Performance Analysis Of a Solar-Powered Air-Conditioning System Using Absorption Refrigeration Cycle And High Efficiency Cooling Technologies Installed In Colombia

Caterina Fella[1], Paolo Silva[2] and Cesar I. A. Rolán[3]

[1] Politecnico di Milano, Milan (Italy)

[2] Politecnico di Milano, Milan (Italy)

[3] Universidad Pontificia Bolivariana, Medellín (Colombia)

Abstract

In this work an analysis of the performance of solar absorption air-cooling system installed to a building office of the Bolivarian Pontifical University (UPB) of Medellín in Colombia is presented. The solar powered, single-effect Li-Br absorption cooling system using evacuated heat pipe collectors, has a nominal capacity of 11,5 kW and it is implemented with a 17,6 kW indirect evaporative cooler (iec), responsible to improve the energy efficiency and to renovate the air in the building, for a total cooling capacity of more than 29 kW (more than 8 refrigeration tons). A system of chilled water capillary mats is implemented for air conditioning, where the heat transfer takes place through a radiant ceiling, reducing recirculation airflow and increasing the temperature of chilled water supplied, allowing a reduction in energy consumption by airflow and an improvement in energy efficiency of the solar-powered absorption cooling system. The performance is evaluated for three main subsystems composing the air conditioning system the absorption chiller, the heat pipe evacuated tube collectors and the indirect evaporative cooler. A series of test-measurements (TMs) are carried out varying the value of one or more particular parameters that affects the performance. Some indicators, which describe the performance, are calculated and compared, so an evaluation of the different TMs is provided. In more is presented an economic comparison with a conventional scheme in order to assess the potential energy savings of the system, the simple payback period of the investment is calculated for different size and cooling demand for Colombian and Italian case.

Keywords: Solar Cooling system, absorption chiller, heat pipe evacuated tube collectors, indirect evaporative cooler, performance analysis, experimental test-measurements, simple payback period

1. Introduction

Buildings produce high CO2 emissions as they consume almost 40% of worldwide energy, and a remarkable percentage of this energy is used for achieving thermal comfort conditions, both in heating and cooling. Thermal comfort is one of the first priorities, as it represents around 65% of building energy consumption [1]. Solar energy is the most widely renewable energy all over the World [2]. Solar power is sufficient to cover the thermal comfort demands in medium and low latitude regions [1] and it is not a fortuity that a solar-powered absorption cooling system was projected in Medellín, whose latitude is 6° 15' 6 N. Among the various solar air-conditioning alternatives, the absorption cooling system appears to be one of the most promising methods[3], because of the fact that the pick concentration of solar energy coincides with the hours when the cooling is required, in more as no chlorofluorocarbons (CFCs) are used, the system is presented as environmentally friendly. The project presented is the 7th out of 8 projects belonging to the National Strategic Plan of Energy Management that includes 4 years last projects in order to implement energy efficiency of Colombia's electric and thermal energy production systems, reduce the emissions and the environment resources depletion. This project was born as implementation of air-conditioning system for a building office, called Building 24, of UPB by Li-Br solar absorption cooling system with evacuated tube collectors. The office selected hosts about 50-80 persons and has an area of 239 m^2, for improving the energy efficiency of the air conditioning system, an indirect-evaporative air cooler (iec) responsible in

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renovation of the air, was installed, resulting that the air entering is provided at lower temperature than the temperature outside. Employing both system the thermal capacity needed for the office building is satisfied. The peculiarity of this air-conditioning system consists on the use of chilled water capillary mats instead of conventional fun coils. The heat transfer takes place through a radiant ceiling, reducing recirculation airflow and increasing the temperature of chilled water supplied, allowing a reduction in energy consumption by airflow and an improvement in energy efficiency of the solar-powered absorption cooling system.

2. System description

Medellín is also known as the city of eternal spring, presenting full year mostly sunny weather, with diurnal temperature which oscillate between 22°C and 35°C and a relative humidity between 40% and 75%. For this reason, it suits the solar powered air conditioning system very well. The figure below shows a schematic picture of the system with its principal component by the control screen of the engineering software used in order to test, measure and control its data. The picture with typical values from operation of 5th TM during chilled water production is shown below. For a good understanding, a brief description of the most important sub-system that characterize this configuration of Li-Br solar absorption air conditioning system, in order to justify the technologic choices that have been taken and focus on the specifications needed for proceed to a performance analysis is presented.

