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Performance assessment of an integrated free cooling and solar powered single-effect lithium bromide-water absorption chiller

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Abstract

In this study, performance assessment of an integrated cooling plant having both free cooling system and solar powered single-effect lithium bromide–water absorption chiller in operation since August 2002 in Oberhausen, Germany, was performed. A floor space of 270 m^2 is air-conditioned by the plant. The plant includes 35.17 kW cooling (10-RT) absorption chiller, vacuum tube collectors' aperture area of 108 m^2, hot water storage capacity of 6.8 m^3, cold water storage capacity of 1.5 m^3 and a 134 kW cooling tower. The results show that free cooling in some cooling months can be up to 70% while it is about 25% during the 5 years period of the plant operation. For sunny clear sky days with equal incident solar radiation, the daily solar heat fraction ranged from 0.33 to 0.41, collectors' field efficiency ranged from 0.352 to 0.492 and chiller COP varies from 0.37 to 0.81, respectively. The monthly average value of solar heat fraction varies from 31.1% up to 100% and the five years average value of about 60%. The monthly average collectors' field efficiency value varies from 34.1% up 41.8% and the five-year average value amounts about 28.3%. Based on the obtained results, the specific collector area is 4.23 (m^2/kW_{cold}) and the solar energy system support of the institute heating system for the duration from August 2002 to November 2007 is 8124 kWh.

Keywords: Solar cooling; Free cooling; Absorption chillers; Lithium bromide–water

1. Introduction

Conventional air-conditioning systems require high quality energy, electricity, generated from primary energy resources. However, in spite of CFC-free cooling liquids, energy consumption and pollution levels are rising. Therefore, an assessment form the ecological point of view needs to be implemented as the greenhouse gases effect remains a threat to the environment. In fact, most of the buildings' cooling demands in summer are associated with high solar energy availability, which offers an opportunity to further exploit solar energy for cooling. Solar cooling technology provides an important contribution to both economical and ecological energy supply. Thermally driven refrigeration systems are providing cold energy by using heat as motive energy. This heat can be obtained from combined heat and power systems, waste-heat sources or solar energy. Solar absorption cooling has been researched for almost a century (Pérez, 2007). To date, about 100 solar cooling air-conditioning systems are installed in Europe (K4RES-H, 2006). Their primary energy savings potential is between 30% and 60%, but these potentials are often not yet realized with the current systems. Balaras et al. (2007) cited that a total of 54 solar air-conditioning projects in Europe are in operation, 33 of them are working with lithium bromide–water absorption chillers, and, the present plant is one of them. Based on the policy of Fraunhofer Institute for Environmental, Safety and Energy Tech-

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nology (UMSICHT)-Germany, in this study, extensive search has been carried out and has been mainly focused on the available literature on solar-operated absorption chiller systems working with water as a refrigerant.

