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A Trnsys Simulation of a Solar-Driven Air Refrigerating System for a Low-Temperature Room of an Agro-Industry site in the Southern part of Italy

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[b]*Sun Energy Europe S.r.l., Academic Spin-Off, Via B. De Capua 26, 81043 Capua (CE), Italy**Abstract*

Solar cooling technologies are of great interest because the cooling load in buildings is directly correlated to the intensity of solar radiation. In this paper, a feasibility study of a solar cooling system for a cold room in the Southern part of Italy is carried out. The cold room is part of an agro-industrial structure; the set point of temperature is 10°C. The room is 11.0 m long, 9.5 m wide and 3.5 m high. Two sliding doors 3.0 m high are present and strawberries are inside mainly. Thermal loads are evaluated by means of the software TRNSYS 17. Parabolic Though Collectors (PTCs) are employed to capture the solar energy. The solar field extends for 210 m^2. Two double-effect chillers (with a total installed power equal to 150 kW) for the collectors are employed. In addition, a conventional heat pump is installed to meet the demand for refrigeration during unfavorable weather conditions or during the maintenance of chillers. There are two different storage tanks: a tank with volume 10 m^3 for hot water storage and a tank with volume 40 m^3 for cold storage. The terminal part is constituted by dry coolers. Results are given in terms of heat transfer rates and solar fraction. Besides, an economic analysis is carried out to evaluate the payback time.

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Nomenclature

Q	thermal load [kW]
c_1	Collector's coefficient [W/m^2 K]
c_2	Collector's coefficient [W/m^2 K]
f	Solar fraction
NPV	Net present value
SCS	Solar cooling system

1. Introduction

Solar energy for the residential and industrial sector remains the best technology in air conditioning and refrigeration; it is an environmentally-friendly solution since the system uses thermal energy to produce cold which allows the use of solar thermal energy, waste heat and other sources of low enthalpy heat. Recently absorption cycles are available commercially in large amount, also double and triple effect cycles. Shirazi et al. [1] accomplished a feasibility study of a solar cooling system with a single effect, double effect and triple effect absorption chillers for heating and cooling applications. Their results demonstrated that by using concentrating collector with multi-effect chillers over solar single effect chillers in climates with low direct normal irradiance level there are not advantages. PTC collectors and solar multiple effect chiller are used in climates with direct normal irradiance fractions above 60% because it is necessary that the solar field is small. Al-Alili et al. [2] evaluated the performance in a hot climate as Abu Dhabi of a solar cooling system with single effect absorption chiller. They demonstrated that the electrical energy consumed is 47% less than the conventional vapor compression cycles, besides their results show that the collector area has a predominant impact on the payback time in terms of economic performance. Mammoli et al. [3], focused on the energetic performance of the solar single effect absorption chiller following different control strategies throughout the year. Their results showed that if the hot water storage tank is well insulated, the solar fraction can be boosted by 60%. Eicker et al. [4-5] analyzed a solar cooling system in different climates, they focused on economic and energetic performance, showing that it is possible to reduce the primary energy of 40-70% and the system design and energetic load has important roles in the performance of the system. Cucumo et al. [6] have carried out a thermodynamic analysis and evaluation of the performance of solar plants with parabolic though collectors cooled by atmospheric air. The plants were studied in two operating modes: at a variable flow rate and constant temperature at the outlet collectors, and at constant flow rate and variable outlet temperature. The results obtained demonstrate a very good performance by this type of plant, which utilizes the ambient air in place of the expensive and more problematic fluids such as synthetic oils and molten salts used in already constructed plants; it is very simple from the constructional point of view and does not need any water because the working fluid in the engine is the air and the intercooling of the compressor can also be done by atmospheric air. Regue et al. [7], in their study have tried to minimize losses in order to increase the performance of a solar thermal concentrator. They have carried out an experimental analysis, which consists of converting solar radiation into thermal energy using a cylindrical parabolic solar concentrator. The developed theoretical model involves a number of parameters such as the average monthly solar radiation that allows us to estimate the direct radiation at the reflector, the geometrical concentration and exchange of heat between the opening of the collector and the receiver, allows the evaluation of the temperature at the latter. This model of concentration leads to levels of temperatures between 70 C° to 200 C°. Cascetta et al. [8] have carried out a feasibility study of a solar cooling thermally driven system configurations for an office. They have compared three different configurations: 1) FPC (flat plate collectors) with a single effect absorption chiller, 2) ETC (evacuated tube collectors) with double effect absorption chiller and 3) PTC (Parabolic through collectors) with double effect absorption chiller. They have demonstrated that for mild climate the third configuration is optimal because in the summer time the solar cooling system accommodates totally the cooling energy demand. Besides they carried out an economic analysis considering the incentives and the NPV value, the payback time was 3.4 years. In the work of Drosou et al. [9], it was shown, the use of concentrating solar collectors leads to significantly higher output temperatures that can enable the use of two-stage absorption chillers with a higher COP. Alternatively, when low or medium temperature heat is required, the use of CST systems takes less space to cope with it than traditional flat plate collectors. The combination of these parameters can contribute to removing key barriers associated with the broader diffusion of solar cooling technology, enhancing the potential to become more competitive to the conventional air conditioning technologies.