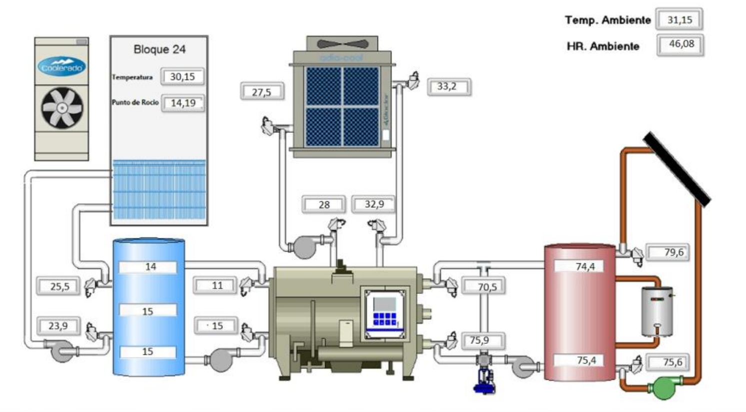

Fig. 1: LabView Control screen of the system

1. Evacuated heat pipe tube collectors (ETCs). The operating temperature of the hot water supplied to the generator of the LiBr-water absorption refrigeration system is about 70°C and 90°C. The lower temperature limit is imposed from the fact that hot water must be at a temperature sufficiently high to be effective for boiling the water off the solution in the generator[4]. For a number of 20 tubes for 10 collectors, the effective lighting area installed resulted of 20.4 m^2. The ETCs are connected directly to the hot water storage tank. Because no evaporation or condensation above the phase-change temperature is possible, the heat pipe offers inherent protection from freezing and overheating. The collectors were installed on the roof of the building chosen. Although it is recommended to incline the collector according to the latitude in order to catching the most direct radiation of the sun, it resulted that didn't permit the natural cleaning of the collectors by the rain and for this reason an inclination of 10° was chosen, and being installed in north hemisphere, they are directed to the south.

2. The absorption chiller constitute the core of the configuration: the refrigerant steam enter into condenser and is cooled down by cooling water flowing in the pipes, that passes from about 35°C to about 25°C, and the refrigerant water gathered in condenser enter into the evaporator by throttling. Because of low pressure (0,6 kPa) in the evaporator, obtained thanks to a vacuum pump, the refrigerant water will be evaporated by absorbing the heat of chilled water flowing in pipes, so that to lower the temperature of chilled water and realize the refrigeration; it is possible to obtain chilled water from 20°C until 7°C (in the picture the temperature in the evaporator are higher

because the screen shot is taken when chilled water production is still not started). In order to work, the temperature out of the tank could not be lower than 65°C and the cooling capacity increases with increase of this temperature.

3. Due to the intermittent nature of available solar energy, hot and chilled water storage tanks of a capacity of 1000 L are needed. They are carbon steel inertia buffer tanks, characterized of a high thermal inertia and insulation, allowing correct management of energy, ideals for the closed cooling circuit, because they act as the installation energy regulation[5]. Because of natural thermal stratification, the top zone of the tank is kept at the maximum temperature, and the bottom one at the lowest.

4. An auxiliary gas heater, of 40 kW of nominal power, directly connected to the tank, activated whenever the heat pipe collectors outlet temperature is less than the temperature required in the generator of the chiller, so that the temperature out of the chiller is not enough to assure the system functioning.

5. Since in an absorption-refrigeration cycle heat must be rejected from the absorber and the condenser, a cooling water system must be employed in the cycle. Because the absorber requires a lower temperature than the condenser, the cool water from the cooling tower is fist directed to the absorber and then to the condenser[4]. It has been chosen a particular closed circuit cooling tower because it is necessary that the water does not present problems of contamination and encrustation. The closed-circuit evaporative cooler is composed by wet panel type adiabatic cooler, in order to decrease the air temperature at the inlet of the coils, two coils of copper tubes and aluminum fins installed in V with an axial fan of variable volume installed on its top, so the air is cooled by ventilator and by adiabatic cooling, to assure high efficiency. The closed-circuit assure to minimize energy consumption and water consumption (the nominal power at maximum capacity is at 1,9 kW) and to avoid contamination problem.

6. Water capillary mats were chosen instead of conventional fun coils, because they result more efficient, exchanging heat by radiation. For this reason, they need a higher minimum temperature for being effective (about 7°C or even until 18°C water temperature, against the fun coil's temperature of 5°C), reducing recirculation airflow and increasing the temperature of chilled water supplied, allowing a reduction in energy consumption by airflow and an improvement in energy efficiency of the solar-powered absorption cooling system. This system is installed in a superimposed form on the "false ceiling", of the top of building 24, required in order to avoid to damage the ceiling in case of condensation of the water inside the tube.

7. Five pipe sections are identified in the system. The tabled below shows, at the sections considered, the diameter of the pipeline and the nominal parameters of the pumps selected (the head, the number of step-speed that could be selected and the power with respect to the working speed):

Tab. 1: Water Pipeline specifications

Section	Diameter internal	Head max	Speeds	Max power input
1 Hot water storage tank - Collectors	0.032 m	10 m	1	245 W
2 Hot water storage tank - Absorption chiller	0.032 m	10 m	1	245 W
3 Chilled water storage tank - Absorption chiller	0.032 m	9 m	3	150-179-196 W
4 Chilled water storage tank - capillary mats	0.063 m	8,5 m	Adaptable	9..144 W
5 Cooling tower - Absorption Chiller	0.04 m	27 m	1	740 W

8. In order to improve the energy efficiency, to renovate the air in the building, a 17,6 kW indirect evaporative cooler (iec), called Coolerado, was installed, therefore the air entering is at lower temperature than outside; the system draws a maximum of 750 watts of power at full flow[6]. Coolerado system is based on Maisotsenko cycle, a particular case of indirect adiabatic (evaporative) cooling air conditioning system. Humid working air, that constitute about half of the air that enters the heat and mass exchanger is released back into the atmosphere, carrying energy removed from the conditioned air. Chilled fresh air with no added humidity is supplied.

