Heat rejection control strategy for stationary tests of discontinuous thermally driven chillers

Radko Brock[1], Roberto Fedrizzi[1], Patrizia Melograno[1], Massimo Fabricatore[2],Erich Lantschner[2]

[1]EURAC Research, ViaIeDruso, 1, I-39100 Bolzano, Phone (+39) 0471 / 055611, Fax (+39) 0471 / 055699, radkoalexander.brock@eurac.edu

[2]VELTA Italia s.r.l., Via della Miniera, 9, I-39018 Terlano, Italia

1. Introduction

Sorption machines for heating and cooling have the capability to be supplied with waste heat or heat out of renewable energy sources. Especially in areas with a high solar radiation in Europe, the use of thermally driven chillers have the potential to be an alternative to common compression chillers. In the summer period, when air conditioning in buildings is needed, sorption chillers can significantly reduce the high electrical energy consumption, reducing the load peaks on the electric grid. Therefore the development of this technology shows increasing interest.

On this basis, since 2010, Eurac and Velta Italia began to collaborate at the development of an innovative standardized system used to cover thermal loads for heating, cooling and domestic hot water preparation, through the exploitation of a large fraction of solar energy. The project objective is the development of a system consisting of solar collectors, hot water tanks, a thermally driven chiller and a compression heat pump in combination with a floor distribution system, used for heating in the winter as well as for cooling in summer. The target selected are residential buildings with surfaces between 150 and 250 m^2.

Within this project, stationary tests were carried out in laboratory on the employed sorption machine to verify manufacturer's performance values, to define and develop a control strategy for the heat rejection system and to validate simulation's results. The paper presented shows the preliminary results of the stationary tests carried out to define a control strategy for the heat rejection system to be used both during the tests and on-site operation.

2. Stationary Laboratory Tests

The stationary tests were carried out at EURAC laboratory [Sparber et al., 2007], a configurable energy system with an adsorption chiller in the middle, the heat production section (bottom right), the heat rejection part (to the top) and the cold distribution (to the left) as can be noticed in Fig. 1.

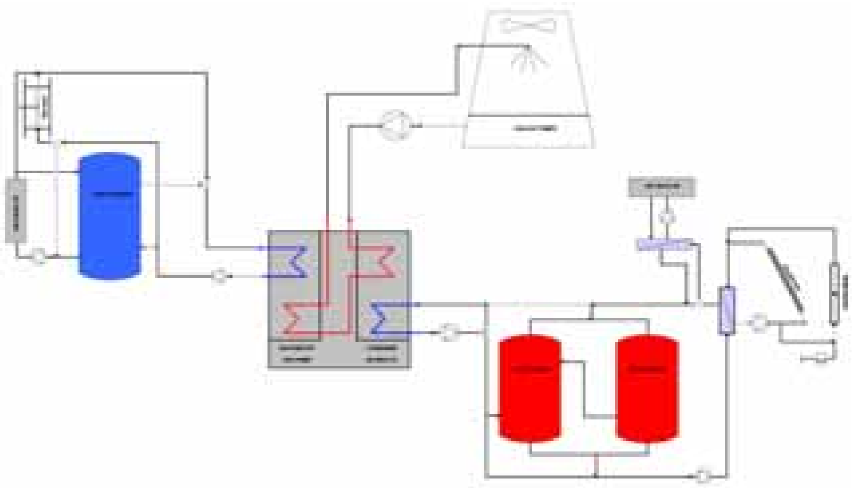

Fig. 1: Schematic overview of the test facility

The heat production is realized with a thermo-regulator, which provides a user-defined temperature at the output. It can be used singularly or in conjunction with solar collectors that are set up on the roof of the laboratory. These are connected to independent circuits and connected to the main circuit with a heat exchanger. In the main circuit two barrels are setup that can be arranged in series, parallel or completely bypassed. For the mentioned tests, the thermo-regulator was the unique heat source. The smallest tank (500 l) was also employed to ensure smooth temperature conditions at inlet of sorption chiller generator.

The heat rejection circuit is equipped with a GÜNTNER air cooler, working as a dry or hybrid cooler with a maximum cooling capacity of 50 kW. Three fans provide a maximum flow rate of 24600 m^3/h, with electric power consumption of 2,1 kW. The hybrid cooler is equipped with a sprinklers circuit that is activated to enhance the cooler capacity. The sprinkled water is preconditioned by a reverse osmosis system to ensure a clean surface of the cooler.

The cold water circuit is similar to the heat production one, with an independent thermo-regulator and a cold water storage tank. The water cooled by the chiller can be stored or directly delivered to a virtual user (the thermo-regulator). Again the cold water storage was used to provide stationary temperature conditions (as much as possible) to the chiller's evaporator.

