Heating and Cooling of a Building by Absorption Heat Pumps driven by Evacuated Tube Solar Collectors (ETCs)

Lazzarin, Renato[1], Noro, Marco[2]

[1] *Department of Management and Engineering, University of Padova – 36100, Vicenza (Italy), renato@gest.unipd.it*

[2] *Department of Management and Engineering, University of Padova – 36100, Vicenza (Italy), marco.noro@unipd.it*

Abstract

Solar assisted heat pumps till now have used the solar collectors as a cold source. Solar collectors provided when possible direct heating, otherwise they offered temperature levels higher than outside air for the heat pump evaporator. At the same time, solar thermal cooling exploits solar collectors and the absorption chiller only in hot months. The paper considers the possibility of employing ETCs to drive an absorption heat pump that is in summertime the absorption chiller. The cold source is the ground which is recharged by the solar collectors in mid seasons and by the cooling of absorber-condenser in summer. The study analyses the system behavior in yearly operation evaluating also the role of suitable storage capabilities in a temperate climate.

1. Introduction

An interesting option to increase attractiveness of solar thermally driven systems with respect to the actual more competitive PV heating and cooling systems can be the utilization of Evacuated Tube Collector(s) (ETC) as driving energy at the generator of a thermally driven heat pump. This can be nowadays possible thanks to a series of improvements regarding the solar thermal collectors occurred in the last years [1], thus a first order thermal losses coefficient of 1 W m^{-2} K^{-1} or even less is normal in ETC largely available on market. As a matter of fact, thanks to the high thermal efficiency also during colder months, suitable outlet temperature can be reached if a sufficient solar radiation impinges the collectors. Obviously, the performance depends on the climate of the resort considered, in particular on the outside air temperature and the clearness index. For such an annual utilization of solar thermal energy, a thermally driven chiller that can operate as heat pump with the suitable temperatures at the three heat exchangers (generator, evaporator, absorber/condenser) has to be coupled.

In this study, a simulation model of a thermally driven dual-source heat pump/chiller that faces the heating and cooling loads of an existing building is developed. The main novelty of this study is the utilization of thermal energy produced by ETC as both driving and heat source energy of a thermally driven heat pump/chiller.

2. Background

2.1. The simulation model

The case study concerns an existing real building, the F92 sited at ENEA Casaccia Research Centre near Rome (Italy) that houses the "Scuola delle energie" ("Energies School"). It is developed in three-storey (basement, ground floor and first floor, the lasts being identical) for a volume of 620 m^3 and a total floor surface of 230 m^2. The building/plant system has been modelled in Trnsys on the basis of transparent and opaque surfaces characteristics, internal gains and thermal/cooling plant schedules data supplied by ENEA [2]. The heating and cooling loads are calculated on the basis of climatic data of the Italian resort Belluno (46.14° N latitude) [3].

The original feature of the HVAC plant is the solar section composed of ETCs that drive a thermally driven absorption heat pump which satisfies all the loads. Moreover, the ETCs provide direct heating when possible, or act as heat pump cold source. The plant is set up by four main loops, and the various components are connected via suitable storage tanks (Figure 1). The control logic is based on a survey of solar radiation intensity (S), determining an operation threshold of 50 W m^{-2}, and on a comparison between the various suitable tanks set-points and the available temperatures in the different circuits (see next section 2.2).

The ETC considered is a modern collector available on the market, with very low first order heat loss coefficient (η_0=71.8%, a_1=1.051 W m^{-2} K^{-1}, a_2=0.004 W m^{-2} K^{-2}). It is modelled by type 71, with a tilt of 30° and an azimuth of 0°. The ground field is modelled by type 557a with a ground thermal conductivity of 2.87 W m^{-1} K^{-1}, and a storage heat capacity of 2016 kJ m^{-3} K^{-1}. The ground field is composed by nx50 or nx100 m in a row vertical tube U heat exchangers (n varying with the ETC field area), with an outer diameter of 32 mm and a thickness of 2.9