Single-effect lithium bromide-water (LiBr–H_2O) chiller, the most popular machine in solar cooling due to its low temperature operability, has been incorporated in numerous studies including the following demonstration and pilot projects. Ward and Löf (1975) reported that the first integrated system providing heating and cooling to a building by use of solar energy had been designed and installed in a residential-type building at Colorado State University. Solar heated liquid supplies heat both air circulating in the building and a lithium bromide absorption air conditioner. They reported that approximately two-thirds of the heating and cooling loads are expected to be met by solar energy. Ward et al. (1976) cited for the Solar House I, solar heating and cooling system became operational on July 1, 1974. From August 1, 1974 to January 13, 1975, approximately 40% of the cooling load was provided by solar energy. Furthermore, Ward et al. (1978) reported that the use of cool storage in conjunction with residential lithium bromide absorption chillers allows improved operating conditions of the cooling subsystem and significantly improves the seasonal average coefficient of performance of the cooling system. Pérez (2007) reported that (Sayigh and Saada, 1981) had installed a 12.31 kW cooling (3.5 RT, Refrigeration Ton, 1 RT = 3.517 kW cooling) solar-driven absorption chiller in Riyadh, Saudi Arabia. The system included flat collectors with a total area of 56 m^2, a hot water storage tank with a volume of 24 m^3 that was able to operate the system for 15 h at chilled water outlet temperatures of 17 °C. Hattem and Dato (1981) installed a solar absorption cooling system in Ispra, Italy, which consisted of a 4 kW chiller and 36 m^2 flat plate collectors and two hot water storage devices of 0.3 and 2 m^3. They reported that the chiller coefficient of performance (COP) and the overall system efficiency were 0.54% and 11%, respectively. Bong et al. (1987) designed, installed and operated a solar powered 7 kW absorption chiller in Singapore. The system included heat pipe collectors with a total area of 32 m^2, an auxiliary heater, a hot water storage tank, and a 17.5 kW cooling tower. They reported that the overall average cooling capacity provided was 4 kW, COP of 0.58 and a solar heating fraction of 39%. The fraction of the total driving heat load, which is covered by solar energy, is referred to as solar heat fraction. Al-Karaghouli et al. (1991) reported the operation results of a solar cooling system installed at the Solar Energy Research Center in Iraq, which was considered to be the largest solar cooling system at the time. The system was equipped with two absorption chillers of 211.02 kW cooling (60-RT) each, 1577 evacuated tube collectors and two thermal storage tanks of 15 m^3, five cooling towers with various backup systems. They reported that the daily average solar collection efficiency is 49%, chiller COP of 0.618 and solar heating fraction of 0.604. Yeung et al. (1992) designed, constructed and implemented a solar-driven absorption chiller of 4.7 kW nominal cooling capacity at the University of Hong Kong. The system included flat collectors with a total area of 38.2 m^2, a 2.75 m^3 hot water storage tank, a cooling tower and other auxiliary equipments. They reported that the collector efficiency was estimated at 37.5%, the annual system efficiency at 7.8% and an average solar fraction at 55%, respectively. Meza et al. (1998) reported the performance parameters of an experimental pilot solar assisted system installed in Cabo Rojo, Puerto Rico, that consisted of a 35.17 kW cooling (10-RT) absorption chiller, powered by 113 m^2 of selective surface flat plate collectors, a 5.7 m^3 hot water storage tank, a 84 kW cooling tower and other auxiliary equipments. The overall absorption system collector array efficiency was 30.5%, the nominal cooling capacity was measured at 25 kW with a COP of 0.63% and a 95% solar heating fraction. Best and Ortega (1999) summarized the results of a solar cooling project from 1983 to 1986 in Mexico. The system include 316 m^2 flat plate collectors, 30 m^3 heat storage tanks, a maximum capacity of 90 kW LiBr–H_2O absorption chiller and a maximum capacity of 200 kW cooling tower. Their system after modification improved the yearly solar heating fraction increased from 59% to 75% and the chiller efficiency varied from 0.53 to 0.73 when hot water was provided at temperatures between 75 and 95 °C at coolant water temperatures of 29–32 °C and chilled water temperatures of 8–10 °C, respectively. Li and Sumathy (2001) presented the results of a solar powered air-conditioning system with a partitioned hot water storage tank. The system employs a flat-plate collector array with a surface area of 38 m^2, a LiBr–H_2O absorption chiller of 4.7 kW cooling capacity and a two partitioned hot water storage tank of 2.75 m^3. Syed et al. (2005) reported the performance of a LiBr–H_2O absorption chiller with 35.17 kW cooling (10-RT) nominal cooling capacity driven by hot water from 49.9 m^2 flat plate collectors with 2 m^3 hot water storage tank installed at a typical Spanish house in Madrid. Since the solar system was originally designed for 10 kW cooling capacity, the absorption chiller yielded the maximum cooling capacity of only 7.5 kW and daily and period average COP were 0.42 and 0.34, respectively. Kim (2007) cited that (Storkenmaier et al., 2003) reported the development of a 10 kW water-cooled absorption chiller. The machine is capable of producing chilled water at 15 °C when the driven hot water is at 85 °C, being cooled by cooling water at 27 °C with the COP 0.74. The cooling capacity varied between 40% and 160% of the nominal capacity with the hot water temperature increasing from 56 to 105 °C. Furthermore, Kim (2007) cited that Safarik et al. (2005) presented the performance data of a water-cooled absorption chiller. The machine produced about 16 kW cooling at 15 °C at driving hot water of 90 °C and cooled by cooling water at 32 °C with COP of 0.75. Zambrano et al. (2007) presented results of a solar absorption cooling plant which has 35.17 kW cooling (10-RT) nominal cooling capacity at Seville in Spain. The plant has flat collectors with a total area of 151 m^2, a 2.5 m^3 hot water storage tank and an auxiliary gas heating system. They tabulated the instantaneous values of the solar, gas power supplied to the generator, and the chiller cooling capacity for one day running in May only, no information was cited about the long-run performance. Pongtornkulpanich et al. (2007) designed and installed a 35.17 kW cooling (10-RT) solar-driven absorption cooling system in Thailand in 2005. The system has a 0.4 m^3 hot water storage tank and 72 m^2 evacuated tube solar collectors that delivered a yearly average solar heating fraction of 81%.

Double-effect LiBr–H_2O machines were also used in some solar-driven absorption cooling projects. Due to the requirement of a high driving temperature of about 150 °C, in most cases, the hot water from concentrating solar collector is used. Kim (2007) cited that (Lokurlu and Müller, 2005) reported a system installed in Turkey, which consisted of a steam-driven double-effect machine, a trough type parabolic solar collector and a backup steam boiler. The trough collector with a 180 m^2 aperture area heated pressurized water up to 180 °C and this water in turn generated 144 °C steam (4 bar) for a 110 kW double-effect LiBr–H_2O chiller. Duff et al. (2004) reported that in 1998 a new integrated CPC reflector evacuated solar collector with modified double effect absorption chiller to operate with 150 °C hot water from the solar collector array for building cooling was installed. The 106.5 m^2 collector array consisted of 336 evacuated tubes. Daily collection efficiencies of nearly 50% and instantaneous collection efficiencies of about 60% were achieved throughout the first two years of operation. Daily chiller COP of about 1.1 was achieved as well.

To summarize, current solar-operated lithium bromide-water absorption chillers for space air-conditioning application are working in different locations worldwide having a solar heating fraction ranging from 39% to 95% with different COP values. However, to the best of our knowledge, no information have been published so far concerning an integrated combined free cooling and solar powered single-effect lithium bromide-water absorption cooling system as a combination of both cooling techniques is always limited by the prevailing local weather conditions.

This work presents a performance assessment of an integrated cooling plant with combined free cooling and solar powered single-effect lithium bromide–water (LiBr–H_2O) absorption chiller in operation for the duration of five years. The plant is a part of the infrastructure at Fraunhofer Institute (UMSICHT) in Oberhausen-Germany. The plant was driven by solar energy only from 2002 to 2004 under the typical weather conditions in the Central Europe. From 2005 until now, the plant has been integrated in the institute heating system that can provide solar heating portion in heating season and utilize the available hot water of the Institute heating system (gas boiler and micro gas turbine) as supplementary source in case the solar collector field supply heat is not enough to drive the chiller at cooling season. The plant includes a 35.17 kW cooling (10-RT) absorption chiller, vacuum tubes collectors with gross and net areas of 108 m^2 and 72 m^2, a hot water storage capacity of 6.8 m^3, a cold water storage capacity of 1.5 m^3 and a 134 kW cooling tower. The plant provides air-conditioning for a floor space of 270 m^2.