9. Auxiliary and measurement components.

In order to evaluate the performance of the solar cooling system a performance analysis for the three main subsystem that characterised the air conditioning in the Building 24 is carried out: the absorption chiller, the solar collector and the Coolerado.

3. Absorption chiller performance analysis

In order to test the performance of the equipment a series of test measurements (TMs) in different conditions are carried out.

In order to describe and compare the performance of the absorption chiller in different TMs done the performance indicators are estimated by an overall steady-state energy balance on the absorption chiller, neglecting net losses that may occur to the surroundings. Being possible to experimentally calculate the flow rate into the cooling tower ($\dot{m}_{c\ tower}$) from the manifacture's pump flow rate delivered/ power supply curve [12], the power removed by the cooling tower, from the the absorber to the condenser, was calculate as

$$\dot{Q}_{c\ tower} = \dot{m}_{c\ tower} * (T_{cooling,out} - T_{cooling}) \qquad \text{(eq.1)}$$

Where
$\dot{m}_{c\ tower} = 1{,}389$ kg/s
$T_{cooling}$ ($T_{c\ tower,out}$) is the temperature of condensation or "Cooling water inlet temperature" to the condenser, coming from the cooling tower.
$T_{cooling,out}$ ($T_{c\ tower,in}$) is the outlet temperature at the condenser, entering the cooling tower.Starting from the nominal performance curve of the equipment[11], furnished by the constructor, with respect to the chilled water outlet, cooling water inlet and hot water inlet temperature respectively, the equation curves of the cooling capacity were extrapolated. In particular, for each measure the cooling capacity ($\dot{Q}_{eva}$) with respect to the following parameter was calculated:

- Temperature of evaporation (T_{evap}) that is "Chilled water outlet temperature" at the evaporator, going to the chilled water tank, according to eq. 2

$$\dot{Q}_{eva} = 0{,}2698 * T_{evap} + 9{,}3811 \qquad \text{(eq 2.) [11]}$$

- Temperature of condensation ($T_{cooling}$) according to equation 3

$$\dot{Q}_{eva} = -\,0{,}889 * T_{cooling} + 37{,}681 \qquad \text{(eq.3) [11]}$$

- Temperature of generation (T_{hot}) that is "Hot water inlet temperature" to the generator, coming from the hot water tank), according to equation 4

$$\dot{Q}_{eva} = 0{,}3197 * T_{hot} - 17{,}482 \qquad \text{(eq.4) [11]}$$

Indeed for each TM three cooling capacities were found and the correction factor calculated as

$$f_1(T_{evap}) = \frac{\dot{Q}_{eva}(T_{evap})}{\dot{Q}_{nom,evap}} \qquad \text{(eq.5)}$$

$$f_2(T_{cooling}) = \frac{\dot{Q}_{eva}(T_{cooling})}{\dot{Q}_{nom,evap}} \qquad \text{(eq.6)}$$

$$f_3(T_{hot}) = \frac{\dot{Q}_{eva}(T_{hot})}{\dot{Q}_{nom,eva}} \qquad \text{(eq.7)}$$

Where

$\dot{Q}_{nom,eva}$ is the nominal capacity of the absorption chiller equal to 11,5 kW

Assuming valid the overlap of the effect the Coefficient of Performance was find for each TM ($COP_{i,TM}$) as

$$COP_{i,TM} = COP_{nom} * f_1(T_{evap}) * f_2(T_{cooling}) * f_3(T_{hot}) \quad \text{(eq. 8)}$$

Where:

COP_{nom} is the norminal Coefficient of Performance of the machine equal to 0,69.
f_1 is the correction factor of the cooling capacity due to the influence of the chilled water outlet temperature.
f_2 is the correction factor of the cooling capacity due to the influence of the cooling water inlet temperature.
f_3 is the correction factor of the cooling capacity due to the influence of the hot water inlet temperature.

The indicators of performance ($COP_{i,TM}$ and $\dot{Q}_{eva}$) were analyzed during a period in which the air-conditioning supplied by the transient operation of the chiller could be approximated to idealized steady-state operational conditions (the transient operation of the chiller leads to approximately 8% lower COP than would be expected if transients would neglected[7]). In order to compare the performance between each test measurement was chosen an equal period of 45 minutes.
Finally, knowing $COP_{i,TM}$ and $\dot{Q}_{i,c\ tower}$ for each TM, following parameters that characterize the system performance are calculated as:

$$\begin{cases} COP_{i,TM} = \dfrac{\dot{Q}_{eva}}{\dot{Q}_{gen}} & \text{(eq. 9)} \\ \dot{Q}_{c\ tower} = \dot{Q}_{eva} + \dot{Q}_{gen} & \text{(eq. 10)} \end{cases}$$

$$\begin{cases} COP_{i,TM} = \dfrac{\dot{Q}_{eva}}{\dot{Q}_{gen}} & \text{(eq. 11)} \\ \dot{Q}_{c\ tower} = \dot{Q}_{gen} * (COP_{i,TM} + 1) & \text{(eq. 12)} \end{cases}$$

$$\begin{cases} \dot{Q}_{gen} = \dfrac{\dot{Q}_{c\ tower}}{(COP_{i,TM} + 1)} & \text{(eq. 13)} \\ \dot{Q}_{eva} = \dot{Q}_{c\ tower} - \dot{Q}_{gen} & \text{(eq.14)} \end{cases}$$

Where

$\dot{Q}_{gen}$ is the heat consumption from the generator.