mm, distance 6 m. The tanks are modelled by type 4a (Generator Tank, Cold Tank) and type 60d (Heat Source Tank, Hot Tank). Finally, as no specific type is available in Trnsys for modelling a thermally driven heat pump, type 927 is used to model the water-water heat pump based on nominal data from a manufacturer concerning thermal power, cooling power and thermal power consumption at various heat source, heat sink and generator temperatures, considered varying in useful ranges according to source and sink temperatures (respectively $2 \leq T_{source\_tank} \leq 20$ °C and Hot Tank or Heat Source Tank max 40 °C), and generator temperature (75-95 °C).

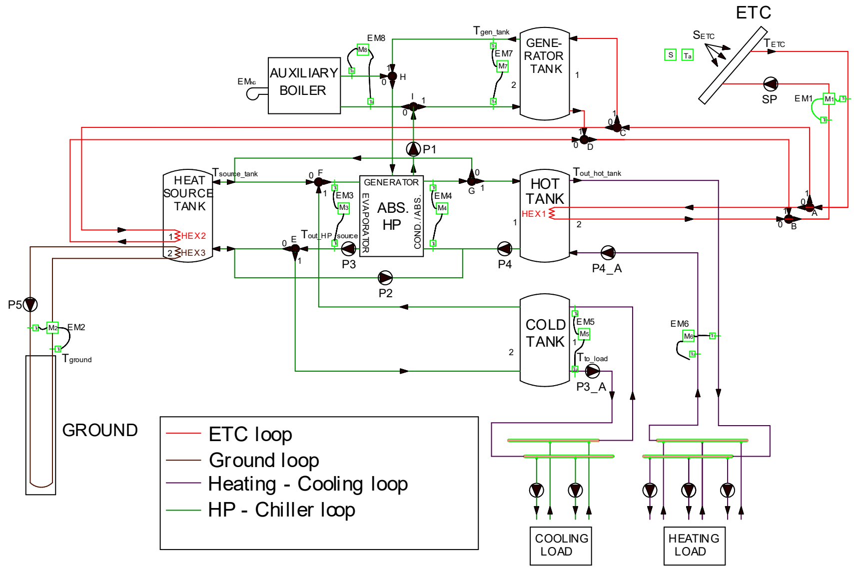


Figure 1 - Schematic of the main equipment of the plant

2.2. Control strategy of the plant

During heating season, the operation of the ETC field is based on the useful thermal energy producible by the plant in correspondence of two values of the reduced temperature T_{red}, defined as a function of the mean temperature of the water entering and leaving the collector (T_m), the air temperature (T_a) and the global solar radiation on the tilted surface (G_β) (Equation (1)):

$$T_{red} = \frac{T_m - T_a}{G_\beta} \tag{1}$$

T_{red1} is calculated as a function of the return temperature from the Generator Tank, whereas T_{red2} in function of the return temperature from the Hot Tank (with $T_{red1} > T_{red2}$). In correspondence of these two values, the thermal efficiency as defined in Equation (2) is calculated (EN ISO 9806:2017):

$$\eta_{th,ETC} = \eta_0 - a_1 T_{red} - a_2 \left(G_\beta\right) \cdot T_{red}^2 \tag{2}$$

where η_0 is the optical efficiency, a_1 is the heat loss coefficient, a_2 is the temperature dependence of the heat loss coefficient. The useful thermal energy producible by the plant in the two cases has been then determined (where COP_{HP} is the Coefficient Of Performance of the heat pump, HP):

1. $Q_{us,1} = \eta_{th,ETC,1} \cdot G_\beta \cdot COP_{HP}$ (indirect heating by HP)
2. $Q_{us,2} = \eta_{th,ETC,2} \cdot G_\beta$ (direct heating by ETC)

The ETC field is operated on the basis of the greatest Q_{us}: if $Q_{us,1} > Q_{us,2}$ the ETC field feeds the Generator Tank (A=B=C=D=1 - indirect heating by HP), whereas if $Q_{us,1} < Q_{us,2}$ the ETC field feeds the Hot Tank (A=B=0 - direct heating by ETC). If $T_{red} < T_{red2}$ the ETC field feeds the Heat Source Tank (A=B=1, C=D=0) (Figure 1).