In this work, a feasibility study of a solar cooling thermally driven system for a low-temperature room of an agro-industrial company in Mediterranean area is carried out. The work focused on meeting the energy needs of the building and on designing of an optimal configuration of the solar cooling system. The results of the simulation are in terms of thermal loads, solar fraction, energy consumption and economic performance. The planning of meeting the energy needs of the room with an optimum level of solar energy is studied.

2. Model description

In this work the optimal configuration of a solar cooling system is defined by using high-temperature solar thermal collectors (PTC) with multi-effect (LiBr – H_2O) absorption chillers. The SCS (solar cooling system) are designed to accommodate the cooling energy demand of low-temperature room of an agro-industrial company, in addition, the system is designed to operate throughout all the year. The plan of the room to be cooled is shown in Figure 1. It is possible to observe that the refrigerating room is located inside an industrial barn where there are other refrigerators, besides, there is a non-conditioned area adjacent to the room. The solar system to be dimensioned considering that it must cool the room maintaining a constant temperature equal to +10 °C.

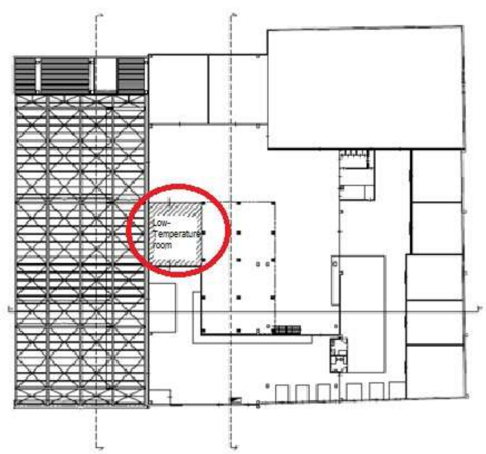

Figure 1. Location of the refrigerated room

2.1 Governing equations

Thermal loads are evaluated as follow [10]:

$$Q_{tot} = Q_T + Q_P + Q_I + Q_L + Q_{PP} + Q_F + Q_{TR} \quad (1)$$

Where Q_{tot} is the total thermal loads of the room, Q_T the transmission losses evaluated by means Trnsys 17 [11] considering the thermos-dynamical characteristics of the room, Q_P is the product load heat that must be removed to bring products to storage temperature and heat generated by the products (mainly fruits and vegetables) in storage. The quantity of heat to be removed can be calculated as follows:

$$Q_P = mc\left(T_{in} - T_{out}\right) \tag{2}$$

Where m is the mass of the product, c is the specific heat of product above freezing, T_{in} is the initial temperature of the product and T_{out} is the lower temperature of the product.

Q_I is the infiltration load, it can amount to more than half the total refrigeration load of distribution warehouses and similar applications. It takes into account infiltration by air exchange, infiltration by direct flow through doorways and by sensible and latent components.

$$Q_I = qD_tD_f(1-E) \tag{3}$$

Where $Q_{I,1}$ is the average heat gain for the 24h, q is the sensible and latent refrigeration load for fully established flow, D_t is the doorway open-time factor, D_f is the doorway flow factor and E is the effectiveness of doorway protective device (in this work E=0, there is not the protective device). The value of q is defined by the equation:

$$q = 0.221A(h_i - h_r)\rho_r\sqrt{1-\frac{\rho_i}{\rho_r}}\sqrt{gH}F_m \tag{4}$$

Where A is the doorway area, h_i is the enthalpy of infiltration air, h_r is the enthalpy of refrigerated air, ρ_i is the density of infiltration air, ρ_r is the density of refrigerated air, g is the gravitational constant, H is the doorway height and F_m the density factor evaluated as follow:

$$F_m = \left[\frac{2}{1+\sqrt[3]{\left(\frac{\rho_r}{\rho_i}\right)}}\right]^{1.5} \tag{5}$$

For cyclical, irregular and constant door usage, alone or in combination, the doorway open-time factor can ben calculated as:

$$D_t = \frac{P\theta_P + 60\theta_o}{3600\theta_d} \tag{6}$$

Where P is doorway passages, θ_p is the door open-close time, seconds for passage, θ_o are the minutes that the door simply stands open and θ_d is the daily time period.