The results are shown in term of average in the table below:

Tab. 2: Performance results of the absorption cooling system during the five TMs.

	1st TM	2nd TM	3rd TM	4th TM	5thTM
Average during 45 minutes of air conditioning (stable period)					
$\overline{T_{hot}}$ [°C]	72,690	77,320	75,950	79,140	69,930
$\overline{\Delta T_{gen}}$ [°C]	3,484	3,28	3,166	3,380	4,560
$\overline{\dot{Q}_{gen}}$ [kW]	10,589	9,112	8,706	8,713	17,673
$\overline{T_{evap}}$ [°C]	15,330	13,670	16,150	14,140	13,560
$\overline{\Delta T_{evap}}$	2,648	2,308	2,697	2,652	4,030
$\overline{\dot{Q}_{eva}}$[kW]	**4,273**	**4,473**	**4,240**	**4,698**	**6,699**

$\overline{T_{cooling}}$ [°C]	29,530	29,520	29,460	29,430	27,600
$\overline{\Delta T_{cooling}}$ [°C]	2,56	2,340	2,230	2,310	4,198
$\overline{\dot{Q}_{c\,tower}}$[kW]	14,862	13,585	12,946	12,830	24,372
$\overline{COP}_{i,th}$	**0,404**	**0,491**	**0,487**	**0,539**	**0,379**
$\overline{P}_{\substack{air\,cond\\entire\,period}}$ [kW]	2,8	2,65	2,5	2,4	3,4

Where

$\overline{\Delta T_{gen}}$, $\overline{\Delta T_{cooling}}$, $\overline{\Delta T_{evap}}$ are the average temperature difference inlet and outlet from the generator, the evaporator and the condenser respectively

$\overline{P}_{\substack{air\,cond\\entire\,period}}$ is the average of total power required by the system.

During all the TMs was never reach the nominal cooling capacity of 11,5 kW, that because of an intrinsic limit of the system that never allows to work with a temperature in the generator (T_{hot}) equal to 90°C . During the start up period, indeed the period in which no vapor would have been involved, the generator consumes a lot of power, as initial and final temperature shown in Table 3. This behavior is more remarked during the 5th TM when working with a cooling temperature lower, and less remarked when working at partial load, during 3rd and 4th TMs.

Tab. 3: Start up period of TMs.

	1st TM	2nd TM	3rd TM	4th TM	5thTM
Start up period [min]	8	7	7	5	5
Lost in temperature [°C]	5,8	7,5	6,5	6,5	8,2

Passing from 1st TM to 2nd TM could be noticed that increasing the inlet temperature of the generator of an average of 5°C (it was necessary to heat the hot water storage tank of 13°C mores, from 79°C to 92°C) the COP increase of 17,7 %. Also the cooling capacity increase of 4% and the time employed in chilled water production decrease of about 48 minutes.

The 3rd TM and 4th TM are characterized by a reduction in evaporator's flow rate, working with the second speed of the pump. It brings to an improvement in the effectiveness of the heat exchange that could be reflected also in an improvement of the COP.

If compare the indicators of the 3rd TM with which were obtained in the 1st TM, that are characterized by more similar generator temperature, the COP passed from to be the 58,6 % of the nominal in the 1st TM to the 70,6 % in the 3rd TM. During 3rd TM, for almost the same cooling capacity, the machine exploit much less thermal power than in the 1st TM (8,7 kW versus 10,6 kW), indeed a better COP is reached allowing the machine to work longer. The air conditioning last 25 minutes more respect to the 1st TM: the machine worked longer, because the minimum conditions required in the generator are guaranteed longer.

The best COP is registered during the 4th TM that is characterized to high temperature in the generator (the thermal storage temperature arrived to 98,4 °C) and better heat exchange effectiveness (the highest difference in chilled temperature, due to reduction of the mass flow rate in the evaporator). In this measurement, it was capable in reaching the 78% of the nominal COP and the cooling capacity improve of 5% with respect to the 2nd TM and the lowest time in production of chilled water was registered.

The 5th TM presented the highest cooling capacity, about equal to the 60% of nominal, but presented the lowest COP registered (about the 55% of the nominal one). That is because in order to provide about 7kW of cooling power ($\dot{Q}_{evap}$) consume about 17 kW of thermal power stored. Such situation provoke that the minimum condition requested from the chiller are extinguished very quickly.

4. Collectors performance analysis

The performance of the collectors are described by the efficiency in term of useful energy delivered in joules: a solar collector receives solar radiation $\dot{Q}_s$ from the sun [product of the surface area, $A_c(m^2)$ and the solar radiation perpendicular to the surface I_p (kW/m^2)] and supplies $\dot{Q}_g$ to a heat engine at the temperature T_H. The ratio of supply heat $\dot{Q}_g$ to the radiation $\dot{Q}_s$ is defined as the thermal efficiency of a solar thermal collector[8]:

$$\eta_{coll} = \frac{\dot{Q}_g}{I_p * A_c} = \frac{\dot{m}c_p(T_O - T_i)}{A_c * I_p} = \frac{\dot{Q}_g}{\dot{Q}_s} \qquad \text{(eq.15)}$$