2. Plant description, measurements, data acquisition and processing system

An integrated cooling plant possessing both combined free cooling and solar-operated absorption chiller provides the cooling demands for the Fraunhofer Institute (UMSICHT) in Oberhausen-Germany (51°28′N latitude and 6°51′E longitude), laboratories, meeting rooms and three offices during the cooling season. The office rooms only require cold demand during working hours, whereas the laboratories cold demand is continuous during 24 h. The solar energy collection system is also integrated with the Institute heating system providing support to the central heating system during sunny days in heating season. Fig. 1 shows the plant schematic diagram. The major components of the plant are a roof-mounted vacuum tubes solar collector, a 35.17 kW cooling (10-RT) single-effect LiBr–H_2O absorption chiller, a hot water storage tank, a cold water storage tank, a cooling tower, a free cooling heat exchanger, a roof top heat release heat exchanger, pumps, a control system, a water treatment system, a pressure maintaining system and some other auxiliary equipments. In addition, it is comprised of four main flow circuits which are the solar circuit (with anti-freezing agent), the hot water circuit, the chilled water circuit, and the cooling water circuit. These circuits interrelate with the absorption chillers at the generator, the evaporator, the absorber and the condenser, respectively. While both the cooling water and chilled water circuits are interrelate at the free cooling heat exchanger as shown in Fig. 1. In addition, the solar circuit is connected to the hot water circuit by a plate type heat exchanger HE1.

As the demands for energy efficiency and reduction in carbon footprints continue to bite, the popularity of free cooling chillers is increasing. However, the most significant point in this plant is the free cooling, which is essentially chilled water at no cost.

2.1. Solar collector system

The plant primary energy source is the solar energy, which is converted into thermal energy by the vacuum tubes solar collector field and the water is the heat transfer medium in the circuit integrated with the hot water circuit that includes the hot water storage tank as shown in Fig. 1. The solar collector field is composed of a 108 m^2 apparent area and a 72 m^2 absorbing area for an array of 432 evacuated tubes in nine collector fields which work in the range of 97/105 °C and a capacity of 50 kW (for $I = 1000$ W/m^2), where I is the solar insolation at the collector plane. The collectors' field is mounted on a slightly tilted roof by 2°

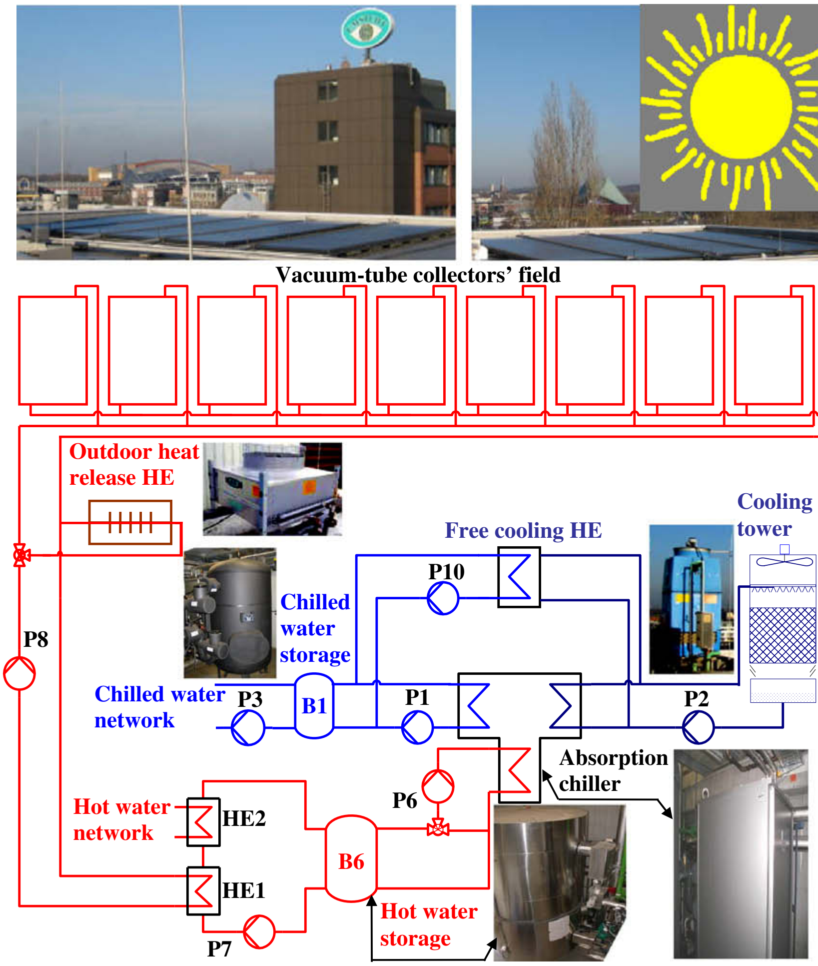

Fig. 1. A schematic diagram of the cooling plant, where B is storage tank, HE is heat exchanger and P is pump, respectively.

with the horizontal for rain derange, while the collectors absorber element is tilted by 30° with the roof. For the plant and collector field overheating protection, the hot water loop has a roof top heat exchanger that can be used to release the excess heat to the ambient in summer time when the plant is shutoff.

