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Development of Test procedures for Sorption Chillers installed in Solar Combi Plus Systems

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Abstract

Given the huge electricity consumption due to the summer air-conditioning - about 90 TWh of electrical energy are used in EU15 for satisfying this demand [1] - the introduction on the market of thermally driven sorption chillers, using solar as driving energy, could represent a valid solution to significantly reduce such consumption and the related CO_2 emissions.

Following this concept, in the last few years, the number of solar cooling plants has recorded a growth by 50% to 100% each year, in particular for small scale installations. This has induced several companies to invest in this technology, introducing new sorption chillers on the market. Nevertheless, an equal updating of the normative scenario has not followed the market development, leaving manufacturers without useful tools and references for marking their products before releasing them on the market. In fact, the tests prescribed by the standards for the evaluation of the sorption chiller performance are, in many cases, inapplicable: they treat all machines as continuous ones and classify them only on the base of the heating phases – i.e. single or double effect -, driving technology – i.e. direct or indirect fired - and sorbent physical properties – i.e. liquid: absorption; solid: adsorption-.

The need of having common references induced the author to assess a test methodology applicable to all chillers technologies. A new way to classify thermally driven chillers based on their working mode - i.e. continuous, semi-continuous and batch mode - has been identified and, based on that, a new test procedure has been developed. In the present document the newly developed test procedure is described, through the definition of rating conditions, test requirements and evaluation methods.

In particular, the validation of the test procedure made on a prototype of ClimateWell CW10 (an absorption chiller working in batch-mode) is presented. The chiller has been tested in the experimental laboratory at EURAC. The innovative test rig, developed within the present research program, is intended to allow the experimental investigation of small scale sorption chillers (up to 20 kW) and solar cooling systems. Thanks to its modular and flexible design concept, it allows

assessing the performance of the chiller investigated under a number of reality-like working conditions (varying climatic conditions, cooling and heating loads) and system's configurations (storage volume and type on the hot and cold side, kind of heat rejection system, control strategy, etc). The results achieved with regard to the tested chiller are compared with the results obtained by partner research institutes from tests carried out on other two sorption chillers. In this way, a validation of the test procedure with regard to the three main chillers' technologies has been carried out.

Acknowledgements

I really would like to thank everybody who accompanied me during the last three years, but if I mentioned all the names, the acknowledgments would be longer than the thesis itself. So, I strongly hope to include all of them in one page.

I would like to express my sincere gratitude to Prof. Giuseppe Franchini for having been my guide and having trusted me.

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My heartfelt gratitude is addressed to Mr. Dipl.-Ing. Wolfram Sparber for the chances he gives me to gain experience in the renewable energy field.

My affectionate thanks go to my past and present nice colleagues at EURAC, for the scientific collaboration but above all for the coffee and chocolate breaks, the parties and the funny moments.

Obviously, I would like to warmly thank my family, for being so close to me despite the distance.

Table of contents

Nomenclature

Below is a partial listing of symbols, Greek letters, subscripts and superscripts. Those that are used infrequently or in limited parts of the present work are defined locally and do not appear in this list. The units of measurement correspond to those of International System of Units (SI). In case of use of different measurement units, it is defined locally.

1. Symbols

c_p	Constant pressure specific heat, kJ/kg·K
CH	Chiller
COP	Coefficient of Performance
e	Specific total energy, kJ/kg
E	Total Energy, kJ
EER	Energy Efficiency Ratio
h	Specific enthalpy, u+Pv, kJ/kg
HP	Heat Pump
m	Mass, kg
$\dot{m}$	Mass flow rate, kg/s
p	Pressure, kPa (or bar)
P	Power, kW
Q	Total heat transfer, kWh
$\dot{Q}$	Heat transfer rate, kW
R_f	Fouling factor
t	Time, s
T	Temperature, °C or K
u	Specific internal energy, kJ/kg
U	Overall heat transfer coefficient, w/m²·K
v	Specific volume, m³/kg
V	Total volume, m³
$\dot{V}$	Volume flow rate, m³/s
V	Voltage
$\dot{W}$	Work or Power, kW

2. Greek Letters

Δ Difference

Δp Pressure drop, kPa

ΔT Difference of Temperature, K

ΔT_{lm} Log Mean Temperature difference, K

ε Effectiveness

η Efficiency

ρ Density, kg/m^3

3. Subscripts

amb Ambient

ch Chilled or Chilling

co Cooling

el Electrical

in Inlet

h Heat or Heating

out Outlet

1 Initial or inlet state

2 Final or outlet state

4. Superscripts

$\dot{}$ (over dot) Quantity per unit time

$\bar{}$ (over bar) Mean value

List of Abbreviations

AC	Air Conditioning
ANSI	American National Standards Institute
ARI	Air Conditioning and Refrigeration Institute
ASHRAE	American Society of Heating, Refrigerating, and Air Conditioning Engineers
CEN	European Committee for Standardization
COP	Coefficient of performance
EER	Energy Efficiency Ratio
EN	European Norms
LiBr	Lithium bromide
LiBr	Lithium Cloride
ODP	Ozone Depletion Potential
PI	Proportional plus integral
PID	Proportional-integral-derivative
SI	International System of units
SPF	Seasonal Performance Factor
UNI	Italian National Standards Institute

List of Figures and Tables

1 Introduction

1.1 Background

The need to ensure high levels of indoor comfort has resulted in the last decade in a market increase in sales of equipment for winter and summer air conditioning of buildings. Index of an increased standard of living is also the large increase of electrical energy consumption in recent years in the residential sector across Europe, which is actually responsible for about 49% of total energy consumption [1].

Cooling demand, in particular, is rapidly increasing in many parts of the world, especially in moderate climates, such as in most EU member states - see Figure 1-1. For example, the energy demands for air conditioning in Germany have increased from 71 GWh/a (2000) to 122 GWh/a (2005), which is a 72% increase over a period of five years. For the same period of time, the EU-15 countries showed a 51% hike in consumption. At the same time, the cost of electricity has increased significantly.

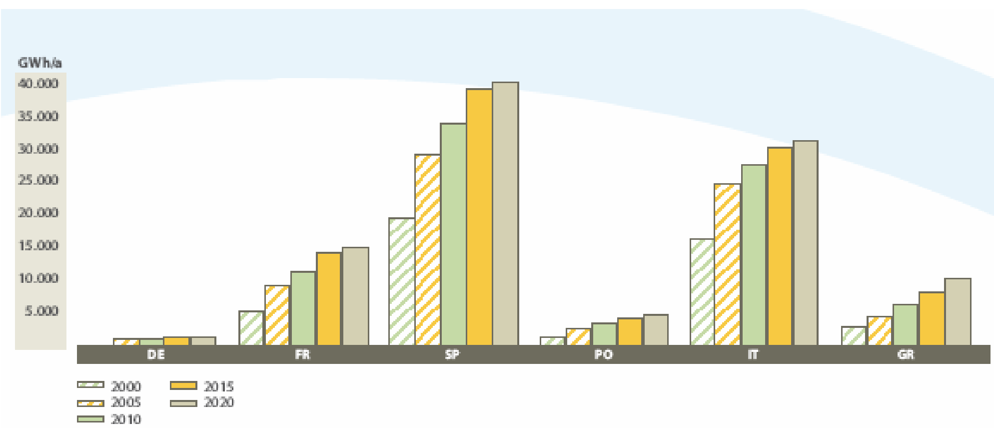

Figure 1-1 Power needed for cooling [in GWh/a] in the following countries: DE-Germany, FR-France, SP-Spain, PO-Portugal, IT-Italy, GR-Greece - Source: EECCAC, 2003

The problem is mostly relevant in hot summer days during which unwanted peaks of fossil and nuclear energy use are revealed, resulting in a menace for the stability of both the grid and of the energy prices.

1.2 Solar Cooling

Electricity and heat demands are increasingly covered from renewable energies. Notably, many private households are choosing to install solar systems for their home heating and domestic hot water preparation.

However, in summer, the opportunities provided by solar installations are not yet used to their maximum potential. The large amount of available energy from the sun cannot be efficiently used, since heating requirements are very low during this season. Most of the solar installations are therefore up to now designed (collectors' field size) to saturate the domestic hot water demand in summer. When large installations are setup to partially cover the winter heating needs, overheating problems are assessed during summer – see Figure 1-2 -.

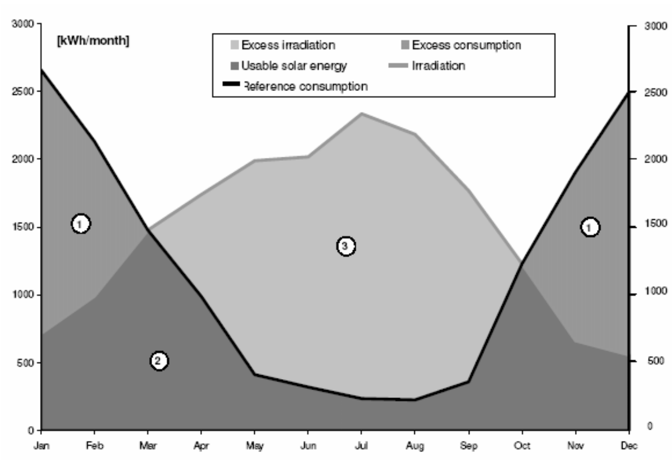

Figure 1-2 – Heating demand and Usable Solar radiation – Source: TU-Graz

In recent years, solar energy has also found application in systems for the production of cold water (sorption machines) for air-conditioning, applications encouraged mainly by simultaneity between high solar radiation and cooling demand. They use the excess energy/heat produced during the summer months for the cost-efficient and climate neutral summer air-conditioning of buildings. Solar assisted air-conditioning is so a promising approach to reduce electricity loads and primary energy consumption, compared to conventional air-conditioning solutions.

These machines can be effectively incorporated into so called *Solar Combi Plus* systems, in which solar energy is used to produce domestic hot water and for both space heating and cooling (see Figure 1-3). The primary energy demand for building air-conditioning (i.e. heating and cooling) can be significantly reduced through the year-round use of the solar installation.

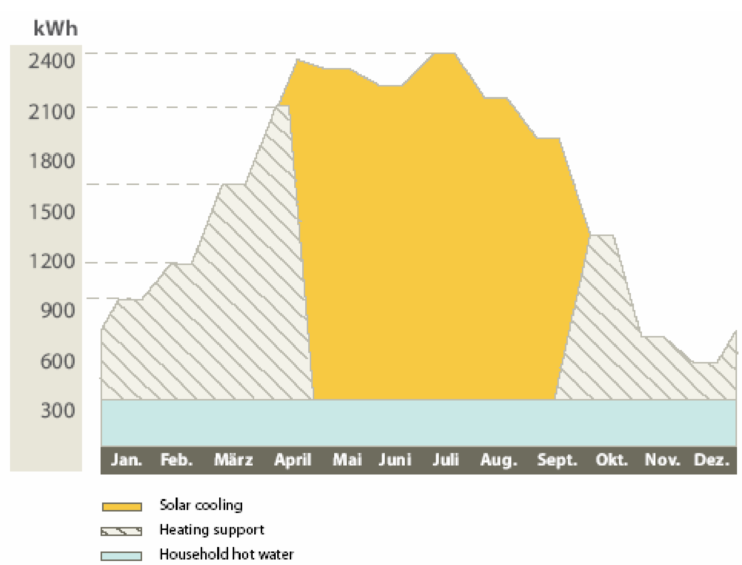

Figure 1-3 Energy yield of solar cooling and heating consisting of a 40m² gross collector surface and the Suninverse absorption chiller – Source: Measurements 2004, SK SonnenKlima GmbH

Solar driven sorption chillers were up to now only manufactured in the high power range (>100 kW_{ch}). Today, machines with rated power between 5 and 30 kW_{ch} are available to be included in solar Combi+ systems for small applications, which make up for the major part of heating and a constantly growing part of cooling demand.