In order to calculate the flow rate $\dot{\boldsymbol{m}}$ circulating into the collectors were assumed negligible energy loss into the hot water tank and the average flow rate came out from the energy balance was assumed as real constant flow rate delivered by the collectors' pump. The table below resume the collector's performance registered during 3rd TM, 4th TM and 5th TM (measurement for which was possible to measure the solar radiation I by means of the use of pyrometer). In the table below $\overline{\boldsymbol{\eta}_{coll}}$ is the average efficiency of the solar collectors during the whole period of operation; $\bar{\boldsymbol{I}}$ is the average solar radiation registered during the period of operation; $\boldsymbol{P_{peak}}$ is the maximum power output from the collectors registered; $\overline{\boldsymbol{P_{coll,inst}}}$ is the average power output from the collectors; $\boldsymbol{E_{tot}}$ is the total amout of energy provided by the collectors.

Tab. 4: Results of performance analysis of solar collector installed in the Building 24.

	3rd TM (27/03/2017)	4th TM (28/03/2017)	5th TM (25/05/2017)	Weighted Average
$\overline{\boldsymbol{T_{amb}}}\ [°\boldsymbol{C}]$	31,40	29,50	30,90	
$\bar{\boldsymbol{I}}\ [\boldsymbol{W/m^2}]$	746,50 (725,08*)	723,20 (668,17*)	655,70 (775,6*)	
Collectors ON	11:29 (11:53*)	10:06 (11:23*)	11:12	
Collectors OFF	16:00	15:13	16:00 (13:30*)	
$\overline{\boldsymbol{\eta_{coll}}}$	0,56 (0,57*)	0,50 (0,53*)	0,38 (0,50*)	0,53*
$\boldsymbol{P_{peak}}$	14,91	15,60	11,08	13,86*
$\overline{\boldsymbol{P_{coll,inst}}}[\boldsymbol{kW}]$	9,82 (9,88*)	9,34(9,79*)	6,14 (8,30*)	9,50*
$\boldsymbol{E_{tot}}[\boldsymbol{kWh/day}]$	44,34 (40,65*)	47,82 (37,48*)	29,5 (21,9*)	40,97

*The value is referred to the average calculated during the period in which operate also the absorption chiller

The values remarked by the asterisk in the table, resulted higher because when the chiller works and exploit thermal power stored, the collectors' contribution is much more remarkable. Because the generator of the chiller is consuming the power collected into the tank, the temperature entering the collector is lower; indeed the collectors are much more effective, being the difference between temperatures in and out of the collectors higher. This is the condition C of the figure below that point out the performance of the collectors by the temperature difference in and out of the collector in orange related to the radiation incident on the collectors in blue, during the 4th TM. In particular collector efficiency, referring also to the system condition, is affected by the auxiliary heater contribution, that heating the water inside the tank increase its temperature inlet to the collector, so to make inefficient the collector contribution in case the radiation is not enough or the temperature inlet is too high (see Condition D in the figure). This phenomenon is even worse in case the chiller does not work and the auxiliary heater is switched on in order to heat the storage faster. When the Chiller is starting up, a not stable state persists (see Condition B) that addresses an appropriate control behavior; the same happens when the hot water pump is switched on in order to make the temperature into the tank homogenous (see Condition A).

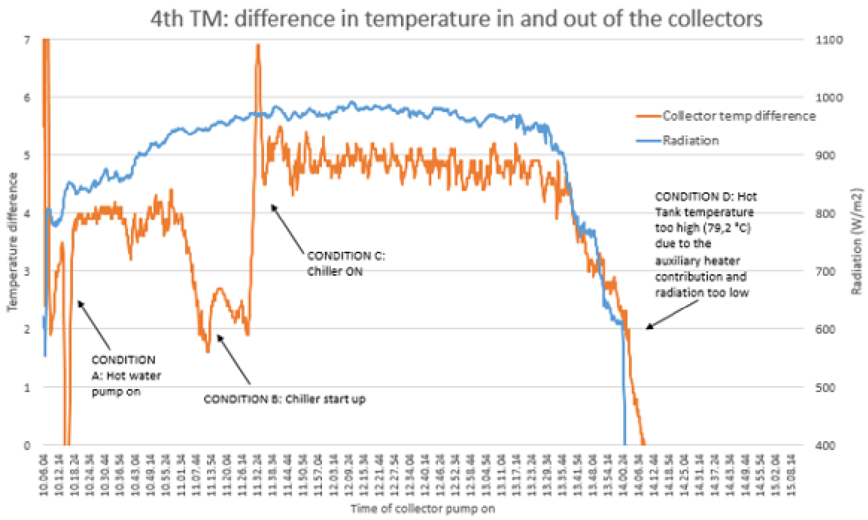

Fig. 2: Collector performance analysis during 4th TM

The 3rd TM registered the best value of global efficiency, instant power output and energy output because presented an higher average radiation and low generator temperature (see Table 4) indeed the condition D happen later. It is the longest measurement in which the collectors works simultaneously with the absorption chiller and the difference in temperature in and out from the collectors was higher (Condition C).