During the tests, the adsorption chiller ACS08 from Sortech was analyzed. Tab. 1 reports manufacturer's data for the tested machine:

Tab. 1 - Chiller's performance as declared by the manufacturer

SORTECH ACS08		
Cooling capacity max.	*11*	kW
COP thermal max	*0.65*	-
Cooling capacity, nominal	*8*	kW
COP therm. nominal	*0.6*	-
Power consumption	*7*	W

The chiller is equipped with an internal control that allows two different working modes: POWER and ECO mode. In POWER mode the chiller is working with maximum power to reach the chilled water set point temperature. To be noticed is that, due to the cyclic behavior, the inlet water is cooled down to the average (over 1 cycle) of this temperature. If the set temperature is reached, the chiller automatically switches to ECO mode to reduce the electric consumption at the heat rejection. In ECO mode the chiller is controlled internally to reach the chilled water outlet average temperature by means of longer cycles compared to power mode operation. The reported stationary tests are carried out in POWER mode.

The following conditions are kept during the tests at generator, condenser and evaporator inlet. The mass flows (see Tab. 2) are identical to the manufacturer's test conditions. The test temperature conditions are shown in Tab. 3. Generator inlet temperatures are selected to allow direct comparison of the results with manufacturer's ones. Condenser inlet temperatures are selected for the tests to respect standards EN 14511 and EN 12309 (tests of compression and thermally driven heat pumps). The evaporator inlet temperature (18°C) was kept constant for all tests, if the chilling capacity was enough to maintain 15°C average chilled water temperature. If not, the inlet temperature was decreased to maintain 15°C average outlet water temperature.

Tab. 2: Mass flows at adsorption chiller ACS08

Circuit	Mass flow	Unit
Generator	*1.6*	m^3/h
Condenser	*3.7*	m^3/h
Evaporator	*2.0*	m^3/h

Tab. 3: Inlet temperature to adsorption chiller

Circuit	Inlet temperature	Unit
Generator	*75/85/95*	°C
Condenser	*27/30/35*	°C
Evaporator	*18*	°C

All test were carried out following the next steps:

1. The system with the heaters is started until the set temperatures in the storage tanks are reached
2. The sorption chiller is switched on
3. The adjustment of the three way valves is done to control the correct supply temperature to the sorption chiller circuits
4. The chiller is operated until the behavior of the temperature plot over the time does not change anymore: the last two cycles behave in the same way in terms of cycle time, powers delivered and average temperatures to the chiller.
5. ***The average inlet temperatures to the three circuits of the chiller have to remain in a range of ±0.5°C with respect to the set temperatures.***
6. Four half cycles (two cycles) are carried out, during which relevant values are logged in a .txt file, with a ***time period of 5 seconds***.
7. The averages of the three inlet temperatures (Generator, Condenser, Evaporator) are calculated over the two cycles. If the average is in the above mentioned range, this test is taken for further evaluation. If only one average is out of range, the test is repeated.

This stated, due to the discontinuous behavior of the adsorption chiller, stationary conditions cannot be reached at any time. Fig. 2 shows a common temperature plot of the chiller's inlet and outlet temperatures. Two cycles are plotted: each chamber operated twice in adsorption and desorption phase.

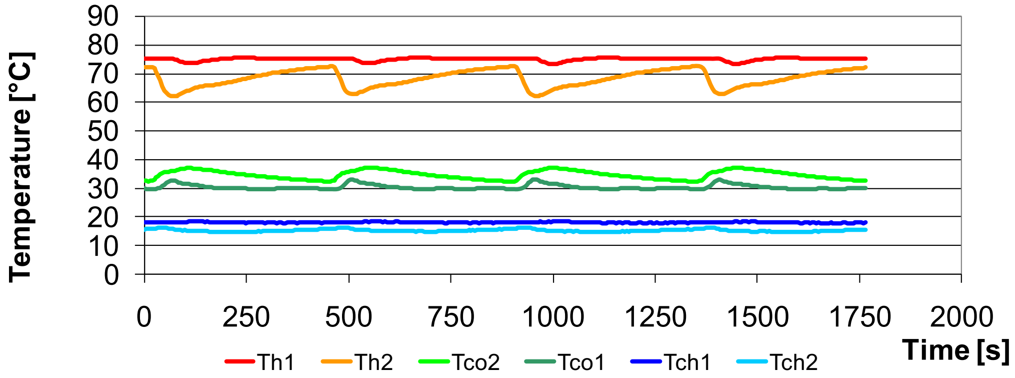

Fig. 2: Example of Sorption chiller temperature profile. Inlet average temperature 75°C at the generator, 30°C at the condenser and 18°C of the chilled water

As can be seen, time dependent temperature conditions at outlet of the machine propagate to the temperature values at chiller's inlet. The plot shows small variations (less than 0.5°C) with respect to the set point for the inlet chilled water T_{ch1} and relatively small variations (less than 2°C) for the heating fluid temperature T_{h1}; a basic control of the boundary temperature conditions (by means of three-way valves) was used in these cases due to the limited temperature ranges, easily complying with the tests requirements (point 5).