Following this concept, in the last few years, the number of solar heating and cooling installations has recorded a growth of around 50% to 100% each year, in particular for small scale installations. This induced several companies to invest in this technology, introducing new sorption chillers on the market.

To the author's knowledge, 113 large scale solar cooling systems and 163 small scale systems are documented worldwide. The large majority of those (254 installations): **Error! Reference source not found.** shows the distribution of the worldwide number of installations on Countries, classified in small or large scale [IEA Task38].

Most installations are dedicated to office, but 28% of the overall small scale installations serve residential buildings. The overall cooling capacity of thermally driven chillers assisted by solar energy calculated on 268 systems amounts to 15.7 MW: 24.4% of it is installed in Spain, 19,5% in Germany and 17.4% in Italy 7. 14.1 MW of such cooling capacity goes to large scale systems and 1.6 MW to small scale systems.

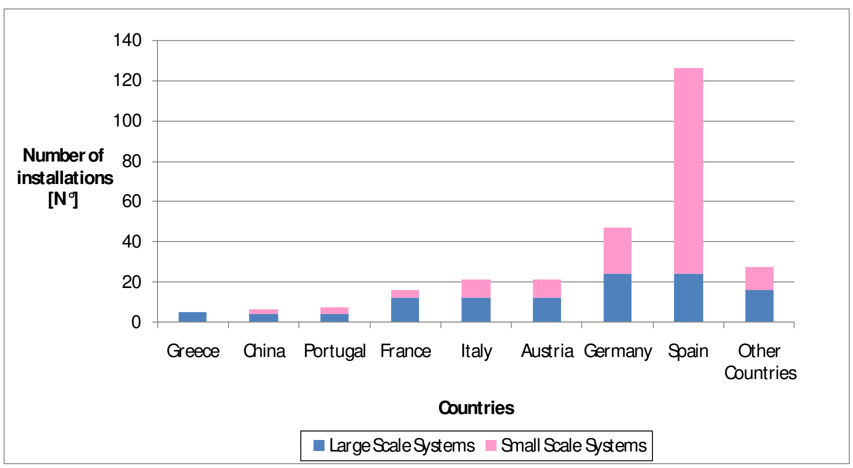

Figure 4 - Total number of solar heating and cooling systems installed in different countries and classified in small or large scale systems. "Other Countries" include: Armenia, Australia, Belgium, Denmark, Egypt, Japan, Kosovo, Lichtenstein, Malta, Mexico, Netherlands, Singapore, South Africa, Switzerland, Syria, Turkey, UK, United Arab Emirates and USA [Source IEA Task38].

1.3 Sorption Chillers Testing

An equal updating of the normative scenario did not follow the market development, leaving manufacturers without useful tools and references for marking their products before the release on the market. In fact, the tests prescribed by the standards for the evaluation of the sorption chiller performance are, in most cases, inadequate.

The need of having common references induced the author to assess a test methodology applicable to all chillers technologies.

Therefore, the first phase of the research work consisted, on one hand, in the assessment of the state-of-the-art of the sorption technologies employed in the air-conditioning sector; on the other the most relevant standards for chillers' performance testing were analyzed. The inapplicability is mainly due to the fact that the existing standards treat sorption chillers as if they were all continuous machines without taking into account the real nature of their operation. Moreover, all chillers are classified only on the basis of the number of heating phases – i.e. single or double effect -, driving technology – i.e. direct or indirect fired - and sorbent physical properties – i.e. liquid: absorption; solid: adsorption-. To overcome this limitations, a new way to classify thermally driven chillers based on their working mode - i.e. continuous, semi-continuous and batch mode - was identified and, based on that, a new test procedure was developed. The applicability of its prescriptions was tested on the most relevant sorption chiller technologies.

Secondly, an experimental facility was designed and installed at EURAC Research in Bolzano, suitable for testing sorption chillers with nominal cooling capacity up to 20 kW. Thanks to its modular and flexible design concept, the laboratory allows evaluating the performance of the chiller investigated under a number of reality-like working conditions (varying climatic conditions, cooling and heating loads) and system's configurations (storage volume and type on the hot and cold side, kind of heat rejection system, control strategy, etc).

The laboratory was used to validate the test procedure. A prototype of ClimateWell CW10 was used to the purpose, since it represents the most general case of sorption technology: it is an absorption chiller working in batch-mode; with respect to this, all other technologies are sub-cases. The results achieved with regard to the tested chiller were compared with the results obtained by partner research institutes from tests carried out on other two sorption chillers. In this way, a validation of the test procedure with regard to the three main chillers' technologies was carried out.

The test procedure was validated only with regard to small-scale sorption chillers. Anyway, it has to be précised that it is applicable also to medium and large scale sorption machines as well as to thermally driven heat pumps (for which different rating conditions and COP shall be defined although). Since the standards dealing with this topic have all the same structure, it was thought to employ it also for developing the new test procedure – see Figure 1-5. The advantage of using this approach lies in the fact that the procedure is developed step by step making easier the analysis and the definition of the prescriptions. Furthermore, a procedure developed in such a way might favor its future standardization.

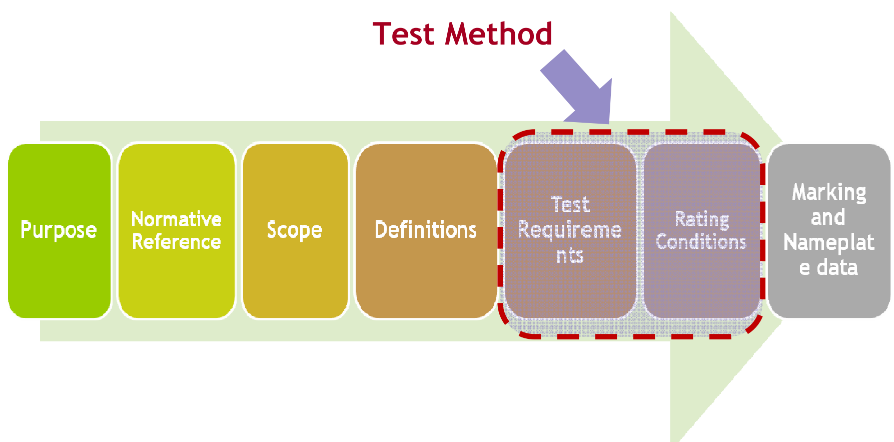

Figure 1-5 - Typical structure of standards concerning test procedures and performance evaluation methods for chillers and heat pumps

1.4 Structure of the Document

The main body of this work is presented in *Chapters 2-6*. In detail, in *Chapter 2,* the basic working principles of sorption chillers and their new classification based on their working modes – i.e. continuous, semi-continuous and batch mode - are explained. In *Chapter 3*, an analysis of relevant existing standards concerning test procedures for sorption and conventional chillers is carried out and, on the base of achieved results, new test procedures for sorption chillers based on their working modes are defined in *Chapter 4.* In particular, for each working mode, rating conditions, test requirement e evaluation method are individuated and explained. In *Chapter 5* the tool used for the validation of the developed test procedure and for testing the chiller CW10 – i.e. a test facility for small scale sorption chillers installed in Solar Combi Plus systems – is presented. In Chapter 6, the validation of the test procedure on different sorption chillers and the analysis of obtained results are carried out. In particular, the methodology and tests performed on considered chiller – i.e. CW10 - are described and the achieved results analyzed and compared with those obtained by applying the developed procedure on other commercial chiller.

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2 Thermally driven chillers

The thermally driven chiller is the core component of *Solar Cooling* technology. It plays a key role both in the planning and in the running of these kinds of plants: dimensioning, choice of components to install, control strategies as well as first costs – and so the *Simple Pay Back*– depend on it. Although it has been revaluated just in the last years thanks to its capability to use waste heat – e.g. cogeneration systems – and to be coupled with renewable energies, it is an old technology. Conceived already at the end of 18th Century and experimentally studied by Michael Faraday in 1824, the first sorption cycle was set by Ferdinand Carré in 1859 for the ice-fabrication based on cold production by evaporation. He used a mixture of Water and Ammonia as working fluid. This first prototype was employed for the sea freight of chilled meat and, in few years, reached a predominance position on mechanical compression systems, which hold till the first decades of 20th Century. Nevertheless, in 30s, the development of electricity production and the reliability of electric engines let compression systems come up again, till when the arrival of CFC as coolant fluids confirmed their complete supremacy. Only in the 70s, the worldwide energetic crisis and the birth of a new “ecological” sensitivity, imposing the rational use of energy, favoured the development of new applications for thermally driven chillers. Nowadays, the technological progresses and the availability of new materials have allowed the reintroduction of this technology with good thermodynamic efficiencies also for small scale air conditioning systems, representing a valid alternative to conventional ones.

In this work, special attention is given to these components. In particular, in the present chapter, the basics of sorption cycles are explained and a description of the operation of existing thermally driven chillers is provided.

2.1 Fundamentals

The term "*chiller*" or "*heat pump*" – depending on the purpose – denotes a group of technologies that transfer heat from a low temperature to a high temperature source. Such transfer, to occur, requires a thermodynamic input in form of either heat or work. This is made clear in the Clausius statement of the second law of thermodynamics which can be stated as:

> *"It is impossible for any system to operate in such a way that the sole result would be an energy transfer by heat from a cooler to a hotter body"*

In thermally driven chillers, this input is in the form of heat. Just such aspect characterizes the whole technology and differentiates it from that conventional, for which the input is in form of work.

From a thermodynamic point of view, sorption systems – both chillers and heat pumps – can be seen as a combination of two - direct and indirect - idealized energy conversion cycles. To understand better this concept, it's necessary to remind some notions of thermodynamics related, in particular, to Carnot and Rankine cycles, in order to describe the operation of these systems within the larger context of what is thermodynamically possible. Figure 2-1 shows a Carnot cycle for power generation on a temperature-entropy diagram. It consists of four reversible transformations processed in clockwise direction. An isotherm – represented by process line AB -, in which the heat Q_2 is added to the working fluid at temperature T_2. An adiabatic transformation – process line BC – in which there is an isentropic work production. An isotherm – process line CD – in which the heat Q_1 is rejected at the temperature T_1 and, finally, an isentropic input of work W_{input2} represented by process line DA. To be noted that the energies supplied to the cycle are conventionally represented by an arrow pointing to the process line; while energies produced or released to the ambient are conventionally represented by an arrow pointing out from the process line. Since the whole cycle is reversible, it's possible to find some graphical correspondences between areas and involved energies. The area ABCD represents the net amount of work produced, W; while the area CDEF the amount of thermal energy

Q_1 rejected by the cycle at T_1, assuming that E and F are at T=0*K*. The area ABFE is the sum of the other two areas and represents the amount of heat Q 2 supplied to the cycle as confirmed also by the first law of thermodynamics (Equation 2-1).