The 5th TM was the measurement in which the Condition C lasts less (only 2h 36min with respect to 3h 50min of the 4th TM and 4h 22 min of the 3rd TM) and indeed the global efficiency is lower. In more, the chiller during its operation required much more thermal power in order to produce a bigger cooling capacity: the collectors could only partially provide to it. This, in fact, suppose that the average temperature difference in and out of the collector, and so the global efficiency and the power output, during absorption chiller operation was lower than in the previous cases because the chiller faster consumes the power provided by them, together with the auxiliary heater.

5. Indirect Evaporative Cooler (iec) performance analysis

Being the IEC performance depending on temperature and relative humidity two different cases were simulated, assuming a constant volume air flow equal to $\dot{V}$= 0,76 m^3/s (maximum value registered in the measurement, assuming the ventilator of the iecworks always at the same speed). Were chosen two favorable weather condition typical of the city of Medellín: relative humidity (r.h.)equal to 30% and dry bulb temperature equal to 33°C. The product air temperature ($T_{product\ air\ flow}$) was calculated through equations furnished by the constructor: with respect to the external static pressure of Medellín the air product temperature should approach at 94% to the wet bulb temperature(T_{wb}). In equation is also considered an about of 2,5% of worsening in performance because of the height of Medellín of 1538 meters [13] :

$$(T_{amb} - T_{wb}) * 0{,}94 * (1 - 0{,}005 * 1538/305) = X \qquad \text{(eq.16)}$$

$$(T_{amb} - X) = \text{Design } T_{wb} \qquad \text{(eq.17)}$$

$$T_{product\ air\ flow} = \text{Design } T_{wb} + 1{,}1 \qquad \text{(eq.18)}$$

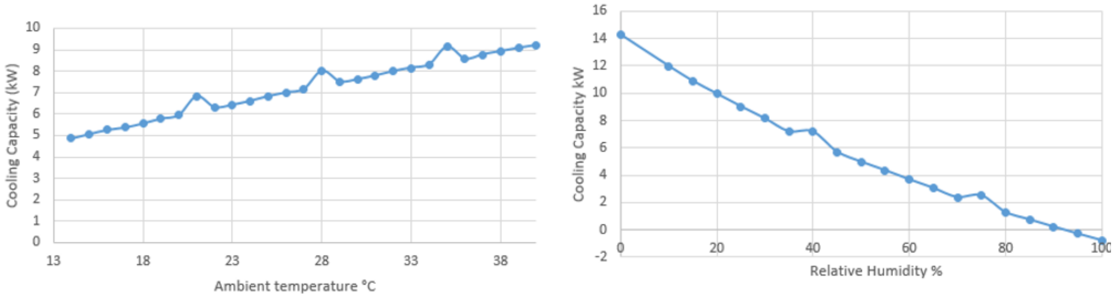

Fig. 3: Iec Performance at a defined relative humidity (r.h.=30%) and dry bulb temperature ($T_{amb} = 33\,°C$)

By these plots it could be seen that in these conditions it is never reach the maximum nominal capacity (17,6 kW): in the first simulation (at r.h.=30%) about 9,2 kW of cooling power is reach at the temperature equal to 40 °C and at temperature under 35°C the capacity is halved. At the ambient temperature of 21, 28 and 35 °C the capacity improve because the difference in wet and ambient temperature registered[9] is higher. Instead, in the second case (at T=33°C) the cooling capacity reaches a maximum of 14,3 kW but decrease really rapidly at relative humidity few lower (reaching negative values at high relative humidity, because the outlet temperature resulted even higher than inlet temperature).

Considering all the tests measurement done so far, it could be said that the air conditioning with the use of iec at Building 24 happens when the temperature oscillates between 33 °C to 23°C and relative humidity between 30% to 72% (without take in account the day in which rain, because in that case is not convenient to work with theiec). Indeed taking into account these ambient condition, and the maximum flow rate registered experimentally, the maximum cooling capacity of the iec is more than 8 kW (a cooling capacity of 8,2 kW was obtained at the best ambient conditions of the air cooling, T=33°C, r.h.= 30%) and an energy consumption of maximum 681,3 W (the maximum experimentally registered).

6. Economic analysis

In order to analyze the economic effort and justify the initial capital cost of the system a comparison with an air cooler Chiller, the most used conventional refrigeration system is done. An analysis of fixed and annual cost was done, taking in account only the purchased equipment costs (PEC) in the fixed capital investment (FCI). The simple payback period was calculated as [10]:

$$Simple\ Payback[years] = \frac{Cost}{Benefit} \qquad \text{(eq. 19)}$$

Where

the $Cost$ are represent by the fixed cost difference between solar and conventional [€]

the $Benefit$ are represented by the variable (annual) cost savings, in primary energy consumption and Operational and Maintenance (O&M) costs, represented by the variable cost difference between conventional and solar systems [€/year]. The analysis was carried on for different size of the system (indeed for different cooling capacity), from the actual case (about 30 kW_{fr}) up to 1300kW_{fr} , and assuming that the expenditure is justified over a long period of cooling demand, the analysis was also done for different operational hours ($t_{op\ hours,year}$) from 3000 h/year up to 1000 h/year. Two different cases were analyzed: Colombian and Italian case, differentiating only for the specific cost of Electrical Energy (EE). For both variable and fixed cost the following assumption were made:

- Specific costs decrease with the increase of the size of the equipment (exchange rate of June20th 2017)
- The sizing of the collectors installed was estimated accounting for cooling capacity and the thermal storage.
- For the equipment which was not possible to provide the specific costs the relationship between the size and the actual cost with a coefficient equal to 0,6 was used.

