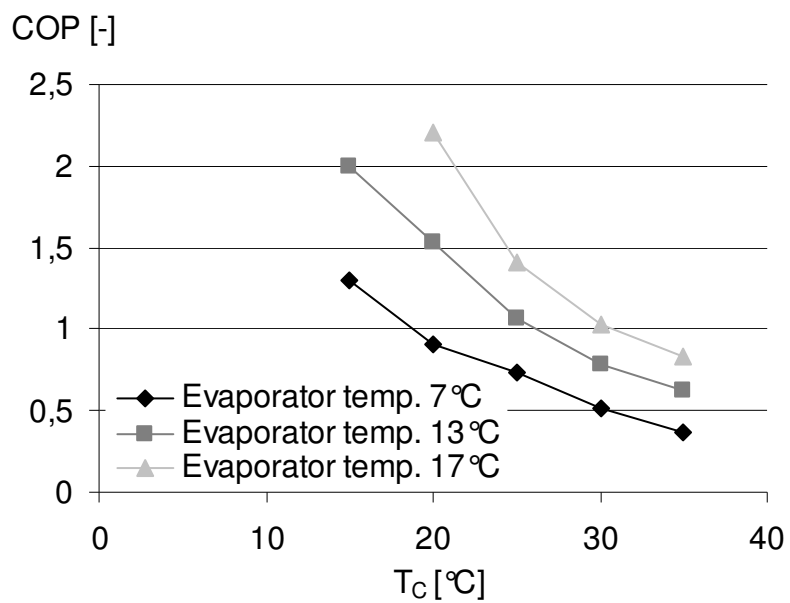


Figure 9 COP of the SJEC depending on the condenser temperature T_C at different evaporator temperatures T_0 .

Tables of the Manuscript SOLAR DRIVEN STEAM JET EJECTOR CHILLER

Parameter	Value PTC	Value VTC	Unit
Optical efficiency η_0	0.6	0.64	[-]
Linear heat loss k_1	0.1	0.688	[W/m ² /K]
Quadratic heat loss k_2	0.0075	0.004	[W/m ² /K ²]
Heat capacity c	7.5	13.06	[kJ/m ² /K]

Table 1 Parameters of the efficiency-equation.

Location	Essen, DE	Toulouse, FR	Genova, IT	Safi, MA	St. Katrine, EG	Unit
Latitude	51.2	43.4	44.3	32.2	28.4	[°]
Ratio. I_{dir}/I_{global}	0.36	0.48	0.44	0.57	0.69	[-]
Specific heat $Q_{PTC,RMT}$	87	198	171	375	643	[kWh/m ² /a]
Specific heat $Q_{PTC,PT}$	140	317	260	553	874	[kWh/m ² /a]
Specific heat Q_{VTC}	242	443	407	687	855	[kWh/m ² /a]
Mean $\eta_{PTC,RMT}$	0.18	0.23	0.22	0.28	0.35	[-]
Mean $\eta_{PTC,PT}$	0.27	0.35	0.33	0.41	0.48	[-]
Mean $\eta_{PTC,VTC}$	0.23	0.3	0.29	0.36	0.39	[-]

Table 2 Specific heat and mean efficiency of the solar collectors.

Location	Essen, DE	Toulouse, FR	Genova, IT	Safi, MA	St. Katrine, EG	Unit
Max. wet-bulb temperature	22	24	27	29	21	[°C]
Ratio wet-bulb/ air temperature	0.91	0.89	0.84	0.96	0.62	[-]
cooling start temperature	12	13	13.3	13.6	15	[°C]
$t_{\text{operating}}$	840	1216	1663	2053	1958	[h]
$t_{\text{operating},50\%}$	620	770	1047	1172	977	[h]
$t_{\text{operating},75\%}$	180	330	562	690	657	[h]
$t_{\text{operating},100\%}$	40	116	54	191	316	[h]
$\text{COP}_{\text{mean},50\%}$	1.08	1	0.97	0.89	1.2	[-]
$\text{COP}_{\text{mean},75\%}$	0.84	0.74	0.68	0.64	0.98	[-]
$\text{COP}_{\text{mean},100\%}$	0.7	0.63	0.56	0.51	0.87	[-]
COP_{mean}	1.05	0.95	0.92	0.83	1.13	[-]

Table 3 Simulated mean COP values of the SJEC.