Equation 2-1 $$Q_2 = Q_1 + W$$

The efficiency of this cycle is calculated as the ratio between the gain – i.e. the net amount of the work produced - and the total heat supplied at high temperature.

Equation 2-2 $$\eta = \frac{W}{Q_2}$$

Using the first law (Equation 2-2) and the second law of thermodynamics, this last written in a simply form since the entropy production is zero for reversible cycles:

Equation 2-3 $$\frac{Q_2}{T_2} + \frac{Q_1}{T_1} = 0$$

the efficiency can be modified as:

Equation 2-4 $$\eta = \frac{T_2 - T_1}{T_2}$$

The efficiency expressed as in the Equation 2-4 is usually termed the Carnot efficiency factor for power generation.

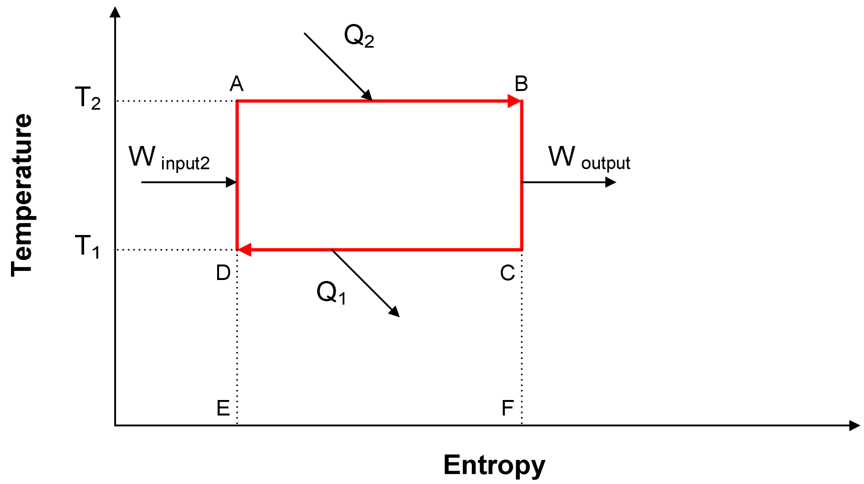

Figure 2-1Carnot cycle for power generation on a Temperature-Entropy diagram

If the Carnot cycle for power generation is run in reverse direction, a heat pump - or chiller - cycle is obtained. In this case, the heat is supplied to the cycle at the lowest temperature and is rejected at highest temperature. For making this possible, a thermodynamic input in form of work is required. Figure 2-2 shows a Carnot cycle operated as heat pump.

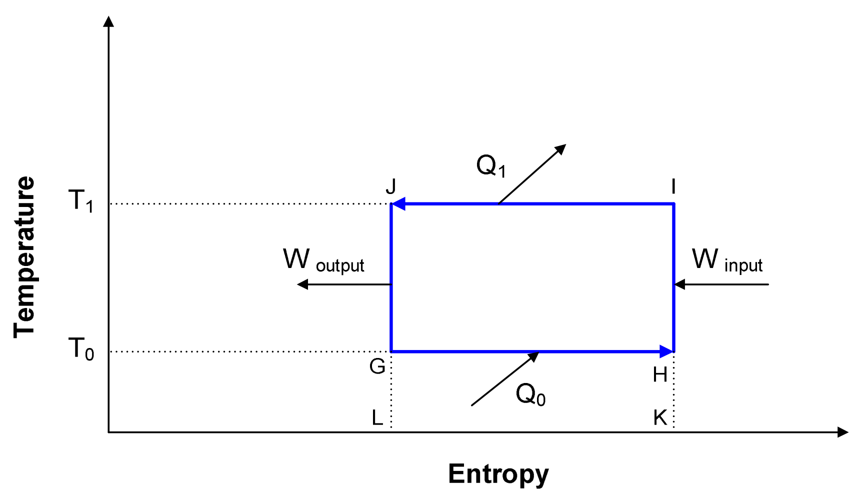

Figure 2-2 Carnot cycle for chilling (or heat pumping) on a Temperature-Entropy diagram

In detail, the heat Q_0 is added to the working fluid at T_0 – i.e. the low temperature - along the isotherm GH; then the fluid is compressed isentropically along the adiabatic process HI and the heat Q_1 is rejected at T_1 – i.e. the high temperature – along the isotherm IJ; finally the fluid is expanded isentropically along the adiabatic process JG. Also in this case, the areas represent the energy transfers. In particular, the area delimited by the cycle, IJGH, represents the net amount of work required as input – and not produced like in direct cycle -.The area HGLK is the amount of heat Q_0 absorbed from the source at lowest temperature, assuming that l and k are at T=0*K*. While the amount of heat rejected Q_1 at T_1 is represented by area IJLK. Regarding the cycle efficiency, since, for reverse cycles, it is always grater than 1.0, the term "*coefficient of performance*" – COP – is customarily used and it is calculated as ratio of the achieved benefit – that, for chillers, is the heat removed from the source at lowest temperature; while, for heat pumps, is the heat rejected to the sink at highest temperature – divided by expenditure – i.e. the net work requirement -. Using the first and the second law of thermodynamics, the COP for heat pumps and chillers can be written respectively:

Equation 2-5
$$COP_{HP}=\frac{T_1}{T_1-T_0}$$

Equation 2-6
$$COP_{CH}=\frac{T_0}{T_0-T_1}$$

Which are connected each other by the following equation:

Equation 2-7
$$COP_{CH}+1=COP_{HP}$$

At this point, it is possible to think to combine the two Carnot cycles before explained into one device as shown if Figure 2-3, where the only requirement is that the net amount of work produced by the first cycle – i.e. power generation cycle – is identical to that required by the second cycle – i.e. chiller cycle -.

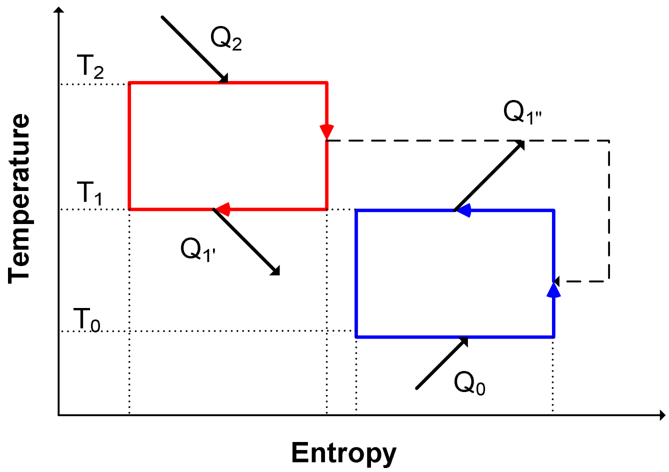

Figure 2-3 Sorption chiller as combination of a power-generation and chiller Carnot cycles

In this way it is obtained a device able to extract – in case of chiller applications - heat from a source at low temperature T_0 and to reject it to a sink at medium temperature T_1 by only using of heat supplied at higher temperature T_2. In this case, the waste heat comes out from both cycles – i.e. it is given by the sum of Q'_1 and Q''_1 - and is rejected at T_1. The coefficient of performance for this device is always given by the ratio of the gain – that in case of chillers is the heat extracted at low temperature; in case on heat pumps is the heat rejected at medium temperature - divided by the expenditure, that is the heat supplied at high temperature. For chilling applications it can be written as:

Equation 2-8

$$COP_{CH} = \frac{Q_0}{Q_2}$$

and by applying the first and second law to the whole device, the equation becomes:

Equation 2-9
$$COP_{CH} = \frac{T_2 - T_1}{T_2} \cdot \frac{T_0}{T_1 - T_0}$$

The Equation 2-9 shows that the performance of these devices, when though as a combination of two Carnot cycles depends only on the three temperature levels.

An analogue reasoning can be done using Rankine idealized cycles. Figure 2-4 shows two Rankine cycles: one chiller cycle and one power generation cycle. Observing the figure, it easy to understand that in order to combine these two cycles into one device the following conditions have to be fulfilled:

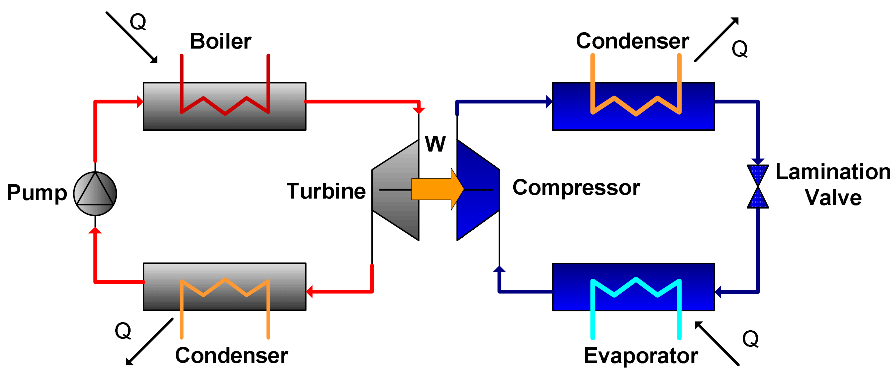

Figure 2-4 Sorption chiller as combination of a power-generation and chiller Rankine cycles

1. The net amount of work produced by turbine in power generation cycle shall be identical to that required by compressor in chiller cycle

2. The stream leaving the boiler of the power generation cycle shall have the same high pressure level and flow rate as the stream entering the condenser of the heat pump cycle, and

3. The stream leaving the heat pump evaporator shall have the same low pressure level and flow rate as the stream entering the power cycle condenser

If all these conditions are respected, the compressor and the turbine can be eliminated and a device driven only by input in form of heat can be obtained.
An important result achievable from this representation concerns the working fluid to be used in these devices. In fact, since the boiler of power generation cycle and the condenser of the chiller cycle are at the same pressure, the working fluid in the boiler shall evaporate at a different temperature from that of the condenser. This temperature shall be higher. This is achievable only if the vapour produced in the boiler comes out from a mixture with a fluid of a much higher boiling point termed absorbent or solvent. This mixture of solvent and refrigerant are in power generator portion and the sorbent circulate only in its own loop.

So, using these two representations, it emerges that in the sorption chillers there are three levels of temperature from which the performance depends on; two levels of pressure and the working fluid shall be a mixture of solvent and refrigerant.