Q_L is the light loads evaluated by the equation:

$$Q_L = L_L A_r \tag{7}$$

Where L_L is the lighting level, the quantity of light useful for square meters and A_r is area of the refrigerated room

Q_P is the heat gain by the occupancy defined as:

$$Q_{PP} = n(272-6T) \tag{8}$$

Where n is the number of people present in the room and T is the set point temperature (10°C).
Q_F is the heat gain by fans of coolers it is evaluated as:

$$Q_F = nP_F \tag{9}$$

Where n is the numbers of the fans and P_F is the power of fans (tab.6, [28])
Q_T is the heat gain by trucks calculated by:

$$Q_{TR} = nP_{TR} \tag{10}$$

Where n is the numbers of trucks and P_{TR} power of trucks.
To choose an optimal configuration of an SCS depends on a lot of factors as [26]:

- thermal loads;
- available area;
- budget;
- incentives.

3. Dimensioning SCS

In the present work, the main requisite is the coverage of at least more than 50% of the annual energy for cooling demand, so at least 60% of the annual demand for energy must be met by the SCS. Later, the quantity of energy saved using the SCS in terms of economic and energetic performances is evaluated. By increasing the area of the solar field, the solar fraction increases linearly tending to the unit value.

Table 1. Components of SCS.

Component	Size
Solar Thermal Collectors (PTC)	236 m^2
Hot storage tank	5 m^3
Cold storage tank	5 m^3
DHW tank	0.5 m^3
Chiller 1	70 kW
Chiller 2	70 kW
Heat pump	76 kW
Pumps, valves and control system	-

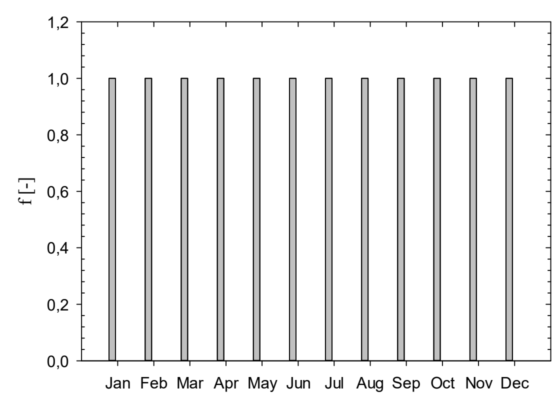

Figure 2. Solar fraction throughout the year

On the contrary, increasing the solar field's area the cost of the SCS increases, making not suitable the design of the system. The solar field and solar thermal collectors are the main components of the SCS, a hot storage tank for the hot water that supplies double effect absorption chiller, a supplementary heat pump to guarantee cooling air when the solar energy is not enough to heat water for the two double effect absorption chillers (the wet cooling tower is built in), a system of fan coils for cooling, valves, pumps and control system. Solar fraction and primary energy consumption are essential parameters for the evaluation of the energetic performance of the present SCS. Solar fraction is defined as the ratio between energy supplied by solar resources and the total energy demand required by the building. The energy carrier is electricity, the primary energy consumption corresponding to each configuration has calculated to provide a common expression of the energy carrier. The system is a considered a closed cycle system (the heat transfer fluid evolves in a closed cycle). The solar energy from the sun is captured by the solar field, which converts the solar radiation in thermal energy, transferring toward a refrigeration cycle. Successively, the cold fluid flow toward fan coils for the cooling. In the Mediterranean Area there are very hot summers, so SCS is the most suitable solution to reduce the primary energy and the CO_2 emissions. The hot storage tank stores heat produced by the solar field and then provide it, when necessary, to the refrigerating machine generator. A heat exchanger has built in the hot storage tank for the heat exchange between the incoming flow from the solar field and the water in the storage. The water, which is inside the hot storage tank, is pressurized (8 bar) because the temperature is maintained at a temperature of 180/200°C. The cold storage tank has the purpose of introducing a gap, in terms of time, between the production of cold by the absorption chiller and its use for air conditioning. This time difference can be several hours, but can also be some weeks or even months depending on the case that this is a daily or seasonal storage. Then the cold storage is like a thermal flywheel; its role is very important when during the maintenance time of the absorption machine, guarantees a cold reserve for many hours. In the present work, the accumulation is daily. When the thermal energy from the sun, which is captured by solar field, is not sufficient to meet demand during critical periods, it is necessary to active the auxiliary heat pump. The SCS is designed with the characteristics shown in table 2. The solar field surface is equal to 236 m^2 (28 modules) for the configuration described in this work; then, the solar fraction is evaluated. Besides, it is assumed that the solar field is acceptable to determine a coverage greater than 100% of the demand of the energy for cooling. A double-effect chiller lithium bromide – water (2x70 kW) compatible with PTC collectors are considered. This choice is motivated in the fact that typical double-effect chillers require heat transfer fluids at high inlet temperatures (around 140-180°C), and the parabolic collectors (PTC) can easily reach these temperatures.