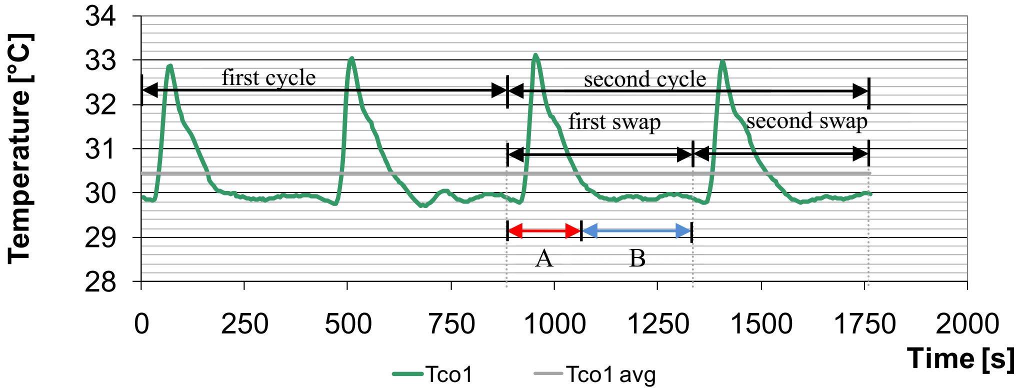

Fig. 3: Example of sorption chiller Tco1 temperature profile

Much larger deviations are encountered when heat rejection inlet temperature (T_{co1}) is investigated, as can be seen in Fig. 3. This is due, on the one hand, to the large thermal powers to be rejected after each swap (see temperature peak in sector A) and to the fact that a storage tank, which could flatten temperature peaks, is missing in the heat rejection circuit; on the other, the inlet temperature value is a direct consequence of the dry cooler management strategy. Again, this is to some extent a limitation of the laboratory infrastructure, since an accurate adjustment of the set conditions requires a complicated management of the heat rejection circuit to be set; however, tests under real working conditions allow 1) a direct assessment of the limits related to the operation of the sorption chiller with a specific heat rejection unit under real ambient temperature and humidity circumstances and 2) the development of control strategies dedicated to the specific machine both for tests and for on-site operation.

Regarding the latter point, the heat rejection fans are responsible for the largest electricity consumption of the sorption cooling system; an optimization of the fans' speed management, following the actual thermal loads to be rejected, enhances significantly the total electric COP of the system.

3. Control of the heat rejection system

To avoid large temperature fluctuations, the heat rejection control needs to operate as fast as possible after every swap, cooling down the condenser's water toward the set point and rapidly rising chilling capacity to maximum values allowed (with respect of the boundary temperature conditions). With this regard, different control strategies were experimented.

First of all a strategy was used to control the return temperature from the hybrid cooler by varying fans and pumps speeds in the heat rejection circuit. A parametric analysis was carried out to optimize the PID values for both fans and pumps, reducing as much as possible the deviations of T_{CO1} with respect to the set point. The extremely non-linear behavior of the condenser outlet temperature after each swap is responsible for results that were not completely satisfying, and many tests needed to be rejected because not compliant with the tests requirements; a more complex control strategy was established. Two parameters were considered for the definition of the improved control strategy:

- external ambient temperature during test (T_{amb})
- divergence of the condenser outlet temperature ($\Delta T_{CO2}/\Delta t$)

Based on these parameters, the actuators at the heat rejection circuit (pump speed, fans speed and sprinklers spraying time) are regulated: when, after every swap, the divergence of T_{CO2} increases above a given value (dependent on the specific system investigated) pumps and fans speeds are set to a Maximum Value and sprinklers are switched on. When the divergence changes to negative (i.e. the temperature peak is reached) a normal PID control of T_{CO1} is implemented.

Still the problem is pretty much dependent on the temperature difference the external-ambient temperature and the average temperature of the fluid inside the cooler. This difference is responsible, together with the fans and pumps speed, for the heat flux evacuated to the ambient through the cooler. Therefore when outside temperature is relatively low and actuators are operated at very high rates, risk of overcooling is found, resulting in unsteady behavior of T_{CO2} and unnecessary high electric consumption; on the contrary, if high external temperature is encountered and reduced performance of the actuators are set, undercooling might result in unnecessary long working cycles or even set conditions might not be reached.