During cooling season, A=B=1 in any case (no direct heating as no heating loads are present). The ETC operation is based on the Generator Tank outlet temperature: if $T_{gen\_tank} < 95$ °C, hot water from ETC is supplied to the Generator Tank to feed the absorption HP generator (C=D=1), if $T_{gen\_tank} > 95$ °C, hot water from ETC is supplied to the Heat Source Tank to regenerate the ground (C=D=0).

The operation strategy of the HP is based on the Hot Tank outlet temperature during heating season ($COOLING\_LOADS = 0$ AND $T_{out\_hot\_tank} < 40$ °C), and on the Cold Tank outlet temperature during cooling season ($COOLING\_LOADS > 0$ AND $T_{to\_load} > 12$ °C). In the first case, the machine operates as heat pump (the absorber/condenser heats up the Hot Tank, the evaporator is fed by the Heat Source Tank). In the second case, the equipment can operate as chiller (the evaporator cools down the Cold Tank) with heat recovery (if $T_{out\_hot\_tank} < 38$

°C the absorber/condenser is connected to the Hot Tank) or without heat recovery (if $T_{out\_hot\_tank}$ > 40 °C the absorber/condenser is connected to the Heat Source Tank). Moreover, when the HP/Chiller is in operation, its generator has to be fed by hot water that is produced by the Generator Tank if T_{gen_tank} > 75 °C (H=I=1) or by an Auxiliary Boiler (efficiency supposed constant and equal to 0.9) if T_{gen_tank} < 75 °C (H=I=0).

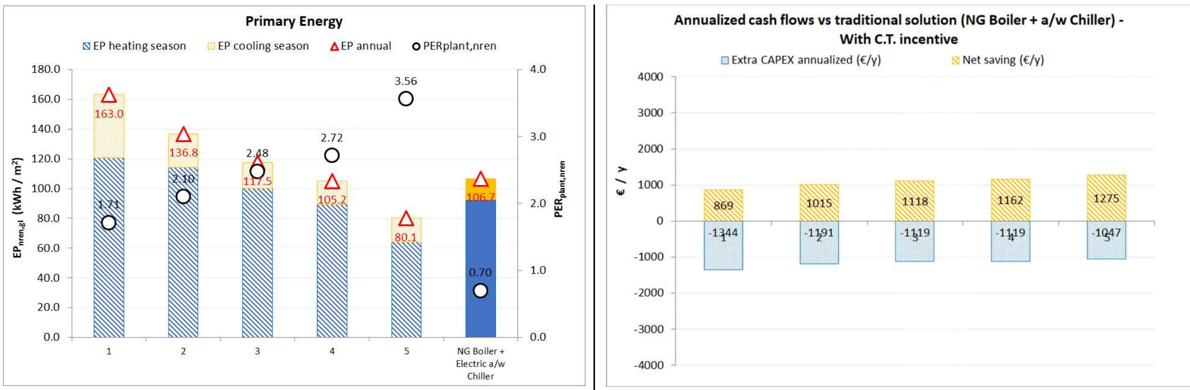
3. Results

As the Authors already proved ([4]-[6]), the length of the ground probes can be reduced when increasing the solar field as the contribution of the solar energy to recharge the ground during summer months is greater. This feature is here considered also in the following economic analysis. Furthermore, the capacity of the Hot Tank and the Heat Source Tank is fixed at 0.8 and 1.5 m^3 respectively, whereas the Cold Tank capacity is fixed at 0.75 m^3.