- The Balance of Plant (BOP) account for a 10% and a 30% of the PEC of the solar and conventional respectively.
- $\bar{P}_{\substack{air\ cond \\ period}} = 2{,}5\ kW$, whose 40% does not depend on the size of the cooling system.
- $COP_{\substack{air\ cooled \\ Chiller}} = 3 \div 4$ variable with respect to the size
- Renewable system works for the half time of its operational hours by thermal energy coming from the auxiliary heater
- $\eta_{auxiliary\ heater} = 0{,}92$
- The O&M costs equal to 4% of the equipment cost of the conventional cooling system and the middle for the solar one.
- $\dot{c}_{natural\ gas} = 0{,}3$ €/Smc (tax free)
- $\dot{c}_{EE,\ Colombia} = 0{,}16$ €/kWh for Colombian case, $\dot{c}_{EE,\ Italy} = 0{,}21$ €/kWh for Italian case (tax free).

The Figures below show the results of the analysis for the Colombian and Italian case.

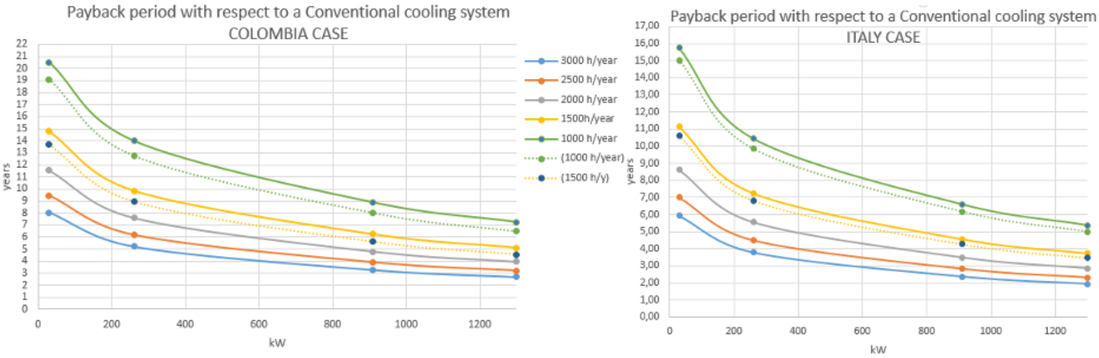

Fig. 4: Results of the payback period between Solar and Conventional cooling system

It is very clear the dependence in size of the equipment: the benefit, indeed the annual primary energy savings in term of money increase so that the ratio to the benefit over the cost decrease exponentially, increasing the size of the system. In Colombia, because of the very low cost of electric energy (0,16 €/kWh), this technology resulted economically convenient (payback period less than 6 years) from capacity few lower than 200 kW if the cooling demand is minimum of 3000 h/years. The situation seems much more convenient for Italy, where the price of electrical energy is 25% higher. The dotted lines represents the payback period for different size of the solar cooling system in case in which it operates 2/3 of the operational hours by sun radiation and 1/3 by thermal energy given by the auxiliary heater, indeed at different share in primary energy consumption. This assumption is done only when the operational hours are lower and indeed is enough to suppose that are present these conditions (1000 and 1500 h/year). For both cases a remarkable improvement could be seen passing from 1000 to 1500 h/year. That because the electric energy consumption has much more heavy contribution for high cooling demand, and the savings are more remarkable with respect to the cost of the investment: it might be not justifiable if the electric energy consumption is not so elevated. In more, should also be noticed that for the electrical energy consumption of the conventional system was assumed a COP that improves with the size; it is not always verified as true, even more if considered that over time the machine is affected by a drop in performance due to wear. Therefore, the payback period curve with respect to the size could present an even more inclined trend, that is traduced with the fact that even shorter period in return of the investment could be expected.

7. Conclusions

The absorption chiller evidenced a cooling capacity lowered of more than the 50 % of the nominal capacity, essentially due to impossibility of the system to maintain a high temperature in the generator. The machine, in fact cannot exploit the nominal thermal power required by the generator (that should be equal to 16,7 kW). Starting from the performance curve provided by the constructor and assuming a linear overlapping of the effects of the generator, evaporator and condenser temperature in the performance of the machine, a lower power furnished by the generator was registered, (values between 10,5 to 8,7 kW). The problem could be solved doing some modification of the layout:

• The pump that goes from the hot water storage tank to the absorption chiller takes water from the bottom of the tank; indeed, it affects negatively the generator operation due to the thermal stratification inside the tank. It is appropriate to connect the pump to the top of the hot water storage tank, in order to bring into the generator always the hottest water stored.

• The auxiliary heater input is located directly to the hot water tank. A future simulation could be to connect the auxiliary heater in series with the thermal storage. It could lead to provide always the nominal temperature required by the machine, and to avoid the stoppage of the machine, assuring the minimum temperature required in the absorption chiller.

• The pump that goes from the chilled water storage tank to the absorption chiller takes water from the bottom of the tank; indeed, it always sent the coolest water to the evaporator, affecting negatively the heat exchange effectiveness in the evaporator. Move the downstream handle of the chilled water pump to the top part of the tank will increase the difference between inlet and outlet temperature of the heat exchange of the evaporator, increasing the cooling capacity.