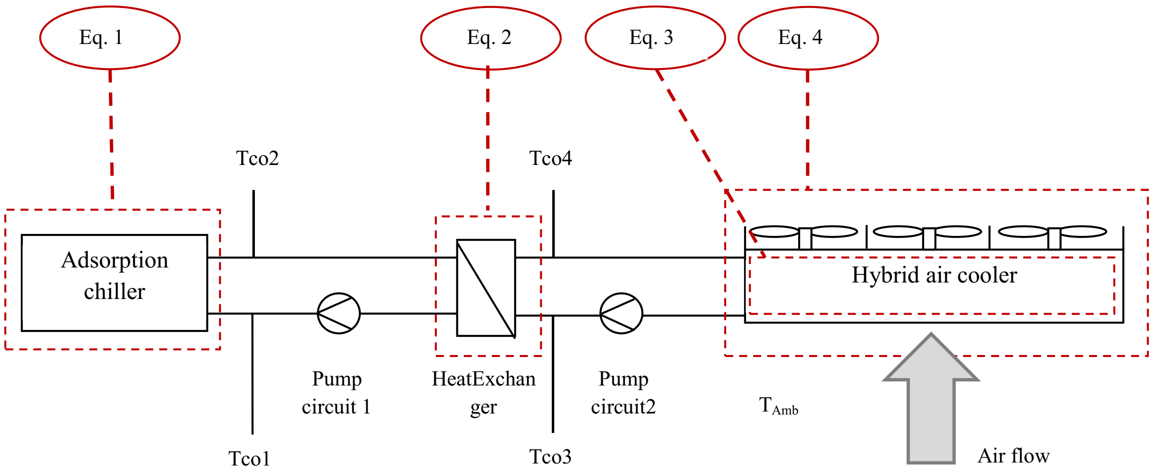

Fig. 4: Model of the heat rejection system

The calculation of the Maximum Value to be given to the component is done with a simplified time independent model of the heat rejection circuit (see Fig. 4), in which thermal losses are not taken into consideration. The minimum mass flows (water and air) are computed allowing given thermal power to be evacuated by the chiller for given ambient temperature. To elaborate this, stationary energy conservation equations are elaborated for each component of the circuit: adsorption chiller's condenser, heat exchanger, water side of the cooler, cooler's air side. The equations used are listed hereafter:

$$Q_{th-condenser} = \dot{m}_{co\ \ nom} \cdot c_P \cdot (T_{co2} - T_{co1}) \qquad \text{(eq. 1)}$$

$$Q_{th-condenser} = \varepsilon_{Hx} \cdot \dot{m}_{co2-nom.} \cdot c_P \cdot V_{Max} \cdot (T_{co2} - T_{co3}) \qquad \text{(eq. 2)}$$

$$Q_{th-condenser} = \cdot \dot{m}_{co2-nom} \cdot c_P \cdot V_{Max} \cdot (T_{co4} - T_{co3}) \qquad \text{(eq. 3)}$$

$$Q_{th-condenser} = \varepsilon_{HC} \cdot \dot{m}_{Air\ \ nom.} \cdot c_P \cdot V_{Max} \cdot (T_{co4} - T_{Amb}) \qquad \text{(eq. 4)}$$

The composition of the four equations allows the percentage Maximum Value to be determined. The results obtained with the analysis are reported in Fig. 5 to Fig. 7. They show the Maximum Value as a function of the external ambient temperature and the power to be rejected from the condenser, for given set temperature T_{CO1} at adsorption chiller inlet. As it can be seen, the maximum percentage of fans/pumps speeds reach 100% only for a limited number of combinations.