Many different combinations of ETC area, Generator Tank capacity and length of the ground probes have been simulated. For the sake of brevity, only the ones described in Table 1 are compared in Figure 2(a) on a yearly basis in terms of no-renewable primary energy consumption EP_{annual} (divided into two contributes relative to heating and cooling seasons), each with the counterpart of a traditional solution (NG Boiler for heating + a/w Chiller for cooling). The yearly primary energy ratio ($PER_{plant,nren}$) is reported as well. It is calculated on the basis of the no-renewable primary energy consumption of the whole plant (Auxiliary Boiler and parasitic power of pumps with the primary energy factors as defined by Italian Decree DM 26/06/2015: $f_{P,nren}$ (NG) = 1.05; $f_{P,nren}$ (electricity from the grid) = 1.95).

Table 1 – ETC area, ground probes number and length, and Generator Tank capacity for the main alternatives simulated – CAPEX of the main equipment

	ETC (m^2)	Ground (n x m)	GenT (m^3)	CAPital EXpenditures (CAPEX)	
1	60	2 x 100	4	Variable Ground Bor. cost (€ m^{-1})	25
2	60	2 x 50	4	Fixed Ground Bor. cost (€)	10000
3	80	2 x 50	4	ETC cost (€ m^{-2})	300
4	80	2 x 50	5	Adsorption HP/Chiller (€ kW^{-1})	600
5	100	2 x 50	6	El. Trad. Sol. a/w Chiller (€ kW^{-1})	300



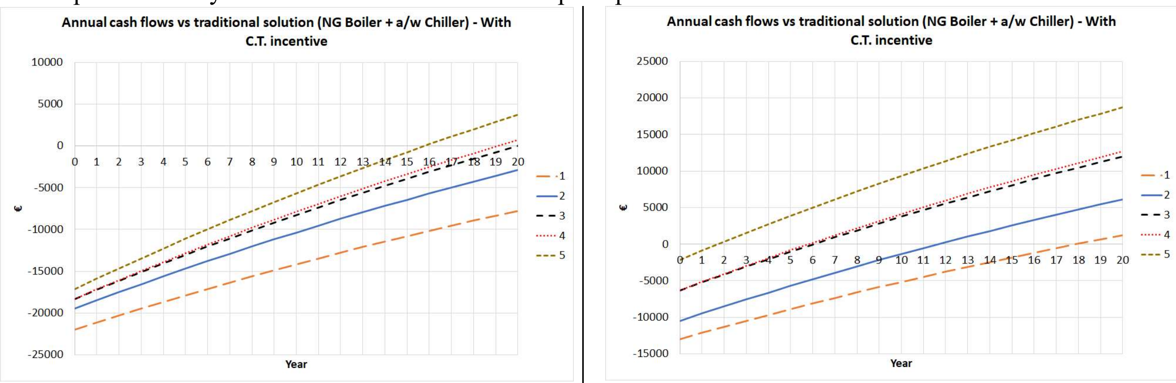
(a) (b)

Figure 2 - Comparison between the different alternatives compared with the conventional one, on the base of annual no-renewable primary energy consumption ($EP_{nren,gl}$) and primary energy ratio ($PER_{plant,nren}$) (a), and on the base of annual discounted saving and extra-CAPEX (b)

Comparing solutions 1 and 2 in Figure 2(a) reveals that a correct balance between the solar and the ground fields is necessary in order to improve energy performance of the plant. Moreover, increasing the ETC area is beneficial as EP_{annual} (red triangle) decreases: looking at solutions 1-3-5, the increase of the ETC area decreases the no-renewable primary energy consumption of the plant both in heating season (blue line bars) and in cooling season (orange line bars) as more thermal energy for driving the HP/Chiller is produced thus reducing the NG consumption of the Auxiliary Boiler. Increasing the capacity of Generator Tank has a similar effect, that is the plant EP_{annual} decreases (comparison between solutions 3 and 4). When EP_{annual} decreases, $PER_{plant,nren}$ increases. For the BL climate here considered, it is not advantageous to have lower than 60 m^2 of ETC field, as in this case EP_{annual} would be in any case greater than the traditional solution. The best solution appears to be the number 5

(100 m^2 ETC, 2x50 m ground probes, 6000 L Generator Tank), with an EP_{annual} of 80.1 kWh m^{-2} $year^{-1}$ and a $PER_{plant,nren}$ equal to 3.56.