2.2. Absorption chiller

A Water Fired Chiller (WFC-10 RT) with a rated capacity of 35.17 kW cooling (10-RT) when it is operating at a driving hot water temperature of 87 °C, coolant water temperature of 29.5 °C and output chilled at 9 °C with coefficient of performance (COP) of 0.7 as reported by the manufacture. The cooling demand of the plant is needed from May to September as will be presented in the results section. However, as the local weather data in Oberhausen city, Germany has a wet-bulb temperature in cooling season ranging from 15 to 22 °C. Thus, the cooling tower can provide cold water at a temperature which is less than its rated value (24/31 °C). Moreover, the vacuum tubes collector field is also able to provide driving hot water at a higher temperature than the rated value. Therefore, the chiller cooling capacity can be rising up to 58 kW with a nominal COP of 0.75, corresponding to the manufacture reported characteristics.

2.3. Cooling tower

The cooling tower is serving to reject the heat from the chiller coolant water to the ambient as well as a source of free cooling in the plant. The cooling tower is located at the rooftop of the building as shown in Fig. 1 with a capacity of 134 kW at 24/31 °C when ambient air wet-bulb temperature is 21 °C.

2.4. Thermal buffer system

The thermal buffer system is composed of two tanks. The first is the hot water storage tank, B6, as shown in Fig. 1 with a capacity of 6.8 m^3. This tank acts as a storing and buffers that supplies hot water which enters the chiller generator at constant value. The stored heat in the tank can be from either the solar system or the Institute heating system. The second tank is the cold water storage buffer with a capacity of 1.5 m^3, B1, as shown in Fig. 1. This tank acts as a storing and buffer supply of the cold water to the load. The stored coolness in the tank can be either free cooling obtained from the cooling tower through the free cooling heat exchanger or from the chiller evaporator.

2.5. Cooling load

A floor space of about 270 m^2 is air-conditioned by this plant. The load network is distributed into a central air-handling unit, which provides 100% fresh air to labs, several air cooler units at offices and convective radiators at the central computers labs of the institute. The chilled water is supplied to each of these loads installations via a supply network with a length of about 390 m.

2.6. Measurements, data acquisition and processing system

The volume flow rate at six different locations in the plant is measured by different water meters. Multi-jet dry dial vane impeller hot water meters are used in solar circuit and hot water circuit with a measuring range from 0.2 to 20 m^3/h, Woltmann water meters are used in the re-cooling circuit and the chilled water system with a measuring range from 0.45 to 90 m^3/h, and multi-jet water meters are used for tap up water and waste water with a measuring range from 0.05 to 5 m^3/h, respectively. The temperatures were measured with PT-1000 sensors in protection sleeves, with measuring ranges of 0–60 °C for chilled water and re-cooling water, 0–150 °C for hot water and 0–200 °C for the solar circuit, respectively. The resolution of the temperature measurement is 0.01 °C. The incident total solar radiation on the plane of the absorber level was measured by a Tritec Spectrum Irradiation Sensor 300 having a measuring range of 0–1500 W/m^2, with a standard 4–20 mA-signal output. The relative measurement error amounts corresponding to the statements by the manufacturers varied from 1% to 5%.

Different pumps were used to circulate the fluids in each circuit as shown in Fig. 1. In addition, flow-controlling valves integrated with plant controller system were used to adjust the flow rates.

The entire installation is fully automated, controlled and monitored by a Siemens S7-300 controller (CPU 315DP) in combination with an industry PC with OS MS Windows 2000 and the visualization software Protoolpro© Realtime V6.0. In addition, the data acquisition is managed by Protoolpro Software. The operator can manage the complete system at different automation levels or get control over each single actor and most of the controllers at the manual operation level. The following is the sequence of the plant operating procedure; if sufficient solar radiation is available, the automation system starts the pump P8 to circulate water through the collector array. The hot water-storage loading pump P7 is started when the temperature in the solar circuit is higher than the temperature at the bottom of the storage. The temperatures in the hot water storage circuit and at the top of the hot water storage are monitored. If one of them rises above a specific threshold of about 82 °C, the generator pump P6 and the control valves within the hot water supply line get the allowance of operation. In parallel, if any chilled water is needed at this moment as well, this means that the putting in operation order for the chiller is activated, the evaporator pump P1 and the re-cooling pump P2 are also set in operation and a volume flow can be measured within chilled water and re-cooling-water circuit. In this case, the generator pump P6 starts operation and the hot water valves to the generator will open. Then, the absorption chiller is started to operate and produce chilled water. The time span to reach a first capacity output from the chiller can be about 15 min. This time became longer, especially if the chiller has not been in operation for a longer time, e.g., within the transition time in spring and autumn or after wintertime. The temperature of water entering the chiller's generator is continuously measured to ensure delivery of sufficient thermal energy to drive the internal thermosyphon pump effect within the generator and to avoid crystallization. In case the hot water supplied from the hot water storage tank drops below 78 °C the backup heater is activated. In case of water entering temperature dropping below 75 °C, the chiller is shutdown. During operation, the chilled water temperature exiting the chiller is monitored to verify that the system is operated correctly. This implies that the control system must keep the cooling machine working at the desired operating point and this is achieved by keeping the machine inlet water temperature at the given set-point. The free cooling is obtained from the plant when the cooling tower outlet water temperature is in range of the supply chilled to the load. At this condition, the free cooling heat exchanger and the pump P10 are integrated in the chilled water circuit that shown in Fig. 1, by the control system.