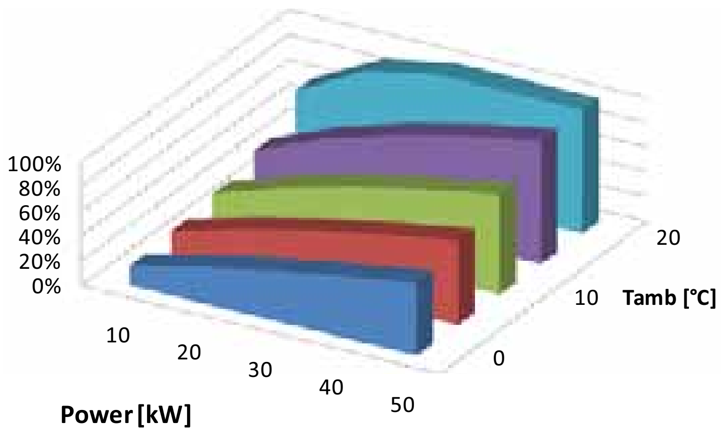

Fig. 5: Results of MaxValue for T_{col} set to 25°C

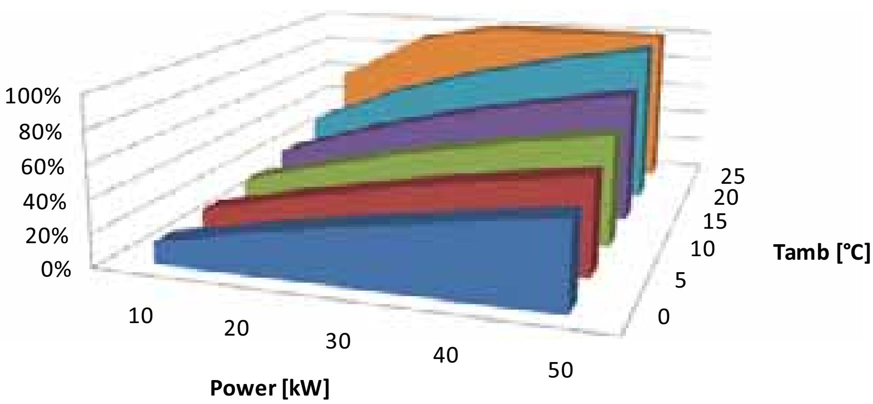

Fig. 6: Results of MaxValue for T_{col} set to 30°C

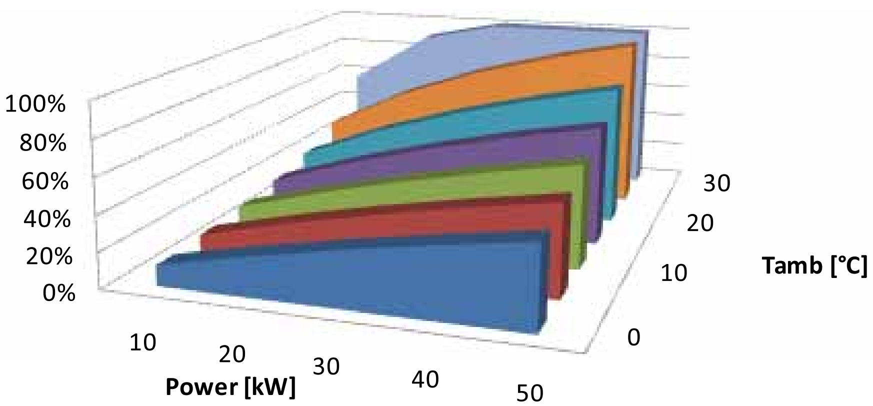

Fig. 7: Results of MaxValue for T_{col} set to 35°C

In order to present the actual implications of the discussed control strategy, a parametric experimental investigation was carried out by varying chiller's generator and condenser set temperatures, together with the percentage Maximum Value used to control the actuators.

4. Results

Chilling, generator and heat rejection thermal powers and fans' electric power consumption were calculated as the average over 2 working cycles of the data acquired every 5 seconds :

$$\bar{Q} \quad = \sum_{k=1}^{N} Q_{k-5_{sec}}/N \qquad \text{(eq. 5)}$$

Thermal COP was computed as the ratio of the evaporator and generator average powers over 2 working cycles:

$$COP_{th} = \frac{\bar{Q}_{EVAP}}{\bar{Q}_{GEN}} \qquad \text{(eq. 6)}$$

The data are presented in the followings as a function of the ΔΔT parameter, accounting for the two temperatures differences that are driving the sorption physics ($\mathrm{Th_1} - \mathrm{Tc_1}$ and $\mathrm{Tc_1} - \mathrm{Tch_1}$). The two differences are linked by the Dühring constant (equal to 0.03 for the investigated application).

$$\Delta\Delta T = (\mathrm{Th_1} - \mathrm{Tc_1}) - (\mathrm{Tc_1} - \mathrm{Tch_1}) * \mathrm{Düh} \qquad \text{(eq. 7)}$$

For each test condition mentioned in Tab. 3, tests were performed using the control strategy described in chapter 3 and varying percentage Maximum Value to the actuators between 65% and 100%. All tests reported were carried out in a narrow range of ambient temperatures (between 22 and 26°C) to avoid large effects on the fan loads due to varying temperature difference between warm fluid and air (see eq. 4).