Writing off period	15 [a]	q	1.06
Interest on capital	6 [%]		
Design temperature of the solar collector	150 [°C]		

Location	Essen, DE	Toulouse, FR	Genova, IT	Safi, MA	St. Katrine, EG	Unit
Latitude	51.2	43.4	44.3	32.2	28.4	[°]
Yearly mean ratio. I_{dir}/I_{global}	0.36	0.48	0.44	0.57	0.69	[-]
Q_{VTC}	242	443	407	687	855	[kWh/m ² /a]
Q_{PTC}	145	317	266	553	874	[kWh/m ² /a]
$Q_{PTC,100m^2}$	14500	31700	26600	55300	87400	[kWh/a]
$Q_{PTC,500m^2}$	72500	158500	133000	276500	437000	[kWh/a]
$Q_{PTC,5000m^2}$	725000	1585000	1330000	2765000	4370000	[kWh/a]
$Q_{VTC,100m^2}$	24200	44300	40700	68700	85500	[kWh/a]
$Q_{VTC,500m^2}$	121000	221500	203500	343500	427500	[kWh/a]
$Q_{VTC,5000m^2}$	1210000	2215000	2035000	3435000	4275000	[kWh/a]
Investment PTC, 100m ²	55150	55150	55150	55150	55150	[€]
Investment PTC, 500m ²	208500	208500	208500	208500	208500	[€]
Investment PTC, 5000m ²	1546875	1546875	1546875	1546875	1546875	[€]
Investment VTC, 100m ²	86594	86594	86594	86594	86594	[€]
Investment VTC, 500m ²	342025	342025	342025	342025	342025	[€]
Investment VTC, 5000m ²	3125000	3125000	3125000	3125000	3125000	[€]
Investment costs PTC, 100m ²	5678	5678	5678	5678	5678	[€/a]
Investment costs PTC, 500m ²	21468	21468	21468	21468	21468	[€/a]
Investment costs PTC, 5000m ²	159271	159271	159271	159271	159271	[€/a]
Investment costs VTC, 100m ²	8916	8916	8916	8916	8916	[€/a]
Investment costs VTC, 500m ²	35216	35216	35216	35216	35216	[€/a]
Investment costs VTC, 5000m ²	321759	321759	321759	321759	321759	[€/a]
Operating costs PTC, 100m ²	312	312	312	312	312	[€/a]
Operating costs PTC, 500m ²	1181	1181	1181	1181	1181	[€/a]
Operating costs PTC, 5000m ²	8760	8760	8760	8760	8760	[€/a]
Operating costs VTC, 100m ²	223	223	223	223	223	[€/a]
Operating costs VTC, 500m ²	880	880	880	880	880	[€/a]
Operating costs VTC, 5000m ²	8044	8044	8044	8044	8044	[€/a]
Costs of heat PTC, 100m ²	0.413	0.189	0.225	0.108	0.069	[€/kWh]
Costs of heat PTC, 500m ²	0.312	0.143	0.170	0.082	0.052	[€/kWh]
Costs of heat PTC, 5000m ²	0.232	0.106	0.126	0.061	0.038	[€/kWh]
Costs of heat VTC, 100m ²	0.378	0.206	0.225	0.133	0.107	[€/kWh]
Costs of heat VTC, 500m ²	0.298	0.163	0.177	0.105	0.084	[€/kWh]
Costs of heat VTC, 5000m ²	0.273	0.149	0.162	0.096	0.077	[€/kWh]

Table 4 Calculation of the heat costs provided by PTC and VTC.

Location	Essen, DE	Toulouse, FR	Genova, IT	Safi, MA	St. Katrine, EG	Unit
Electricity price	0.12	0.12	0.12	0.1	0.1	[/kWh]
Water price	2.5	2.5	2.5	1	1	[/m ³]
dQ_c/dt	200	200	200	200	200	[kW]
dQ_H/dt	400	400	400	400	400	[kW]
$dQ_{recooling}/dt$	600	600	600	600	600	[kW]
Costs of the chiller	40000	40000	40000	40000	40000	[€]
Costs of recooling unit	29164	33844	45544	61924	59584	[€]
Additional costs	172910	172910	172910	172910	172910	[€]
Total costs	242074	246754	258454	274834	272494	[€]
Specific costs	1210	1233	1292	1374	1362	[/kWh]
COP_m	1.05	0.95	0.92	0.83	1.13	[-]
W_0	168000	243200	332600	410600	391600	[kWh/a]
W_H	160000	256000	361522	494699	346549	[kWh/a]
W_{RKW}	328000	499200	694122	905299	738149	[kWh/a]
Heat costs	45418	35103	58655	38887	17232	[/a]
Water costs	1218	1853	2577	1345	1096	[/a]
Electricity costs	1455	2246	2997	3236	3243	[/a]
Maintenance costs	4841	4935	5169	5497	5450	[/a]
Total costs	52932	44137	69399	48965	27022	[/a]
Capital costs	24925	25406	26611	28298	28057	[/a]
Total costs	58925	63821	90248	75567	57726	[/a]
Specific costs	0.619	0.337	0.327	0.213	0.147	[/kWh]

Table 5 Calculation of the cold costs with SJEC.