3. Presentation of parameters

From the measured data, results are presented based on using a simple data reduction. The thermal capacity of the equipments is determined by:

$$Q = \dot{m} \cdot c_p \cdot \Delta T \ (\mathrm{kW}), \tag{1}$$

where $\dot{m}$ is the mass flow rate in kg/s, c_p is the specific heat at constant pressure in kJ/(kg C) and ΔT is the temperature difference in °C, respectively. The energy during a certain period is determined by the integration of the capacity during this time as follows:

$$E = \int_{t_o}^{t_f} Q \, \mathrm{d}t \ (\mathrm{kWh}), \tag{2}$$

where t_0 and t_f are the initial and final times. The solar collectors' efficiency (η) is determined by:

$$\eta = \frac{Q_{\mathrm{solar,net}}}{I \cdot A_{\mathrm{absorber}}}, \tag{3}$$

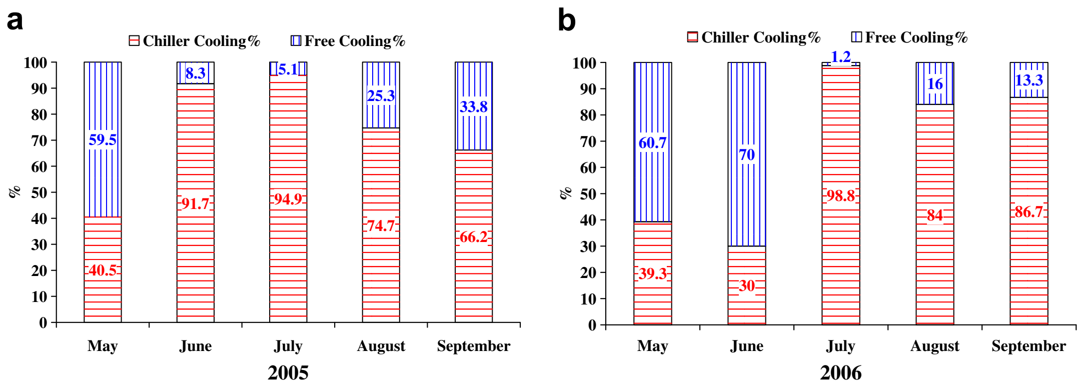

Fig. 2. Average monthly obtained cold energy percentage by free cooling to the total load energy during the years 2005 and 2006 at cooling months.

where $Q_{solar,net}$ is the net thermal power gained by the water from the collectors' field, I is the solar insolation in the collector absorber plane and $A_{absorber}$ is the total absorbing area of the collectors' field, respectively. Neglecting the chiller pump power[2], the chiller coefficient of performance, COP, is defined as the ratio of the evaporator cold capacity Q_E to the heat input to generator Q_G as follows:

$$\mathrm{COP} = \frac{Q_E}{Q_G}. \tag{4}$$

For recalculation of the specific collector area, A_s, which defines the collector area per target cold capacity from the plant, has to be determined on the basis of the results obtained when using the formula presented in Henning (2007) as follows:

$$A_s = \frac{1}{I \cdot \eta_{daily,average} \cdot \mathrm{COP}_{daily,average}} \ (\mathrm{m}^2/\mathrm{kW}_{cold}). \tag{5}$$

4. Results and discussion

There have been enormous amounts of recorded measured data from the plant since its installation in August 2002. However, samples from these measurements are used to extract the results in this contribution, which are in some cases series of results or represent the average values of five years' duration for the plant operation.

4.1. Free cooling potential

The obtained monthly average cold energy percentage by free cooling to the total cold energy during the years 2005 and 2006 at cooling months are shown in Fig. 2. As shown in the figure during these months, depending on ambient air wet-bulb temperature and required chilled water temperature for the load, substantial free cooling can be obtained from the plant. The free cooling percentage in some cooling months can be up to 70%. It varies from one month to the other and from year to year. It can be seen from the figure that in both years the free cooling value is very low for July. For the duration from August 2002 to November 2007, the total supplied cold energy from the chiller is 31,365 kWh while the free cooling is 10,299 kWh. The assessment of the free cooling potential clearly shows that it offers about 25% of the total cooling demand during the 5 years period of the plant operation. These results show an extensive potential for cooling demand by free cooling means in certain locations based on the local climate condition. In order to increase this potential, cold water by free cooling can be produced at nighttime. Moreover, increasing the cold storage capacity and use of the complete chilled water net to store the nighttime free cooling cold water has to be taken into account. It can be concluded that true environmentally efficient systems must delve into total system design. A truly green system will not merely employ a more efficient chiller cooling system; it will significantly downsize the need for that equipment from the outset using less energy-intensive components. Such system incorporates a multifunctional concept, in which individual pieces of equipment as the cooling tower serve multiple design objectives. A typical retrofit using this hybrid design reduces energy usage for cold demand by 25%.

4.2. Collector performance and solar energy heat fraction

The instantaneous performance of the vacuum tube solar collectors' field is presented by the experimentally determined collector efficiency from different measurements of clear sky days data illustrated in Fig. 3a and shown in Fig. 4. The efficiency, η, characteristics are determined according to the ASHRAE Standard 93–77 (Duffie and Beckman, 1991), which is plotted as a function of $(T_m - T_{amb})/I$. The presented data are corresponding to the values obtained 2 h before and after noon (when the solar angle is nearly normal). In which T_m is mean average temperature of the water inside the collector and T_{amb} is the ambient air-dry-bulb temperature illustrated in Fig. 3b. The chosen solar radiation data is close to the stan-

[2] As the used chiller has no solution pump.