The results of the tests are reported in Fig. 8 and Fig. 9 in terms of chilling capacity and thermal COP of the chiller.

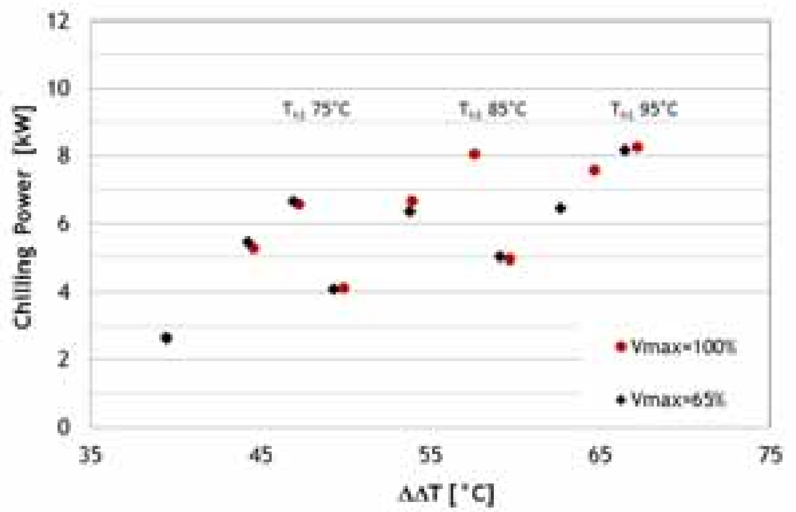

Fig. 8: Chilling power vs. DDT; percentage Maximum Value varies between 65 and 100%

As can be seen, the results obtained with the two different Maximum Values are in quite good agreement with each other, showing that the tests were properly carried out with both strategies. Chilling power ranges between around 3 kW and 8 kW depending on the boundary conditions; values for typical working conditions in solar applications vary between 6 and 7 kW. Thermal COP values range between 0.35 and 0.55, with typical values around 0.4 – 0.5.

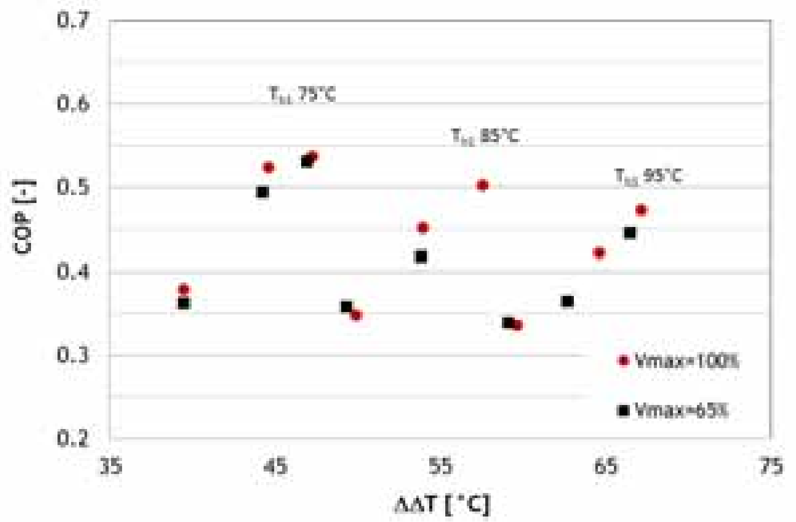

Fig. 9: Thermal COP vs. DDT; percentage Maximum Value varies between 65 and 100%

Fig. 10 reports the results of the tests in terms of electric energy consumed by the dry cooler fans during one operation cycle and for condenser inlet temperature (T_{CO1}) of 35°C versus generator temperature.

For given condenser and evaporator temperature levels, the energy consumption varies as a function of the generator temperature: the thermal power to be rejected increases accordingly to the generator temperature rise. The measured values range between around 16 Wh/cycle and 60 Wh/cycle. Those data cannot be considered as being representative for this kind of application due to the large temperature difference between condenser (T_{CO1} = 35°C) and ambient air ($T_{amb} \cong$ 24°C) temperatures: lower condenser temperatures could be easily obtained for mentioned ambient conditions.