With the actual layout, two different operational conditions of the system could be preferred: the one, which could lead to highest COP registered so far, and another that, provides the highest cooling capacity, at a cost of higher energy consumption. These two different configuration are obtained by the lasts two TMs, the 4th and the 5th, proving indeed that during the experimental tests, the right directions in varying operational conditions has being followed. The operational conditions, which lead to optimize the COP, have been reached at 4th TM. Working with the second speeds of the chilled water pump to the evaporator and at the highest temperature that actually could be possible to maintain in the generator, heating the thermal storage up to the limit that avoid the boiling (about 98°C), the effectiveness of the heat exchange in the evaporator improves and the machine produce chilled water exploiting less thermal power in the generator, indeed obtaining the highest values of COP so far registered, equal to 0,54 that is only 20% less of the nominal value. This configuration assure the longest period of work of the chiller because consumes less instantaneous power from the generator. In more, it is assure the lowest value of energy consumption, because both the cooling tower and the pump required less power input. This configuration could be more suitable in case is needed only to maintain the same temperature in the building, and indeed when the cooling demand is lower.

The operational conditions, which lead to optimize the Cooling capacity, have been reach at 5th TM. As the performance curves of the absorption unit shows, lowering the condensation temperature up to the limit that avoid crystallization of the refrigerant flow, increase the cooling capacity of the machine. In particular, it was working with a Tset in the cooling tower equal to 28°C and at maximum capacity in the evaporator, indeed with the third speed of the chilled water pump. It was possible to reach the 60% of the nominal cooling capacity, once have heated the thermal storage up to the boiling limit. This configuration is very useful in case the cooling demand is imminent and big.

The average maximum collectors capacity registered during the TMs done, resulted equal to 13,86 kW, indeed the actual capacity of the collectors does not resulted enough in order to furnished the nominal thermal power required by the absorption chiller. This justify the use of the auxiliary heater during the whole test measurements. Is evidenced an inappropriate control behavior is registered during start up period of the Chiller. In order to improve the capacity of the collectors some improvement have been proposed.

• The actual arrangement of solar collectors sees a series of 5 manifolds in parallel with the same series. In this way the flow rate provided by the pump is separated into two flux. Neglecting the heat loss from the hot tank, the flow rate resulted by the energy balance into the hot tank, is equal to 972 L/h per manifold. Reducing the flow rate, that referring to the datasheet of similar collectors could be even three times lower, the ΔT of the collectors and indeed the power output could improve. Positioning the collectors in series of two, indeed five in parallel could increase them performance.

• The actual arrangement of the auxiliary heater provoke that in case the radiation is low and the temperature inside the storage tank already high (due to the auxiliary heater contribution) the collectors contribution is not effective. The problem could be solved arranging the auxiliary heater in series with the hot water storage tank, so that the thermal storage is completely depended by the solar energy contribution.

• The actual position of the collectors does not permit to exploit the solar radiation after 4 pm, because the collectors resulted covered by trees' shadows. Very recommended is to move the collector to a different point of the roof of the building.

In order to cover the nominal thermal power required by the absorption unit, dimensioning the capacity of the collectors assuming to receive always an instant power equal to 13, 86 kW, a total of 4,2 m^2 of absorption area of collectors should be added. This corresponds to 4 collectors more (each collectors has 10 tube and each tubes 0,102 m^2 of absorbed area).

From the iec performance analysis, responsible in renovate the air in the building, is evinced that the machine in Medellin condition, and at the actual state of cleaning of filters could furnish a cooling capacity about equal to 8 kW.

From the economic analysis, is evinced the very strong dependence of the simple payback period on the size of the solar cooling system. Increasing the size of the system the benefit, indeed the annual primary energy savings in term of money, increase, so that the ratio of the benefit over the cost decrease exponentially, also due to the lower specific cost of the machine. The payback period is strongly reduced passing from the solar cooling system of a capacity of 30kW to a system of 250 kW. For installed capacity higher than the system analyzed, the technology resulted very convenient not only for a typical Colombian cooling demand (3000 h /year) but also for demand halved, indeed also for an Italian case. Actually in Italy, where the electric energy cost is very high the payback period of the investment resulted in general, much reduced than in Colombia, where the electric energy costs 35% less, due to the wealth of rivers that allows low-cost hydroelectric power production. Considering a cooling demand that could be realistic for Italian weather, equal to 1500 h/year, the technologies proposed resulted convenient (payback period of 6 years) already for system with cooling capacity equal to 500 kW in Italy (the same capacity for Colombian cooling demand has a payback period about of 3years).

Should also be noticed that the economic analysis was carried out considering for the electrical energy consumption of the conventional system a COP that improves with the size; it is not always verified as true, even more considering that over time the machine is affected by a drop in performance due to wear. Therefore, even shorter period in return of the investment could be expected. Anyway, the very good news is that installing an air conditioning system that uses these technologies could make the user richer not only in term of money but also in term of sustainability, because the conditioned ambient in which he will live won't be responsible in remarkable CO2 emissions and won't use any type of refrigerant that impacts the ozone. The user will live not only on a cheap air-conditioned but also environmental-friendly room.

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