In terms of economic analysis, besides the CAPital EXpenditures (CAPEX) reported in Table 1, an extra investment cost of 6000 € for the innovative solutions is considered (for Generator and Heat Source tanks, pumps, etc.) with respect to the traditional one. OPerating EXPenses (OPEX) are determined on the basis of a price of 0.9 € Sm^{-3} for NG, and 0.2 € kWh^{-1} for electricity from the grid. The results of the comparison of the five alternatives here considered with the "traditional" solution are reported in Figure 2(b) (interest rate 2%, time period 20 years). The figure reports both the net saving (calculated on the basis of the OPEX) and the extra-CAPEX annualized discounted cash flows: the economic viability is assured when the former are greater than the latter. As a matter of fact, the economic viability of all the five alternatives is allowed only considering the presence of the Italian economic incentive "Conto Energia Termico", that is determined on the basis of the standard performance of the considered ETC in the Wurzburg location (552 kWh m^{-2}), the ETC area, and the economic incentive by the Italian Decree DM 16/02/2016 (0.13 € kWh^{-1}). Figure 3(a) reports the annual discounted cash flows with respect to the traditional solution: even the best solution from the energy point of view (5) allows a discounted payback of 16 years. A turnaround of this conclusion cannot be excluded, if an important falling price of solar collectors takes place (Figure 3(b)). This would be not unexpected, as an investigation on market price of ETCs in China and India reveals prices already less than a half of current European price lists.



(a) (b)

Figure 3 - Annual discounted cash flows of the different alternatives compared with the conventional one with 300 € m^{-2} (as Table 1) (a) and 150 € m^{-2} (b) ETC investment cost

4. Conclusions

The use of modern high efficiency ETC offers a technical solution really suitable in view of solar heating and cooling systems with high energy performance. This study allows to understand the reasoning of the optimum sizing strategies and annual utilization of the equipment in dual-source (ground+ETC) thermally driven heat pump/chiller. The design of the plant by means of dynamic simulation considers different alternatives by varying the solar ETC field, the ground field, and the Generator Tank capacity, revealing that the most efficient solution features 25% saving of no-renewable primary energy consumption compared to a traditional one. A future decrease in ETC cost could make this kind of solar heating and cooling plant competitive with the actual more advantageous PV driven electric HP/chiller solution.

5. List of References

[1] Lazzarin, R., Noro, M. (2018): Past, present, future of solar cooling: technical and economical considerations. *Solar Energy*, vol. 172, pp. 2-13.

[2] Noro, M., Lazzarin, R., Busato, F. (2014): Solar cooling and heating plants: an energy and economic analysis of liquid sensible vs phase change material (PCM) heat storage. *International Journal of Refrigeration*, vol. 39, pp. 104-116.

[3] EN ISO 15927-4:2005. "Hygrothermal performance of buildings - Calculation and presentation of climatic data - Part 4: Hourly data for assessing the annual energy use for heating and cooling".

[4] Busato, F., Lazzarin, R., Noro, M. (2015): Ground or solar source heat pump systems for space heating: Which is better? Energetic assessment based on a case history. *Energy and Buildings*, vol. 102, pp. 347-356.

[5] Lazzarin, R., Noro, M. (2018): Lessons learned from long term monitoring of a multisource heat pump system. *Energy and Buildings*, vol. 174, pp. 335-346.

[6] Lazzarin, R., Noro, M. (2020): Photovoltaic/Thermal (PV/T) / ground dual source heat pump: optimum energy and economic sizing based on performance analysis. *Energy and Buildings*, vol. 211, 109800.