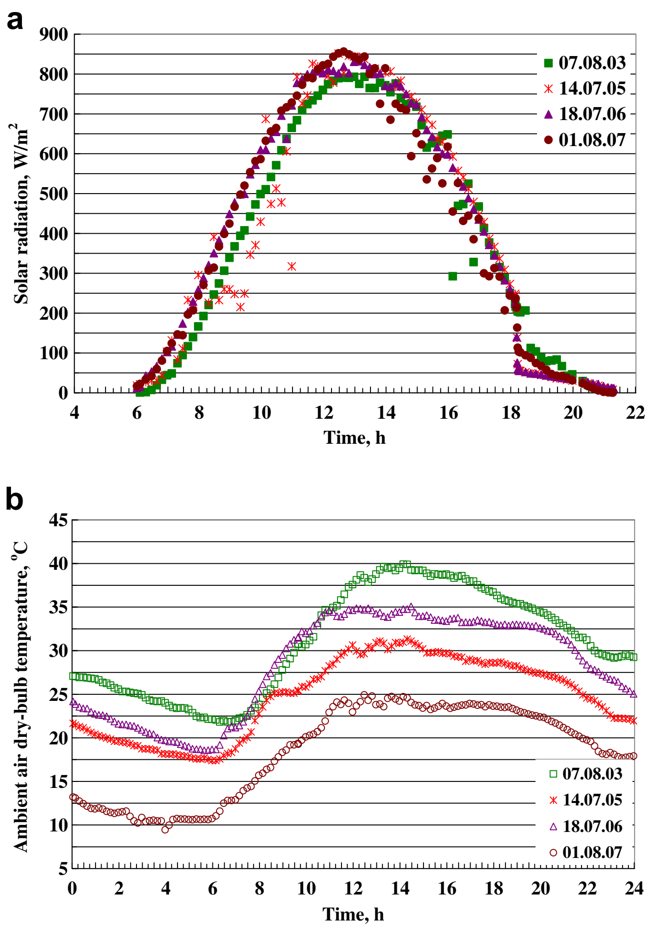

Fig. 3. Illustration of the meteorological conditions in clear sky summer days (a) incident solar insolation and (b) ambient air-dry-bulb temperature.

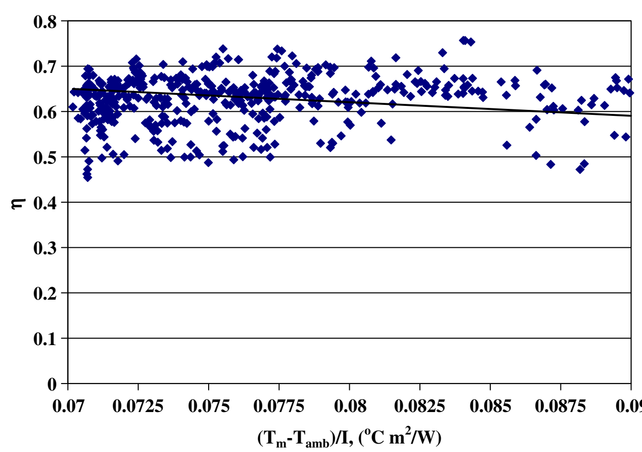

Fig. 4. Instantaneous collectors field efficiency.

dard recommended values. The results shown in Fig. 4 have an instantaneous mean collectors field efficiency value of about 0.63. This value is within the range of the performance data of the manufacturer (61.1%, related to solar irradiation of 900 W/m^2 and $T = 105/98/25$ °C) as the present operating condition is slightly different, but less than would be expected compared with many vacuum tube collectors performance in present working at similar operating condition. This result is attributed to the fact that the absorber in the vacuum tube collector's field has tilt angle of 30° with roof/horizontal while the system installed in the city at 51°28′ N latitude, therefore, this tilt angle is not the optimal orientation one. Moreover, from the visual inspection of the collectors' glass tube, a thin stick fouling material combination of residual combustion gasses and other materials over the tube surfaces is found. These fouling materials are not removable with heavy rains, is also act on the decreases of collector field performance.

The fraction of the total driving heat load, which is covered by solar energy, is referred as solar heat fraction. For the duration from August 2002 to November 2007 the total solar energy supplied to the chiller is 53,914 kWh and the total external energy (gas energy) supplied to the chiller is 35,249 kWh and their percentage are about 60% and 40%, respectively. The average monthly solar heat fraction percentage and the collectors' field efficiency during the cooling months for the year 2005 are shown in Fig. 5. The figure shows that the monthly average value of solar heat fraction varies from 31.1% up to 100%, with a five-year average value of 60%. The main factors affecting the solar heat fraction are the meteorological conditions and the time of day when the plant is operated as both influence the level of incident solar insolation. In addition, in Fig. 5 the monthly average value of the collectors' field efficiency varies from 34.1% up to 41.8%, with a five-year average value of 28.3% (not presented). These monthly and yearly average values of the collector efficiency can be enhanced by adjusting the tilting angle of the collectors' absorber. In addition, cleaning the collectors' tubes outer surface from the stick fouling once a year could be a further method to enhance the solar system efficiency.

4.3. Operational characteristics of a solar powered absorption chiller

The results obtained from instantaneous measured data for the plant used to clarify the operational characteristics of the solar powered absorption chiller are shown in Fig. 6.

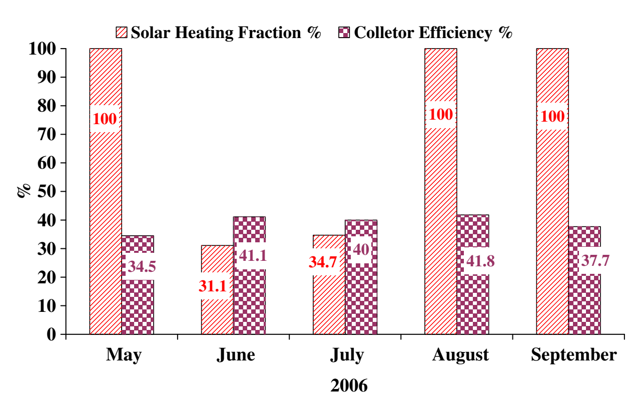

Fig. 5. The average monthly solar heat fraction percentage to the total driving heat load and the collectors field efficiency during the cooling months.