2.2 Sorption cycle: thermodynamic phases

From the thermodynamic explanation of sorption systems done in the previous paragraph with the help of idealized energy conversion cycles, some of the key-aspects characterizing this technology have come out. In particular, it has emerged that, such systems:

- are devices fed substantially by heat supplied at high temperature
- develop approximately on three temperature levels, from which the coefficient of performance depends on, and on two pressures levels
- use a mixture of sorbent and refrigerant as working fluid since components dedicated for evaporation and condensation processes are at the same pressure level and, for having different working temperatures, they shall utilize the boiling point elevation of mixture.

Concerning this last aspect, depending on the sorbent being a liquid or solid phase, the sorption systems are classified respectively in: absorption systems and adsorption systems. Although the physical state of the sorbent determines to a large degree the thermodynamic performances of these machines as well as their design and working modes, it doesn't have any influence on thermodynamic phases that usually form the basic cycle. Such phases are four: Desorption, Condensation, Evaporation and Ad/Absorption and are shown in Figure 2-5. In detail:

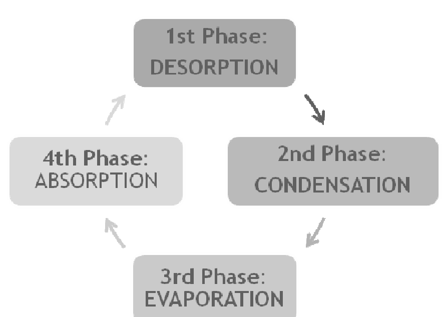

Figure 2-5 Four phases of sorption chiller cycle

- *DESORPTION:* is the phase in which the refrigerant is desorbed from the mixture thanks to the supplying of heat at high temperature and at high pressure. During this phase vapour of refrigerant is produced.
- *CONDENSATION*: is the phase in which the vapour of refrigerant, produced during Desorption phase, condenses through rejecting heat to a sink at medium temperature and at high pressure.
- *EVAPORATION*: the condensed refrigerant evaporates through extracting heat from a source at low temperature. This phase occurs at low pressure – the pressure level is changed thanks to a lamination process -.
- *ABSORPTION*: the vapour of refrigerant produced in the evaporation phase is absorbed by the sorbent through rejecting heat to a sink at medium temperature and low pressure.

At this point it can be useful to see how these four phases are processed in absorption and adsorption chillers. Figure 2-6 shows a block diagram of basic absorption cycle, which is formatted as if it were superimposed on Dühring plot of working fluid. This means that the position of each component indicates the relative temperature, pressure and mass fraction.

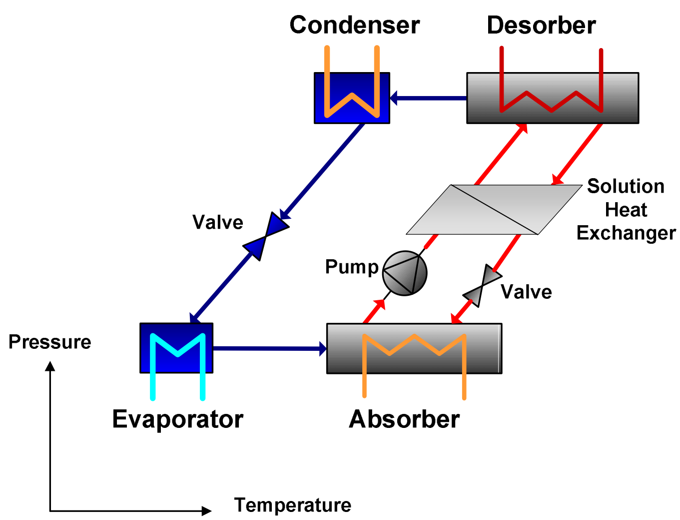

Figure 2-6 Absorption cycle schematic

The cycle is processed in this way:

1. The liquid solution is pumped from to absorber to the desorber (higher pressure) in which the heat at high temperature is supplied to the solution. The Desorption process takes place and the refrigerant starts to evaporate – thanks to its higher volatility respect to the sorbent -. The solution rich of sorbent goes back to the absorber passing thorough a solution heat exchanger where here it releases heat to the poor solution going to the desorber. It has to be noted that the solution circulates only in its own loop between absorber and desorber.
2. The refrigerant vapour produced starts the typical chiller cycle. It goes into the condenser in which is condensed releasing heat to a sink at medium temperature and at high pressure.
3. Then condensed refrigerant changes its pressure thanks to a lamination process and flows to the evaporator where evaporates, extracting heat from low-temperature heat source at low pressure.
4. The vapour of refrigerant flows into the absorber in which it is absorbed by the rich solution. This operation releases heat to medium-temperature sink at low pressure.

In Figure 2-1 is represented the basic scheme of an adsorption chillers.

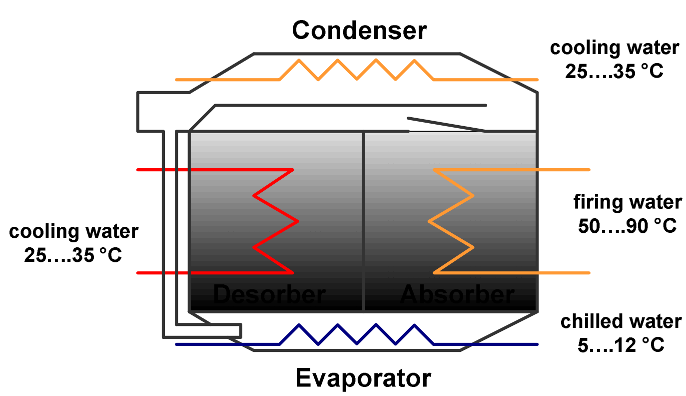

Figure 2-7 Schematic of an adsorption chiller

In this case, since the sorbent is in solid phase, it is not possible to have a continuous flow between desorber and adsorber like in absorption chillers. Thus, to adsorb and desorb the refrigerant, the solid sorbent has to be alternatively cooled and heated. This results in a discontinuous chiller operation.

In these chillers, the sorbent is contained in two under-vacuum chambers, which serve as desorber or adsorber respectively depending on phase they are processing – i.e. absorption or desorption phase -. When, one chamber is completely charged – i.e. the sorbent is completely dry - and the second chamber is completely discharged– i.e. the sorbent is completely wet –, the process is stopped and they interchange their function. Just this aspect makes the operation of these chillers discontinuous.

2.3 Sorption chillers: features and classification

The physical properties of working pairs used in sorption chillers as well as the modalities with which the heat at high temperature is supplied and used within the machine, further to determine strongly the design, represent the criteria with which, at present, these machines are termed and classified. Below, the main features characterizing sorption chillers are briefly explained and the resulting classification and terminology illustrated.

2.3.1 **Working pairs** *(Sorbent physical properties)*

In the previous paragraph, during the description of the working principles of sorption systems, it has been specified that depending on the sorbent physical state – i.e. if it is liquid or solid – the sorption chillers are distinguished in absorption and adsorption chillers. This feature, even if it doesn't influence the thermodynamic basic cycle, determine the machine working mode and the technological solution used to realize it. For this reason, it is employed both in terminology for the chiller denomination and for their classification. By passing, now, to the larger concept of working pairs, it is possible to say that the choice of a specific mixture as working fluid, besides structural characteristics, determines directly the conditions at which the machine can work.

Furthermore, since most of the irreversibility, occurring during the chiller operation, are due to their properties, the performance and the efficiency of the whole machines depend strongly on them. The preference for either working pair is based on several factors, first of all, the specific chiller application. For instance, if the unit is used for air-conditioning purposes, the working temperatures at the evaporator are above 0°C and this means that working pairs using water as refrigerant can be used; while for food storing, for which the working temperatures at the evaporator are below 0°C, the working pairs to be used shall have as refrigerant a substance with freezing point lower than 0°C like ammonia, methanol, etc…

The same kind of reasoning could be done for the condensation temperature and for the driving temperature at the desorber. In many cases, in fact, according to the temperature requirements of technologies coupled with the sorption chillers, it make necessary use specific working pairs. For example, for working with high temperatures at the condenser – e.g. when a dry air cooler is used as heat rejection system -, mixtures of salt solutions like Water-Lithium Bromide, Water-Lithium Chloride, etc should be avoid because the salt could crystallize; while mixtures like Water/Ammonia should be preferred. Instead, depending on the available temperature at the heat source, it should use: adsorption working pairs for very low driving temperatures – lower than 70°C – like Water/Silica gel, Methanol/Activated carbon etc.. ; salt solutions for low-medium temperatures - around 80 ÷ 90°C - and Water/Ammonia, Methanol/Activated carbon, Water/Zeolite pairs for high driving temperatures – 105 ÷ 120°C -. To be noted that in the nomenclature of working pairs the refrigerant is written first, and after the sorbent.

Table 2-1 shows some properties of sorbent-refrigerant pairs commonly used as working fluid in absorption and adsorption chillers.

Table 2-1 Properties of most common working pairs

WORKING PAIRS	*HIGH LATENT HEAT OF REFRIGERANT*	*LOW FREEZING TEMPERATURE*	*SATURATION PRESSURE OF REFRIGERANT*
Absorption Working Pairs:			
Water/ Lithium Bromide	Excellent	Limited Application	Too low
Water/ Lithium Chloride	Excellent	Limited Application	Too low
Ammonia/ Water	Good	Excellent	Too high
Adsorption Working Pairs:			
Water/ Silica Gel	Excellent	Limited Application	Too low
Water/ Zeolite	Excellent	Limited Application	Too low
Methanol/ Activated Carbon	Poor	Excellent	Good
Ammonia/ Activated Carbon	Good	Excellent	Too high

2.3.2 Number of effects

This feature concerns the number of times that heat supplied at high temperature is used within the machine to produce cold. Usually, sorption systems with two or more effects are used both to improve the COP - since the COP of a single effect sorption chiller is around 0.7 - and to use the availability of heat at higher temperature that, otherwise, single-effect machines do not use due to the large increase of irreversibility and crystallization phenomena occurring. in solar cooling applications, single-effect sorption chillers are mostly used since the required temperatures are relatively low. Figure 2-8 shows a schematic of a double-effect absorption chiller. It can be viewed as two single-effect cycles stacked on the top of each other, where the generator of top cycle produces refrigerant vapour at high temperature and pressure, which is condensed at the same pressure. The latent heat of condensation is used as driven heat for the Desorption of bottom cycle which is at lower temperature and pressure. The COP of these kinds of systems is around 1.1 or a little above.

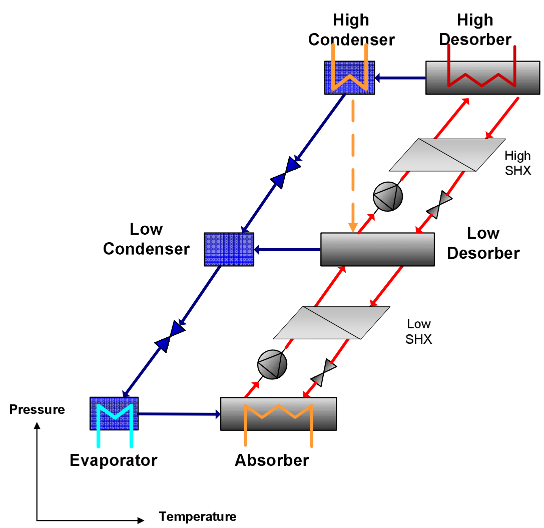

Figure 2-8 Double-effect sorption chiller schematic

Currently only single-effect and double-effect sorption machines are present on the market. Research is focusing also on three, four and half effect systems, as potential improvements of cooling performance can be obtained – i.e. COP by 1.7 to 2.2 –. Nevertheless their requirement to have very high feeding temperature restricts their realization and their application.