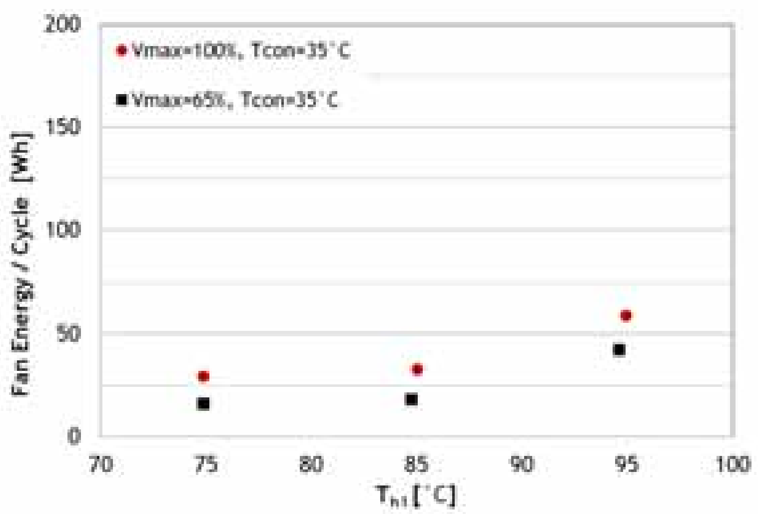

Fig. 10: Electric Energy consumed by the dry cooler fans during one operation cycle vs. generator temperature; percentage Maximum Value varies between 65 and 100%

What is clearly shown is the effect of the percentage Maximum Value on the fans electricity utilization: with regard to the stated test conditions, the operation with 100% Maximum Value produces an enhancement of about 15 Wh/cycle consumption compared to the 65% Maximum value cases, corresponding to a relative rise varying between 40% (at T_{h1} = 95°C) and 80% (at T_{h1} = 75°C).

In Fig. 11 the effect of the set condenser temperature on the electricity consumption is displayed. Energy consumption for inlet condenser temperature of 30°C go from 90 to 180 Wh/cycle. Comparing full (T_{CO1} = 35°C) with empty dots (T_{CO1} = 30°C) an increase between 100% and 200% is appreciated. In terms of electric COP ($COP_{el} = \overline{Q}_{EVAP}/\overline{P}_{FANS}$, accounting here for the only fans' contribution to the systems electricity consumption), values between 15 and 35 are assessed for inlet condenser temperature set to 35°C; a drop to values ranging between 8 and 25 is encountered with regard to the lower inlet condenser temperature (T_{CO1} = 30°C).

Fig. 11 shows also that, as far as set condenser and ambient temperatures approach each other, the benefit of assuming reduced fans' rates is progressively lost. Comparing empty dots with empty square at 75 and 85°C generator inlet temperature, it is shown that energy usage is almost the same for the two working loads. This effect is mainly due to the fact that the dry cooler operated at low fans' speeds is about to reach its operational limits in terms of heat rejection capacity. Moreover, as can be noticed by looking at the empty squares, the electricity consumption encounters a limitation at around 90 Wh/cycle, relative to continuous working conditions at set speed. At 95°C generator inlet temperature, the energy consumption is significantly lower than the corresponding value obtained with 100% Maximum Value; the set temperature could not be obtained though: the test was carried out at around 33°C at condenser inlet.

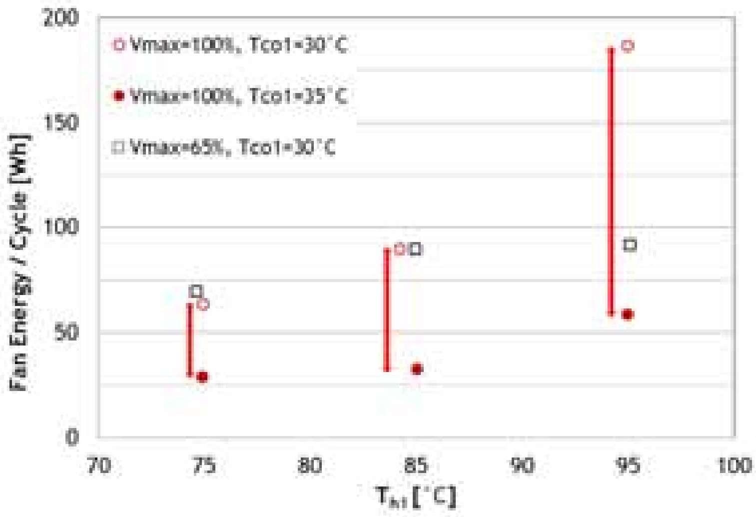

Fig. 11: Electric Energy consumed by the dry cooler fans during one operation cycle vs. generator temperature; percentage Maximum Value varies between 65 and 100% and condenser set temperature between 30 and 35°C

5. Conclusions

A control strategy was shown in this paper suitable for the management of discontinuous sorption machines, accounting for the large non-linearity of the temperature profiles at condenser circuit.