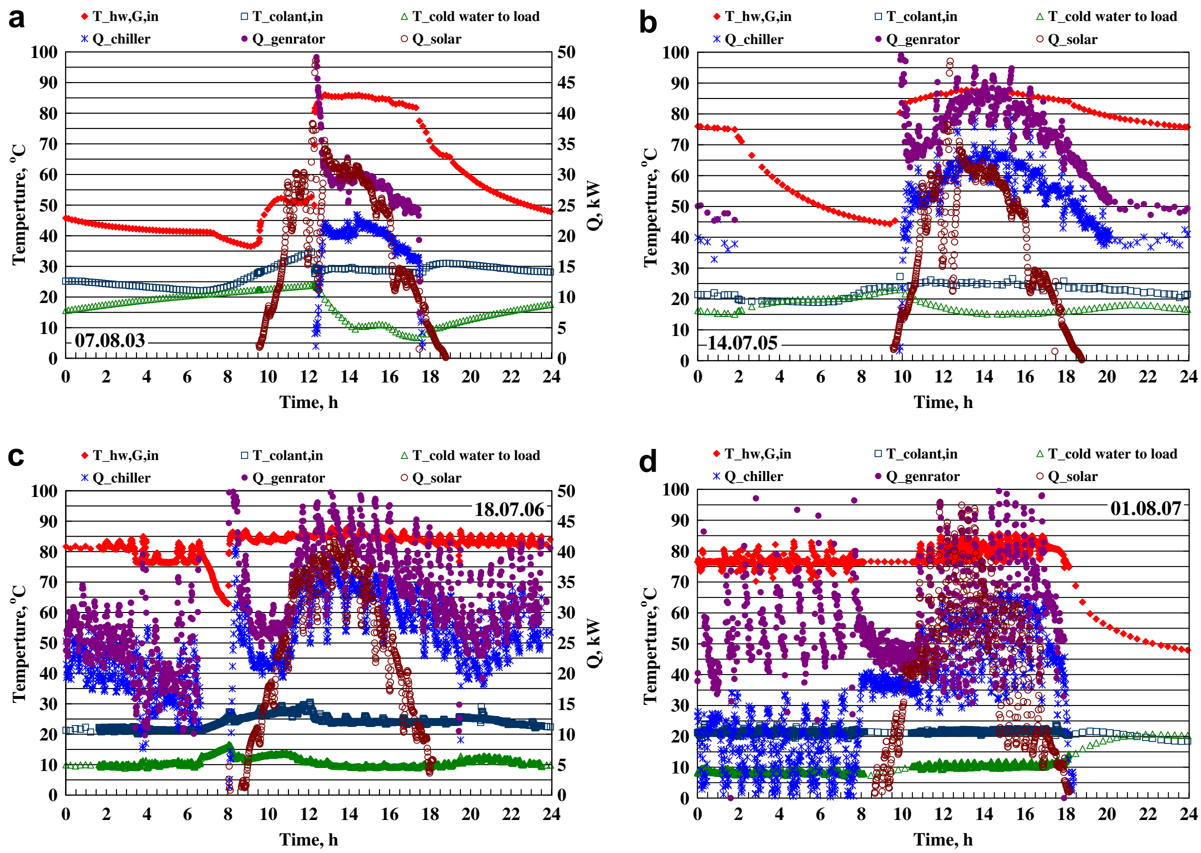

Fig. 6. Instantaneous characteristics of the plant in different years operating conditions.

The figure shows the measured inlet hot water temperature to the chiller generator (T_hw,G,in), inlet coolant water to absorber and condenser (T_colant,in), outlet chilled water temperature from the evaporator (T_cold water to load), the supplied heat power to the chiller by solar energy (Q_solar), the total supplied heat power to the generator (Q_generator) and the chiller cold power produced (Q_chiller) as a function of the daytime. The results shown in Fig. 6 are in correspondence to almost clear sky days free from clouds with meteorological data illustrated in Fig. 3. These days have previous sunny days (not presented) and have almost equal incident solar radiation. The results shown in Fig. 6a are for the case of the plant operation when the driving heat is only from solar energy source. This was before integration of the plant into the Institute heating system. As shown in Fig. 6a, the chiller was started when the temperature of the driving hot water exceeds 80 °C, it was around 12:00 h. The inlet coolant water temperature during this day was around 25 °C and the outlet chilled water temperature decreases with time until reaching 9 °C after 2 h from the beginning of operation. The total energy supplied to the chiller generator was 153.9 kWh, while the net solar heat gain was 177 kWh. The excess solar energy is stored in the hot water storage tank for use in the following day. In this day, the collector field average efficiency was 0.4, a total cold energy produced of 99.6 kWh and chiller COP was 0.65. For the five working hours of the chiller, the rated chiller average cooling capacity is about 20 kW, which is about 57.1% of its rated capacity. The results shown in Fig. 6b are for the eight working hours of the chiller after integrating the plant into the Institute heating system in 2005. From the figure, the rated chiller average cooling capacity is about 24.8 kW, which is about 71% of its rated capacity based on the load cold demand. The daily average value of the results shown in Fig. 6b are determined as follows: the total energy supplied to the chiller generator was 527.1 kWh, a net solar heat gain of 213.7 kWh with a solar heat fraction of 0.41, a collector field average efficiency of 0.481, a total cold energy produced of 198 kWh and a chiller COP of 0.38, respectively. A continuous operating day of the plant results is shown in Fig. 6c. For about 22 working hours of the chiller the rated chiller, average cooling capacity is about 26.1 kW, which is about 74.4% of its rated capacity based on the load cold demand. However, for this day the total energy supplied to the chiller generator was 709.12 kWh, net solar heat gain of 234.63 kWh with solar heat fraction of 0.331, the collector field average efficiency of 0.492, a total cold energy produced of 572.9 kWh and a chiller COP of 0.8, respectively. The results for the plant characteristics, with an outdoor air temperature of about 25 °C during the cooling demands period on a humid day, are shown in Fig. 6d. For about 18 working hours of the chiller, the rated chiller average cooling capacity was about 16.23 kW, which is about 46.4% of its rated capacity based on cold load demand. At that day, the total energy supplied to the chiller generator was 442.2 kWh, the net solar heat gain of 163 kWh with a solar heat fraction of 0.368, a collector field average efficiency of 0.352, a total cold energy produced of 292.2 kWh and a chiller COP of 0.661, respectively. From the results of the energy obtained from Fig. 6, except for Fig. 6a case, for sunny and clear sky from clouds days with almost equal incident solar radiation, it is found that the daily solar heat fraction ranged from 0.33 to 0.41 with the collector field efficiency ranging from 0.352 to 0.492 and chiller COP varying between 0.37 and 0.81, respectively.