2.3.3 Driving technology

The terms "*direct fired*" and "*indirect fired*" indicate the modality with which the heat is supplied to sorption chiller, i.e. if the chiller is fed by direct combustion of fossil fuels or if it is fed by waste heat or hot water/steam produced by solar collectors or by any other heat source. This feature influences largely the chiller design as well as the way for evaluating the heat supplied to the machine. In case of "direct fired", the chillers are provided of a burner for the direct combustion of fuel; in case of "indirect fired", a heat exchanger at the desorber is enough. The choice of either technology depends on many factors often correlated among them. They are:

- *Availability of waste heat or free steam/hot water* – e.g. heat produced by solar collectors.
- *Working pairs:* they required different temperature for the desorption process
- *Required chilling effect:* depending on the desired outlet temperature at the evaporator, the working pairs change and with them, the feeding temperatures. For high feeding temperatures direct fired technology is preferable. For example, if the desired chilling temperature is around 0°C, Ammonia/Water is used as working pair. Since it requires high temperature for desorption of water, a direct fired technology is usually used.
- *Desired cooling technology:* depending on the desired condensation temperature and thus cooling technology, the working pairs change and with them, the feeding temperatures.

2.3.4 Classification

The above-explained features, once specified, provide a quite clear description of sorption chillers at least from a design point of view. For this reason, they are employed both in the terminology commonly used for identify them – e.g. single- effect indirect-fired adsorption chillers – and as criteria for their classification as shown in Figure 2-11. According to this classification, also the few existing standards concerning test and performance procedures for sorption chillers have been developed. Nevertheless, since such features are more related to design aspects, it could happen that, following this classification, chillers having different working modes are treated in the same way.

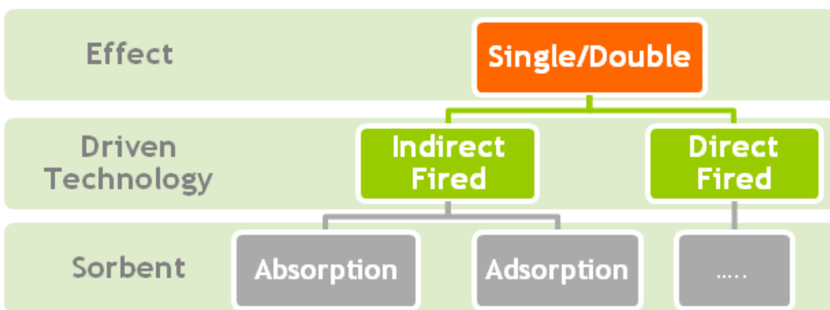

Figure 2-9 Classification of sorption chillers based on Number of effects, Driven Technology and Sorbent Physical State criteria

Also the literature and the few dedicated existing standards for testing and According An example of this can be represented by chiller CW10, produced by ClimateWell, and Suninverse, produced by SonnenKlima, which, being both single effect indirect fired absorption chillers, are classified in the same way.

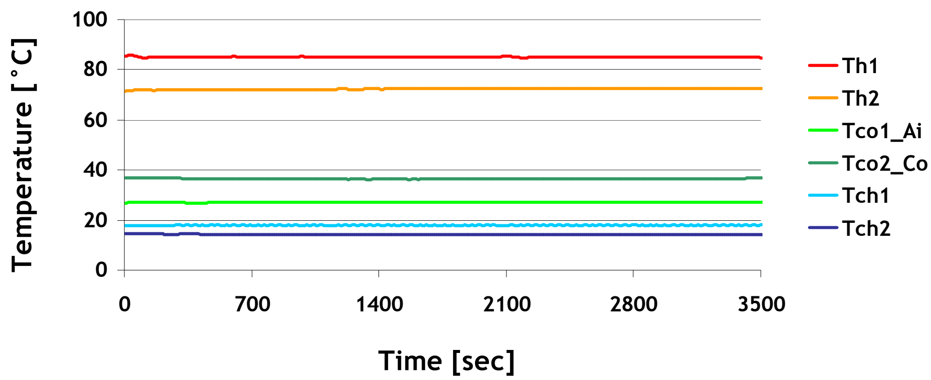

Figure 2-11 Temperature profiles vs. time of Suninverse (SonnenKlima) absorption chiller during its operation - Source: TU Berlin

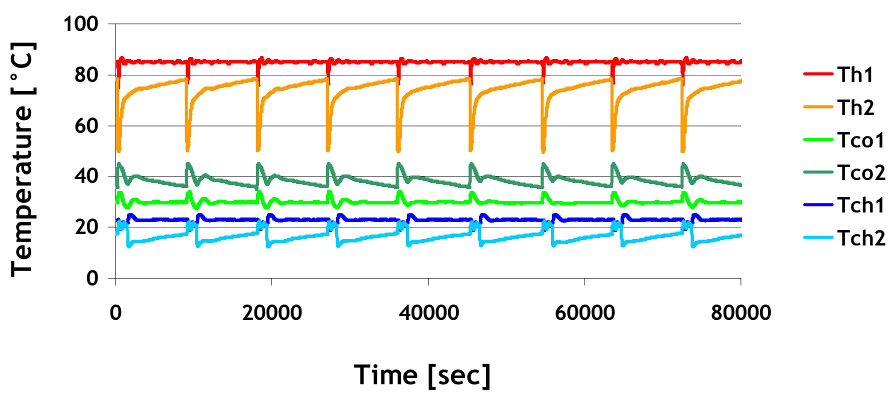

Figure 2-12 Temperature profiles vs. time of CW10 (ClimateWell) absorption chiller during its operation - Source: EURAC

Even so , observing the graphs of their working temperatures shown in Figure 2-11 and Figure 2-12, it's easy to understand that these two chillers are quite different and that, besides these three parameters, it's necessary to find a new ways to take into consideration working aspects. Furthermore, in Chapter 3 it is shown that the current classification is not even adequate for testing the chillers.

2.4 New concepts and classification

The working mode, i.e. the modality with which the four phases of the sorption cycle are processed, is the feature that mostly determines the behaviour and the performances of sorption chillers. In fact, while working pairs meant as sorbent physical state, number of effects and driving technologies are all features mainly related to the design aspect, the working mode concerns uniquely the operation of these machines. According to this feature, three new sorption chillers' categories have been individuated. They are: continuous, semi-continuous and batch mode. it can be possible to define generically:

- *Continuous chillers* as those chillers in which the energy and mass exchanges are stationary, i.e., if the boundary conditions are stationary, the exchanges are uniform in time but not in space;

- *Semi-Continuous and Batch mode chillers* as those chillers in which the energy and mass exchanges vary in time and in space, i.e., if the boundary conditions are stationary, the exchanges vary cyclically.

Below, these three categories of chillers are explained more in details.

2.4.1 Continuous chillers

Continuous chillers are those chillers in which the four phases, desorption, condensation, evaporation and sorption are processed continuously. In order to do this, each phase is processed by a dedicated component within the machine as shown in Figure 2-13

Such components are the four heat exchangers which exchange heat with external sinks/sources at high – desorber-, medium – condenser and absorber – and low temperatures. The characteristic operation of continuous chillers is shown Figure 2-11 and is typical for absorption chillers having dedicated phase components.

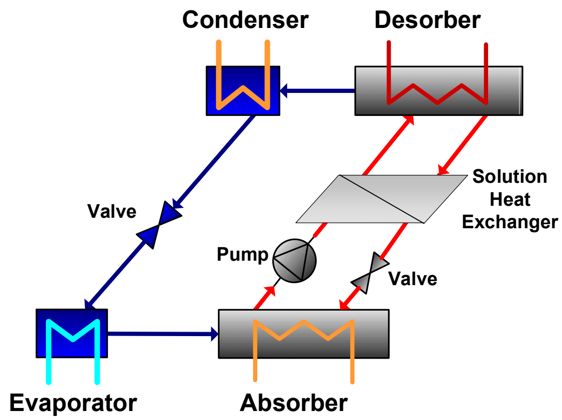

Figure 2-13 Schematic of a Continuous chiller

2.4.2 Semi-Continuous chillers

In semi-continuous sorption chillers, the four phases are periodically shifted among the internal components. This means that the four heat exchangers, interacting periodically with external ambient, change their function. Usually the shift occurs between the phase couples processed at the same pressure level, i.e. between desorption/condensation and sorption/evaporation, this condition is dictated by structural limits since the mixture and the refrigerant have to flow in their own loops.

In some cases, like that shown in Figure 2-14, it could happen that condenser and evaporator remain the same – i.e. they don't change their working phase - and the only components that shift are desorber and absorber.

A fundamental aspect to be underlined for this kind of chillers, is that the two phases' couples – i.e. desorption/condensation and sorption/evaporation - have to be always processed simultaneously. This is the reason why semi-continuous chillers always consist of two or multiples of two units each processing one couple.

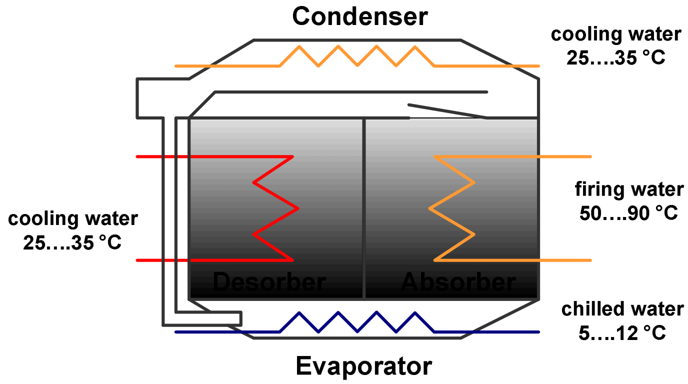

Figure 2-14 Schematic of a Semi-continuous chiller having fixed condenser and evaporator and shifting desorber and absorber

The last aspect to be underlined is that the presence of shift, usually called "swap", determines, during the operation of these machines, rough fluctuations of all controlled quantities – see Figure 2-15- that don't allow the applicability of many of standard prescriptions for their testing and performance evaluation. Furthermore, just after the swap, since the temperatures and the pressures levels have to change due to the shift from a component to the other, these machines require a certain time for reaching again their stationary equilibrium. During this period, any chilling power is delivered. This working mode is typical for adsorption-chillers.