The effect of the control strategy in terms of fans' electric energy consumption was stated both as a function of the set inlet temperature at the machine condenser and of the fans' maximum speed (Maximum value was varied between 65% and 100%). A large increase of electricity consumption is encountered as far as the condenser inlet temperature reduces toward the ambient temperature. The benefit of assuming reduced fans' rates is also progressively lost as far as ambient and condenser inlet temperatures move close to each other.

On the one side, the reported remarks show how a dedicated/optimized control strategy is necessary to manage the heat rejection unit in a sorption cooling system: for on-site operation, the best compromise has to be assessed between reduced electricity consumption of the dry cooler and needed chilling power output by the sorption chiller.

On the other hand, indications were given to develop a whole control strategy for on-site operation of adsorption chillers: the management strategy of the dry cooler can be varied along the cycle to cope with the large condenser's temperature variations. The percentage Maximum Value of the actuators in the heat rejection circuit can be set as a function of the actual monitored thermal power to be rejected (namely generator and evaporator temperatures) and ambient temperature.

A final comment has to be made relating to the noise regulations. An upper limit to the percentage Maximum Value has to be foreseen in on-site application, having in mind that the maximum noise level allowed in residential areas is around 50 dBA in most of the European countries.

6. Acknowledgements

The authors would like to thank the Stiftung Südtiroler Sparkasse for the financial support.

7. References

François Boudéhenn, Mickaël Albaric, Nadège Chatagnon, Julien Heint, Nabil Benabdelmoumene, Philippe Papillon, 2010, Dynimical studies with a semi-virtual testing approach for characterization of small scale absorption. EUROSUN 2010, Graz.

Ghiaus C., Université de la Rochelle, 2000, Fuzzy model and control of a fan-coil. Energy and Buildings 33,

545-551

Kocaarslan I., E.Cam, March 2005. An adaptive control application in a large thermal combined power plant. Energy Conversion and Management 48, 174-183

Kühn A., Ciganda C.J.L., Ziegler F.,2008. Comparison Of Control Strategies Of Solar Absorption Chillers. Eurosun 350

Sparber W. Melograno P., Costa A., Rodriguez J., 2007, A. Test facility for solar-assisted heating and cooling systems. 2nd International Conference Solar Air-Conditioning, Tarragona

Sparber W., Melograno P., Franchini G., Santiago J. R., Palermo, Italy, 2009 B. Experimental Analysis of a Discontinuous Sorption Chiller Operating in Steady Conditions. 3nd Conference Solar Air Conditioning, Palermo.

SPX Cooling Technologies INC, Kansas USA 2008. Technical Report, Cooling Tower Energy and its Management

Marques C.A.X., Fontes C.H. , Embiruçu M., Kalid R.A., Escola Politécnica,2009. Efficiency control in a commercial counter flow wet cooling tower. Energy conversion and Management 50, 2843- 2855.

Melograno P. N., Santiago J. R., Franchini G., Sparber W, 2009. Experimental Analysis of a Discontinuous Sorption Chiller Operated in Steady Conditions. 3rd International Conference Solar Air-Conditioning, Palermo.

Ming-Tsun Ke, Kuo-Liang Weng, Che-Ming Chiang, June 2006. Performance evaluation of an innovative fan-coil unit: Low-temperature differential variable air volume FCU. Energy and Buildings 39, 702-708.

Núñez T. et al., September 2007. Development of an adsorption chiller and heat pump for domestic heating and air-conditioning application, Applied Thermal Engineering 27, Issue 13, pp 2205-2212.

Yu F.W. , Chan K.T., April 2005. Advanced control of heat rejection airflow for improving the coefficient of performance of air-cooled chillers. Applied Thermal Engineering 26, 97-110.

8. Nomenclature

Quantity	Symbol	Unit
Temperature generator inlet	Th1	°C
Temperature generator outlet	Th2	°C
Temperature condenser inlet	Tco1	°C
Temperature condenser outlet	Tco2	°C
Temperature heat exchanger inlet	Tco3	°C
Temperature evaporator inlet	Tch1	°C
Temperature evaporator outlet	Tch2	°C
Temperature ambient	Tamb	°C
Efficiency heat exchanger	ε_{hx}	-
Efficiency hybrid cooler	ε_{HC}	-
Specific heat water/glycol	C_p	kJ/(kgK)
Max value	V_{max}	%
Nominal mass flow 1st circuit	Mco-nom.	Kg/h
Evaporator power (Average on a 5 sec. period)	$Q_{EVAP-5sec}$	kW
Generator power (Average on a 5 sec. period)	$Q_{GEN-5sec}$	kW
Condenser power (Average on a 5 sec. period)	$Q_{REJ-5sec}$	kW
Thermal COP	COP_{th}	-
Dühring constant	Düh	-