4. Results

4.1 System performance

The configuration is analyzed throughout the entire year. Primary energy and solar fraction are the parameters to define the convenience of the system.

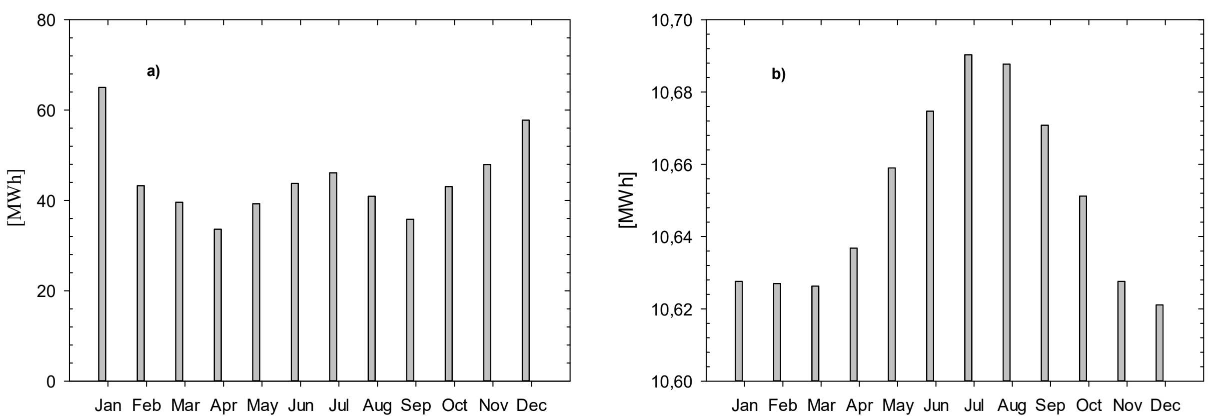

Figure 3. Results of the analysis: a) total radiation on the normal surface; b) total cooling load

The configuration with PTCs has all characteristics to reach this aim; in fact, they could offer their best just when the radiation is greater, that is during the summer. Solar fraction equal to 1 (fig. 2), as it happens in the present case, involves 100% of primary energy saved, compared to the conventional system which consists of a heat pump to electrically powered compression, active both in winter and summer. In Mediterranean Area the demand of electricity for cooling in the summer time is very high, then it is necessary to design a system capable of reducing the consumption of primary energy for cooling. In figure 3 the solar radiation on the normal surface of the PTC collectors (a) and the total cooling load throughout the year are shown. It is evident that the amount of the cooling energy is in the summer time and for this reason the solar field it is designed very large to meet the cooling demand totally.

4.2 Financial analysis

The feasibility study I carried out in order to evaluate the economic feasibility of the solar cooling plant. The economic analysis is carried out on the configuration taking into account both capital and operating costs of the system. The more expensive parts of the SCS are solar field and absorption cooling machines (88700 € and 110000 € respectively), other components have negligible effects. For the economic evaluation of the system, the Italian energy prices in fall 2016 were used. The remuneration for thermal energy was based on commercially valid prices. Overall the plant has an initial cost of 246700 € (table 2), which coincides with the initial investment; to assess the economic feasibility of the plant, it should be made an estimate of the annual cost savings, cash flows accumulated, the payback period. The following table shows some financial data, including the annual savings in euro, achieved thanks to the construction of the plant. Furthermore, it is essential to consider the annual maintenance and electricity consumption related to the electric pumps, the chiller and control devices.