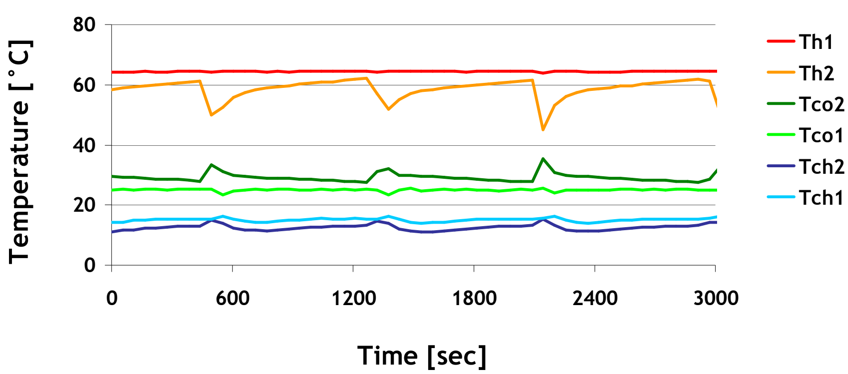

Figure 2-15 Temperature profiles vs. time of semi-continuous chiller during its operation - Source: LESBAT

2.4.3 Batch mode chillers

As far as for semi-continuous chillers, batch mode chillers have a cyclic operation: the four phases periodically shift among the internal components as well. The only difference is that, in this case, the couples of phases - desorption/condensation and sorption/evaporation - might be processes once at a time: simultaneity is not required.

This implies that also a single unit might perform the whole thermodynamic cycle in two consecutive periods with a following intermittent cold production. Obviously condenser and evaporator cannot remain the same but they have necessary to swap. For this reason, these chillers consist of one or more units, each capable of processing one couple of phases once at a time.

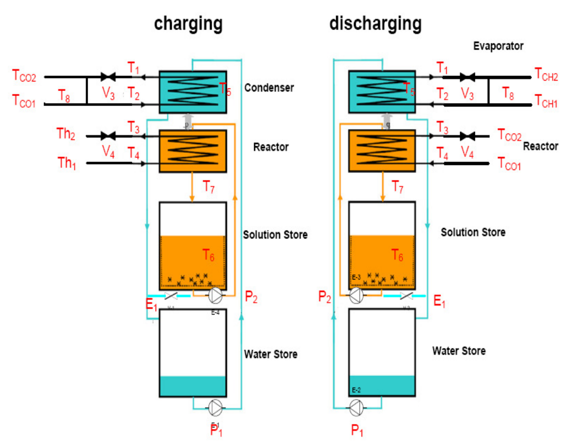

Figure 2-16 Schematic of a Batch mode chiller – Source: ClimateWell

Also in this case, the presence of swaps determines rough fluctuations of controlled quantities but with peaks more accentuated - see Figure 2-12 – causing the inapplicability of the restrictions prescribed by tests' standards. Batch mode chillers can be both absorption and adsorption chillers.

2.4.4 New classification

On the base of working mode, it is possible to classify in the right way all existing chillers which can be considered as sub-types of batch mode chillers – see Figure 2-17.

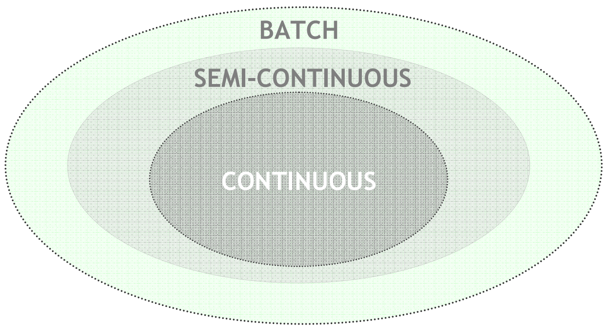

Figure 2-17 Schematic of a Batch mode chiller – Source: ClimateWell

Using working mode as new criteria for the classification of sorption chillers, the new classification is that one shows in figure.

In this case the two sorption chillers CW10 and Suninverse can be so classified:

- CW10: single-effect indirect fired batch-mode absorption chiller
- Suninverse: single-effect indirect fired continuous absorption chiller
-

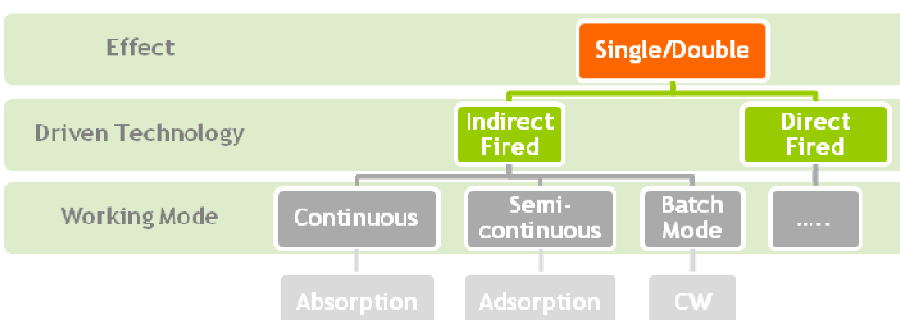

Figure 2-18 New classification

3 Standards for test procedures

Testing and performance evaluating activities carried out on a specific product have, as the primary objective, the marking of the product itself. And, because it represents the interface between the manufacturer and the final customer, besides general information -i.e. nameplate, code …- it shall contain performance data evaluated in reality-like working conditions.

In order to guarantee this, trade associations and professional engineering societies have developed official standards that act as a references to regulate this process. They contain prescriptions concerning the working conditions at which a specific product shall be tested as well as specifications about the equipment and the methodology to be used during the test execution: they all have to be followed by manufacturers before bringing their products on the market.

Since the present work is intended to develop test procedures for sorption chillers installed in air-conditioning systems, it cannot get out of considering what the current normative system places on the market for who - like manufacturers - wants to test and evaluate thermodynamically these kinds of devices. This allows, on one hand, knowing the state of the art of existing dedicated standards; on the other hand, learning their applicability limits, in particular, in case of the new prototypes recently introduced on the market. Furthermore, a deep understanding of normative scenario, allowing individuating the main critical aspects, represents the starting point for the test procedures development. For this reason, in the present chapter, the most relevant standards have been collected and analyzed.

Nevertheless, it must state beforehand that, because all standards applying to air conditioning units refer both to chilling and heating modes, during the analysis carried out, some references related to heat pumps are made, even if the interest of the present work applies to units working in chilling mode.
More detailed information regarding the heat pump working mode are in the Annex A.

3.1 Classification and selection criteria

Before going into the selection and analysis of dedicated standards for sorption chillers and heat pumps, it is necessary to do a first important classification. The interest normative scenario, in fact, can be essentially subdivided in two big categories: commercial standards and statutory standards.

Commercial Standards are those standards issued by trade associations and professional engineering societies such as ASHRAE (American Society for Heating and Refrigeration engineering), ARI (American Refrigeration Institute), IIR (International Institute for Refrigeration), in USA, and the broad-based European organizations cooperating in CEN, in Europe. Depending on the scope, their target is to prescribe rules for the construction, testing and performance evaluation of the product for which they were formulated. In particular, for testing the units, they provide both the standard rating conditions and, for specific cases, application rating conditions which are used when the real working conditions are different from the standard ones. Furthermore they are based on three dots:

1. They are non-binding for the introduction of new products;
2. They are formulated for the professional societies that issued them by chiller manufacturers, practicing engineers, installers, contractors, users and researchers;
3. They are subjected to periodic review and amendment as the state of art of the technology is going on.

Statutory Standards are those standards issued by governmental organizations. They are more related to the safety aspects–i.e. mechanical integrity and robustness of the unit - than their performances. In fact, they prescribe the rules to be followed in order to guarantee the person and good safety.

For the prefixed targets, only the commercial standards have been taken into account since they deal primarily with thermodynamic performance of the units.

Clarified this, the criteria used for the selection of interest commercial standards are the following ones:

1. *Purpose*

Since, in this work, the constructional aspect of sorption chillers is not considered but only their behavior under different working conditions, the commercial standards or parts of them related to rules for the construction of chillers have been left out. While, those ones containing test requirements, test procedures and rating conditions have been selected and used for the analysis.

2. *Scope*

Regarding the scope, all standards applying to devices for space heating and/or cooling have been considered even if they are not directly related to sorption chillers (most of them refer to electrical units). The reason of this choice is due to the fact that since just few standards are dedicated to thermally driven chiller, this is the unique way to have more references for developing test procedures. Furthermore to know how the conventional chillers and heat pumps are tested and evaluated allows comparing in properly way the two technologies. Therefore, the selected standards apply to: electrically compressor chillers and heat pumps, single and double effect, direct and indirect fired sorption chillers and heat pumps.

3. *Performance Figures*

Regarding this third criteria, the selection has been carried out on two levels: the first one including only those standards having the focus on the sole machine without considering its interferences with the external ambient. In this case, only tests at stationary conditions and procedures for instantaneous COPs calculations are prescribed. The second one, including those standards that take into account the combinations between the machine with climate and the user demand and their transitory effects on the machine. In this case, transient and steady tests are prescribed as well as procedures for the calculation of seasonal and instantaneous performance factors. Furthermore instructions for user demand's determination– i.e. heating, cooling and DHW demand- are also included.

3.2 Overview and analysis of relevant existing standards for chillers and heat pumps

On the base of the above explained criteria, the most relevant commercial standards for chillers and heat pumps have been collected and summarized by groups in Table 3-1, depending on the main performance figures calculated inside. The aim of this collection was to compare and analyse, for each group, the standards' prescriptions – i.e. rating conditions, test requirements, tolerances and uncertainties … - and to understand their applicability limits for different types of sorption chillers.

Table 3-1 Relevant existing standards for chillers and heat pumps

PERFOMANCE FIGURE	NAME	SHORT TITLE
COP/EER		
	ANSI/ARI 560	Absorption Water Chilling and Water Heating Packages
	EN 14511	Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors, for space heating and cooling (Parts 1-4)
	EN 12309	Gas-fired absorption and adsorption air-conditioning and/or heat pump appliances with a net heat input not exceeding 70 kW
	prEN255-3	Testing and requirements for marking for domestic hot water units
SCOP/SEER		
	ANSI/ARI 560	Absorption Water Chilling and Water Heating Packages
	VDI 4650-1	Short-cut method for calculation of the annual effort figure for the heat pumps (heating)
	prEN14825	Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors, for space heating and cooling – Testing and rating at part load conditions
	EN 15316-4-2	Heating systems in buildings –Method of calculation of system energy requirements and system efficiencies: Space heating generation systems, heat pump systems (Part 4-2)
SPF/PER		
	EN 12309	Gas-fired absorption and adsorption air-conditioning and/or heat pump appliances with a net heat input not exceeding 70 kW
	EN 15316-4-5:	Heating systems in buildings –Method of calculation of system energy requirements and system efficiencies : Space heating generation systems, the performance and quality of district heating and large volume systems(Part 4-5)

A preliminary analysis carried out on selected standards has showed that:

1. Most of them refer to electrically driven chillers and heat pumps. Only the Standards EN 12309 and ARI/ANSI 560/2000 apply to gas-fired absorption and adsorption chillers and absorption chillers respectively. Water fired?

2. All of them refer to continuous chillers and heat pumps even if the Standards EN 12309 has some references to semi-continuous units. In che modo sono trattati e semi-continui?