To clarify the effect of the daily cold energy demand on the chiller COP, accumulated results for 3 years of operation are shown in Fig. 7. The data for the year 2004 with sample of daily operation condition that shown in Fig. 6a is chosen as a case of the plant operation when the driving heat is only from solar energy source. The plant control system is set to give the priority of using free cooling over the chiller cold produced, when the free cooling is available. Therefore, the variations in COP values shown in Fig. 7 are mainly attributed to the chiller operating conditions that differ from the rated conditions. From the plant data recorded, it is found that the daily average COP of the chiller decreases at lower cold demand is mainly due to a greater portion of the daily cold load demanded of the Institute building is covered by the free cooling, while the chiller covers the rest of the load. From Fig. 7, it can be seen that as the load cold demand is around 600 kWh/day in hot days, the average daily COP of the chiller is around 0.6–0.7, which is close to the chiller nominal value reported by the maker. Under such condition with absence of the free cooling in the plant, the chiller is working with the supplied nominal COP values reported by the manufacture.

4.4. Solar system optimization

The specific collector area in the plant location (Central Europe) based on the obtained results for sunny days presented above considering the average value of I is 750 W/m^2, the average collector efficiency of 0.45 and COP value of 0.7 is recalculated. The new value is 4.23 (m^2/kW$_{cold}$) which has to be considered as suggestion of specific collector area in north-west Germany in order to harvest maximum possible solar heat fraction for thermal driving power of absorption chiller.

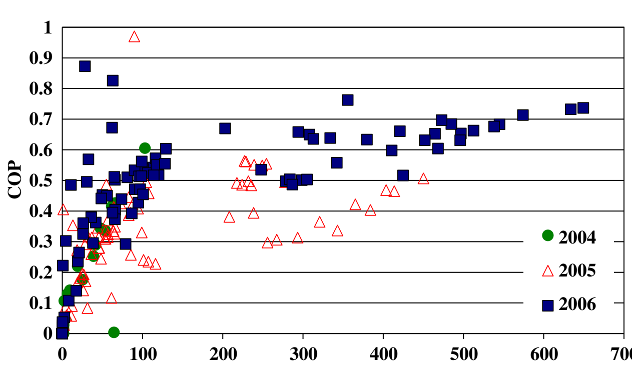

Fig. 7. Dependence of the chiller daily average coefficient of performance on the cold energy demand.

It is worth mentioning that the support of the plant solar energy system to the institute heating system for the duration from August 2002 to November 2007 is 8125 kWh. A climate condition in Central Europe is a convenient possibility to reduce primary energy consumption in both cooling and heating applications. In view of increasing energy prices, solar energy technology could be an alternative to conventional systems.

5. Conclusion

In this study, performance assessment of an integrated cooling plant with combined free cooling and solar powered single-effect lithium bromide-water absorption chiller in operation for the duration of five years is presented. The plant provides the air-conditioning and chilled water demand for Fraunhofer Institute UMSICHT in Oberhausen-Germany since August 2002. From 2005 until now the plant has been additionally operating in connection to the institute heating system in order to use excess solar heat for heating purposes and to utilize the available hot water of the Institute heating system as a supplementary source in case the solar collector field supply heat is not enough to drive the chiller at cooling season. The plant includes a 35.17 kW cooling (10-RT) absorption chiller, vacuum tubes collectors with gross and net areas of 108 m^2 and 72 m^2, a hot water storage capacity of 6.8 m^3, a cold water storage capacity of 1.5 m^3 and a 134 kW cooling tower. The plant provides air-conditioning for a floor space of 270 m^2. The main findings of the present study can be summarized as follows:

- The free cooling percentage in some cooling months can be up to 70%; in addition, it offers 25% of the total cooling demand during a 5 years period of plant operation. It can be concluded that true environmentally efficient systems must delve into total system design. It will significantly downsize the need for that equipment in which individual pieces of equipment as the cooling tower serve multiple design objectives.
- The monthly average value of solar heat fraction varies from 31.1% up to 100% and the five years average value of about 60%.
- The instantaneous mean collectors' field efficiency value was about 0.63, the monthly average value varied from 34.1% up to 41.8% and the five-year average value of 28.3%.
- For sunny days with almost equal incident solar radiation and clear sky from clouds, the daily solar heat fraction ranged from 0.33 to 0.41, collectors' field efficiency ranged from 0.352 to 0.492 and chiller COP varied from 0.37 to 0.81, respectively.

- Based on the obtained results, the specific collector area is 4.23 (m^2/kW_{cold}) in order to harvest maximum possible solar heat fraction for driving an absorption chiller.
- The support of the plant solar energy system to the institute heating system for the duration from August 2002 to November 2007 is 8125 kWh.

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