Table 2. Initial investment

Component	Quantity	Price [€]
Solar Field	236 m^2	88700
Chillers	2	110000
Pumps	5	10000
Electric panel	1	5000
Hot storage tank	5 m^3	6000
Cold storage tank	5 m^3	8000
Accessories (valve, controls etc)	-	2000
Electric chiller (26.4 kW and COP=2.90)	1	12000
Piping	-	5000
	Total	**246700**

Table 3. Economic parameters

Investment	246700.00	€
Incentive	126907.00	€
Energy cost with Conventional cooler	92505.00	€/year
Energy cost with SCS	9975.00	€/year
Electric energy price	0.20	€/kWh
Annual money savings	82530.00	€/year
Annual maintenance	5000.00	€/year
Electricity inflation rate	3.50	%
Interest rate	5.00	%
General inflation rate	3.00	%

Therefore, this consumption represents an expense that must be addressed in any case every year, but that does not involve serious cash outflow. Many incentives are present in Italy for the development of energy from the sun and in general from renewable sources. Each incentive changes in function of the type of plant. The incentive is expressed in years; in the present work, the incentive time is five years. The annual savings in euro has been assessed as the difference between the cost of primary energy with and without the SCS. The difference is equal to 82530.00 € and it is the cost savings achieved at the end of the first year of investment, then by considering the electricity inflation rate, the actual cost savings is estimated, year by year. The payback time is evaluated as:

$$PBT = \frac{I}{NPV} \tag{11}$$

Where I is the initial investment and NPV is the net present value. Net present value and internal rate of return (IRR) has evaluated: NPV represents the profit generated by the investment, expressed in money; IRR is the return offered by the project, evaluated on the initial outlay on the basis of cash flows in future periods. Figure 4a shows that the payback period is 2.5 years and the NPV increases linearly with the years. In Fig.4b the comparison, in terms of energy consumption throughout the year, between the SCS and the conventional electric cooler is presented. The installation of the SCS achieves a reduction of primary energy of 89.2%.

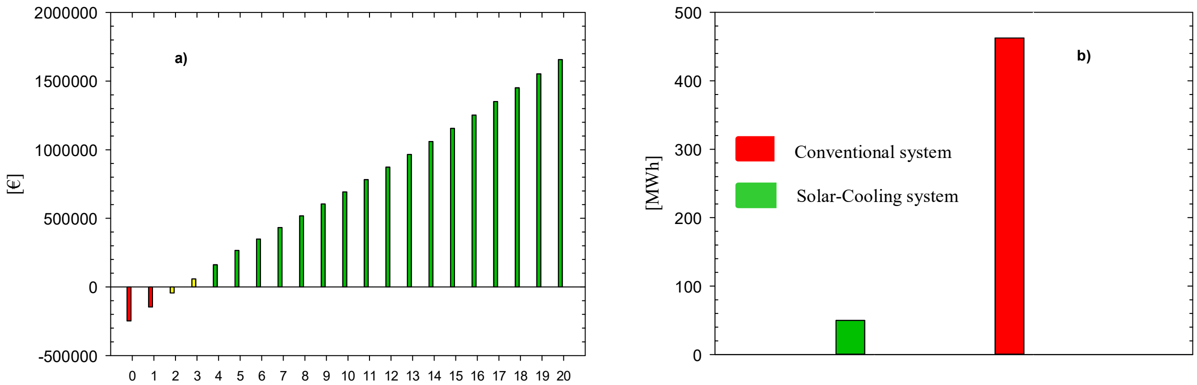

Figure 4. Economical analysis: a) Trend of the NPV as a function of years; b) Energy consumption comparison between SCS and convectional electric chiller

5. Conclusions

The present work focuses on the feasibility study of a solar cooling system for a cold room in an agro-industry site. The solar field occupies an area of 236 m^2 and two storage tanks are present: 5.00 m^3 for the hot water (for absorption cooling machine) and 5.00 m^3 for cold water (reserve of cold water). A supplementary electric cooler of 26.4 kW is present. One of the greatest advantages of an SCS, lies in the contemporary solar irradiation and refrigeration requirements; an another important element is that "solar cooling" uses an absorbing cycle, directly powered by solar energy, which covers up to 98% of the machine's power requirements. This cooling process allows a substantial saving of electricity compared to conventional systems. The energy demand is very low, achieving environmental benefits. In addition to energy saving, there are other advantages using this technology, such as reducing pollutant emissions (CO^2), besides it is silent and durable, because there are not without moving mechanical parts, chillers have no vibration or friction.

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