3. All of them treat units as a black box that can be only probed externally- i.e. not intrusively -. This means that only the coolant temperatures, coolant flow rates and the power input supplied to the tested unit are measured and recorded during the test representing key variables. Furthermore, they prescribe tests mainly at stationary conditions, with exception of transient tests for the occurrence of defrosting cycles.

4. All considered standards present the same structure: purpose, scope, test requirements, procedure for the chilling capacity and COP calculation, rating conditions, marking and nameplate.

On the base of this last similitude, a deeper analysis has been carried out on the following three levels:

1. Standard rating conditions
2. Test requirements
3. Calculation procedure for chilling capacity and COP/EER

3.2.1 Standard rating conditions

Standard rating conditions are those conditions, prescribed by a specific standard, at which a unit shall be tested to be marked. The control variables to which they refer are usually: inlet and outlet temperatures at the condenser and evaporator; depending on the standard (ANSI/ARI 560), water-side fouling factor at the condenser and evaporator, water flow rate at the condenser per kW of cooling power, water flow rate at the evaporator per kW of chilling power; and ambient conditions at test site –i.e. dry bulb temperature. While generator's inlet and outlet temperatures and generator's flow rates are let free as they depend on specific technical choices of manufacturers. The definition of the rating conditions is a function of the technological limits of the equipment that is usually coupled with the unit or that is present on the market, and of the used heat transfer medium – i.e. water, air and brine. Concerning the latter point, the units are denominated in such a way that, if they work in cooling mode, the heat transfer medium for the condenser is indicated first, followed by the heat transfer medium for the evaporator; vice versa if they work in heating mode. The possible couples of heat transfer medium for space cooling units are listed in Table 3-2.

Table 3-2 Heat transfer medium couples for space cooling unit

HEAT TRANSFER MEDIUM		
CONDENSER	***EVAPORATOR***	***CLASSIFICATION***
Air	Air	Air cooled air conditioner
Water	Air	Water cooled air conditioner
Brine	Air	Brine cooled air conditioner
Air	Water	Air cooled liquid chilling package
Water	Water	Water cooled liquid chilling package
Brine	Water	Brine cooled chilling package

Since sorption chillers are essentially liquid chilling packages – i.e. the medium used for heat transfer is water -, only the following couples are considered: Air/Water, Water/Water and Brine/Water. This specification is important because the analyzed standard rating conditions depends on it. Another important specification is that all standards, concerning test procedures or methods for the calculation of seasonal and/or instantaneous COP, refer all to the rating conditions specified in the standards UNI EN 14511-2, UNI EN 12309 -2 and ANSI/ARI Standard 560. For this reason, in the Table 1- 3 , the rating conditions prescribed in these three standards have been collected and subdivided by condenser and absorber temperatures and by heat transfer medium. In this way a direct comparison is possible. Table 3-3and Table A-1 for units working in heat pumps mode are reported in Appendix A page 23.

Table 3-3 Standard Rating Conditions for units working in cooling mode

Standard Rating Conditions - Cooling Mode												
	ANSI/ARI Standard 560	**UNI EN 14511-2**			**UNI EN 12309-2**							
Absorber/Condenser:	W/W, B/W	W/W , B/W	A/W	A/B	W/W	W/B	A/W			A/B		
					T1	T1	T1	T2	T3	T1	T2	T3
Entering Temperature	29.4°C	30	35	35	30	30	35	27	46	35	27	46
(for floor heating or similar application)	-	**30**	**35**	-	-	-	-	-	-	-	-	-
Leaving Temperature	NS	35	NS	NS	35	35	NS	NS	NS	NS	NS	NS
(for floor heating or similar application)	-	**35**	**NS**	-	-	-	-	-	-	-	-	-
Water Flow Rate	0.065 L/s per kW	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS
Water-Side Fouling Factor	0.000044 m²°C/W	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS
Evaporator:												
Entering Temperature	NS	12	12	0	12	0	12	12	12	0	0	0
(for floor heating or similar application)	-	**23**	**23**	-	-	-	-	-	-	-	-	-
Leaving Temperature	6.7°C	7	7	-5	7	-5	7	7	7	-5	-5	-5
(for floor heating or similar application)	-	**18**	**18**	-	-	-	-	-	-	-	-	-
Water Flow Rate	0.043 L/s per kW	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS
Water-Side Fouling Factor	0.000018 m²°C/W	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS
Environmental Conditions:												
Dry Bulb Temperature	-	15°C to 30°C			15°C to 30°C							
Electrical Power Supply	-	NS			NS							

Legend
W/W= Water to Water
B/W = Brine to Water
W/B = Water to Brine
A/W =Air to Water
A/B =Air to Brine
NS = Not Stated

Analyzing Table 3-3, it's possible to observe that the rating conditions are very similar among them since they depend only on the specific unit's purpose – i.e. evaporator temperature - and on the used technology, and not on a particular standard. Figure 3-1 shows the temperature levels prescribed in the considered standards, remarking this aspect. Another aspect which can be noted is that, while for the condenser just the inlet temperature is prescribed, for the evaporator also the ΔT and, therefore, the outlet temperature are specified. In particular, the temperature levels at the evaporator are: -5°C÷0°C; 7°C÷12°C; 18°C÷23°C. The first one is for industrial purposes –e.g. food storing-; and the other two ones are for air conditioning applications using fan coils and radiant floors respectively. Since the technology of chilled water distribution systems developed, for the air conditioning applications, another temperature levels –i.e. 15°C÷18°C- related to the use of cooling ceiling could be insert. Regarding the condenser instead, four different temperatures levels are given: 27°C; 30°C; 35°C; 45°C which depend on the used heat rejection system and heat transfer medium. Nevertheless, because in the present work, air and brine as heat transfer medium at the condenser are not considered as well as direct fired sorption chillers working at low evaporator temperature, 45°C and 5°C÷0°C as temperature levels at the condenser and evaporator respectively are left.

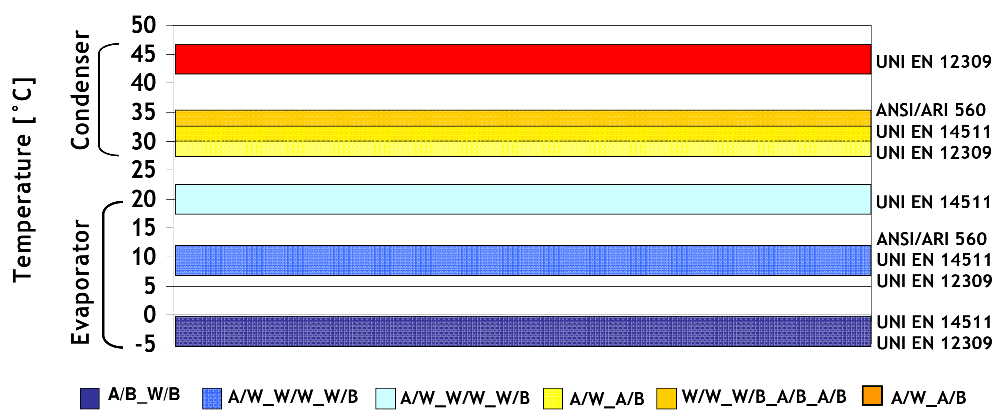

Figure 3-1: Temperature levels of Standard Rating Conditions - Cooling mode

Regarding the specific flow rate per cooling and/or chilling power and the water-side fouling factor of the condenser and evaporator, with the exception of standard

ARI/ANSI 560, any reference is done by standards. While, for the environmental conditions, all standard prescribe to perform the test at a dry bulb temperature varying between 15°C and 30°C.

3.2.2 Test requirements

Test requirements are the minimum requirements, prescribed by norms, necessary to carry out capacity tests in a standardized way. They concern the "quality" and the "typology" of the equipment used for the measurements as well as the "quality" of the regulation system – i.e. the capability to maintain certain conditions during the test execution –, features from which the uncertainty of measurements depends on. Furthermore, they specify the data to be recorded over the data collection period. Regarding the quality of measuring instruments, it is expressed in terms of uncertainty of measurement. Table 3-4 shows, for each measured quantity of interest, the values that the related uncertainty shall not exceed. These values pertain only to UNI EN 12309 and UNI EN 14511, since all UNI Standards refer to this last one, and the Standard ARI/ANSI 560 refers to American ASHRAE Standards 30 for measuring instruments.

Table 3-4 Uncertainties of measurement for the indicated quantities prescribed in Standard UNI EN 12309 and Standard UNI EN 14511

MEASURED QUANTITY (*)	UNIT	UNCERTAINTY OF MEASUREMENT	
		UNI EN 14511-3	UNI EN 12309-2
Liquid			
Temperature Inlet/Outlet	°C	± 0.1 K	± 0.3 K
Temperature Difference	°C	-	± 0.1 K
Volume Flow	m^3/s	± 1 %	± 1 %
Static Pressure Difference	Pa	± 5 Pa (Δp ≤ 100 Pa)	± 5 Pa (Δp ≤ 100 Pa)
		± 5 % (Δp > 100 Pa)	± 5 % (Δp > 100 Pa)
Air			
Dry Bulb Temperature	°C	± 0.2 K	± 0.2 K
Wet Bulb Temperature	°C	± 0.3 K	± 0.2 K
Volume Flow	m^3/s	± 5 %	± 5 %

Static Pressure Difference	Pa	± 5 Pa (Δp ≤ 100 Pa) ± 5 % (Δp > 100 Pa)	± 5 Pa (Δp ≤ 100 Pa) ± 5 % (Δp > 100 Pa)
Concentration			
Heat transfer medium	%	± 2 %	± 2 %
Electrical Quantities			
Electric Power	W	± 1 %	-
Voltage	V	± 0.5 %	-
Current	A	± 0.5 %	-
Electrical Energy	kWh	± 1 %	-
Thermal Power (**)	kW	-	± 2 %
Chilling Capacity (***)	W	≤ 5 %	≤ 5 %

(*) *The uncertainty values related to refrigerant and compressor rotational speed have been left since they don't concern this study.*

(**) *Thermal Power is the input power at sorption chiller generator*

(***) *The uncertainty of 5 % concern the use of liquid enthalpy method for the calculation of chilling capacity - as sorption chillers leave out the use of air as heat transfer medium at the evaporator -. It is independent of the individual uncertainties of measurement including the uncertainties on the properties fluids*

As it's possible to observe, the uncertainty values written for temperatures, pressure differences, mass flows etc. are practically the same for both standards. This means that, for their measuring, the same category of instruments is prescribed – e.g. PT100, Class A, 4-wires for temperature measurements, etc….-. The unique exceptions concern electrical quantities and thermal power for which there's no accordance since they depend strictly on the specific standard purpose. Nevertheless, some of these values, like electric power, electrical energy and thermal power uncertainties, can be used to establish the category of instruments employed for sorption chillers' electric power and thermal power calculation. Regarding the data to be recorded, the standards identify the general information required for the final marking without putting any limit to those that can be obtained. Such data are the mean values taken over the data collection period, with the exception of time measurement. In the Table 3-5, the meaningful quantities to be recorded for sorption chillers have been selected and listed.

Table 3-5 Data to be recorded during the tests execution

QUANTITY TO BE RECORDED	UNIT
Ambient Conditions	
Air Temperature, Dry Bulb	°C
Atmospheric Pressure	kPa
Electrical Quantities	
Voltage	V
Electric Power	W
Thermodynamic Quantities	
Inlet Temperature Evaporator	°C
Outlet Temperature Evaporator	°C
Inlet Temperature Absorber	°C
Outlet Temperature Condenser	°C
Inlet Temperature Generator (*Steam Supply Temperature)	°C
Outlet Temperature Generator (*Steam Condensate Temperature)	°C
Pressure Drops Evaporator	°kPa
Pressure Drops Absorber/Condenser	°kPa
Pressure Drops Generator (*Steam Supply Pressure)	°kPa
Volume Flow Evaporator	m^3/s
Volume Flow Absorber/Condenser	m^3/s
Volume Flow Generator (*Oil/Gas Consumption)	m^3/s
Heat Recovery Heat Exchanger (**)	
Inlet Temperature	°C
Outlet Temperature	°C
Volume Flow	m^3/s
Pressure Drops	°kPa
Heat Transfer Medium	m^3/s
Concentration	%
Density	Kg/m^3
Specific Heat	J/kg·K
Chilling Capacity	W
Ratios	
EER (COP)	W/W
Generic Data	
Nameplate, Date	-
Data Collection Period	sec

(*) *Quantity which refers to direct driven chillers*

(**) *If chilled water or absorber/condenser water (or steam condensate) is used for some other incidental function within the chiller.*

Before recording the data, it's required that chillers are under stationary conditions, which are considered being reached when, for a minimum duration of 1 hour, all observed quantities vary in a certain range around the set values – i.e. the rating conditions - without changing the settings. This means that periodic fluctuations of measured quantities caused by the operation of regulation and control devices are permissible, on condition that the mean value of such fluctuations does not exceed the permissible deviations which are written in Table 3-6.

Table 3-6 Permissible deviations of measured quantities from the set values

MEASURED QUANTITY	Permissible deviation of the arithmetic mean values from set values	Permissible deviation of individual measured values from set values
Inlet Temperature	± 0,2 K	± 0,5 K
Outlet Temperature	± 0,3 K	± 0,6 K
Volume Flow	± 2 %	± 5 %
Static Pressure Difference	--	± 10 %
Voltage	± 4 %	± 4%

Observing the typical behavior of continuous, semi- continuous and batch chillers shown in Figure 3-2, Figure 3-3 and Figure 3-4, it's easy to understand that the applicability of these conditions is restricted to the only case of continuous chillers. In fact, the swaps determine, during the cyclic working of semi-continuous and batch chillers, rough fluctuations of inlet and outlet temperatures as well as of volume flows and static pressures which make impossible to maintain these quantities or their average in the prescribed ranges. It follows the necessity to define new restrictions for stationary conditions.

Concerning the test duration and the sample time for recording data, the prescriptions dictated by the considered standards are quite different. The standard UNI EN 14511 recommends, once that the stationary conditions are reached and maintained, to record all meaningful data continuously and to test the machine for a period longer than 35min.

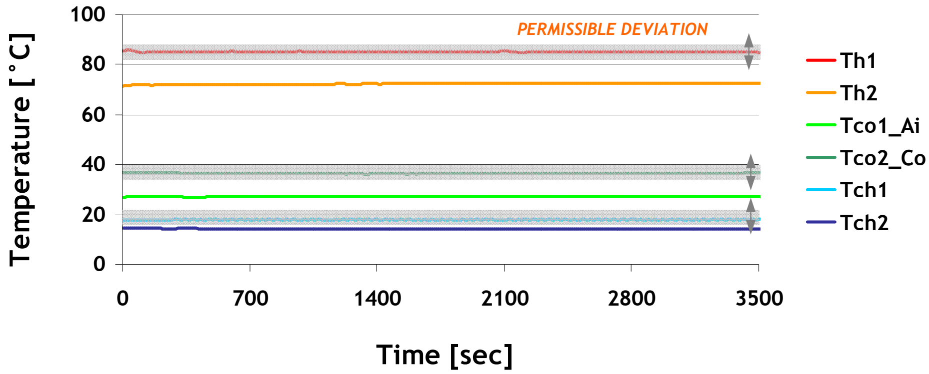

Figure 3-2 Temperature profile of Continuous chiller - Source: TU Berlin

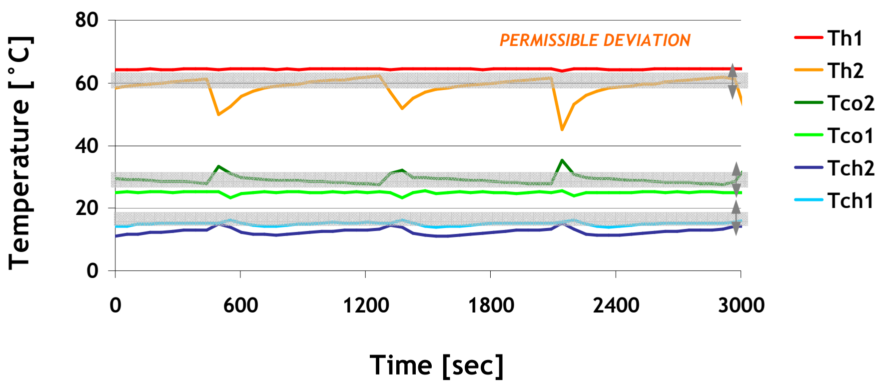

Figure 3-3 Temperature profile of Semi-continuous chiller - Source: LESBAT

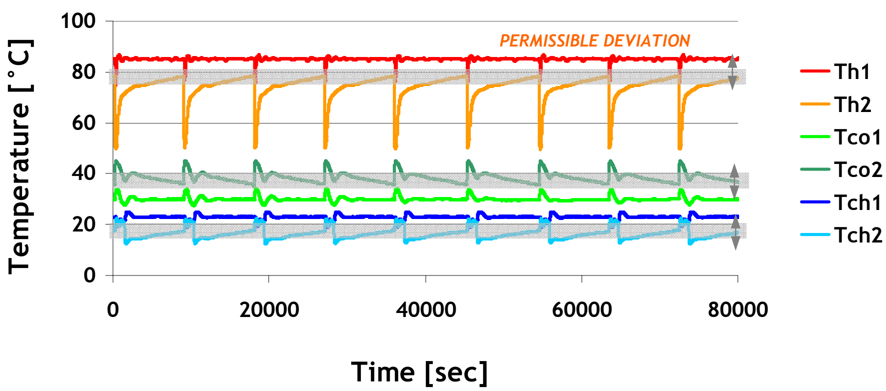

Figure 3-4 Temperature profile of Batch mode chiller - Source: EURAC

It specifies the sample time – i.e. not larger than 30 s - only if recording instruments operating on cycle basis are used??. While the standard UNI EN 12309 recommends recording data continuously, but with a sample time not larger than 2 minutes in case of cyclic recording instruments, and testing the machine for a period longer than 30 min. The standard ANSI/ARI 560, prescribes, always after the stationary conditions have been established, to record data at minimum of 5 minute intervals and, to minimize the effects of transient conditions; it prescribes to reduce these intervals in order to take data as simultaneously as possible. Any prescription is given for the test duration.

From this brief analysis, it emerges that even if the considered standards don't give any real restrictions as regards test duration and sample time with exception of some lower and upper limitations; on the other hand, if any specific indication about these two parameters is not given, wrong tests could be executed. In fact, it could happen that, according to all prescriptions, a manufacturer of semi-continuous or batch mode chillers tests the machines only for 35 min regardless the cycle time of their products and using 30 s or some minutes as sample time. This not only does not take into account the cyclic nature of these chillers but doesn't consider also the transitory effects following swaps.

3.2.3 Procedure for Chilling Capacity and COP calculation

The last aspect to analyze is the methodology used in the standards for the calculation of the rating chilling capacity and the COP. For the determination of the chilling power, the following equation is employed:

Equation 3-1 $$\dot{Q}_{ch} = \dot{v} \times \rho \times c_p \Delta T$$

Where:

Q_{ch} is the chilling capacity, in Watts

$\dot{V}$ is the volume flow rate, in cubic meters per second

ρ is the density, in kilograms per cubic meter

C_p is the specific heat at constant pressure, in joule per kilograms and Kelvin

ΔT is the difference between inlet and outlet temperatures, in Kelvin

Such formula calculates the rated powers as instantaneous values. If this can suit continuous chillers (even if it is always preferable to use an average power on the test duration) it cannot be accepted in case of semi-continuous or batch mode chillers, which are based on cyclic working.

In fact, for this kind of chillers, the instantaneous power varies from a maximum at the beginning of the cycle in which there is the maximum ΔT to a minimum at end of the cycle in which there is the minimum ΔT. So, only the average shall be taken into account as a performance figure. Analogue considerations could be stated for the COP: an instantaneous value is calculated through the following formula:

Equation 3-2

$$COP = \frac{\dot{Q}_{ch}}{\dot{Q}_{h}}$$

In this case, it could happen that, for a certain period, the machine (in particular the batch mode one) absorbs heat without delivering any power. In this case the COP is null. For this reason the exchanged energies during the test or the mean powers shall be used for its calculation.

3.3 Limits and Conclusion

From the analysis carried out on the existing standards concerning test procedures for chillers and heat pumps, it has emerged that, although in the last years the sorption chillers' market has recoded an expansion especially with the introduction of small scale chillers with different working principles, the normative scenario didn't follow this growth. In fact, the few standards that apply to thermally driven chillers – most of them refer to electrically driven chillers – distinguish these machines only on the base of:

1. effect number – usually single or double effect-;
2. driven technology – direct or indirect fired
3. Physical properties of the sorbent – absorbent(liquid) or adsorbent(solid)-

without taking into consideration the eventual cyclical behaviour of the chillers. From here the necessity to find a new way to classify the sorption chillers and on the base of this, a new test procedure should be developed

4 Development of Test Procedures

From the analysis carried out in the previous chapter, it has emerged that the standards placed at disposal of manufacturers and certification bodies by normative system, concerning test procedures and performance evaluation methods of chillers and heat pumps, are inadequate for many of sorption chiller prototypes currently present on the market. Such inadequacy is mainly due to the fact that the existing standards treat sorption chillers as if they were all continuous machines without taking into account the real nature of their operation. For this reason, in present work, new test procedures based on sorption chillers working modes have been developed and, in the current chapter, presented. In this way, the applicability of their prescriptions is guaranteed.

Although in the present work – i.e. *Chapter 5 and Chapter 6* - such test procedures have been validated only on small-scale sorption chillers prototypes, it has to be précised that they are applicable also to medium and large scale sorption chillers as well as to thermally driven heat pumps for which, however, different rating conditions and COP shall be defined.

Since the standards dealing with this topic have the same structure, it has been though to employ it also for the developing of these new test procedures – see Figure 4-1.

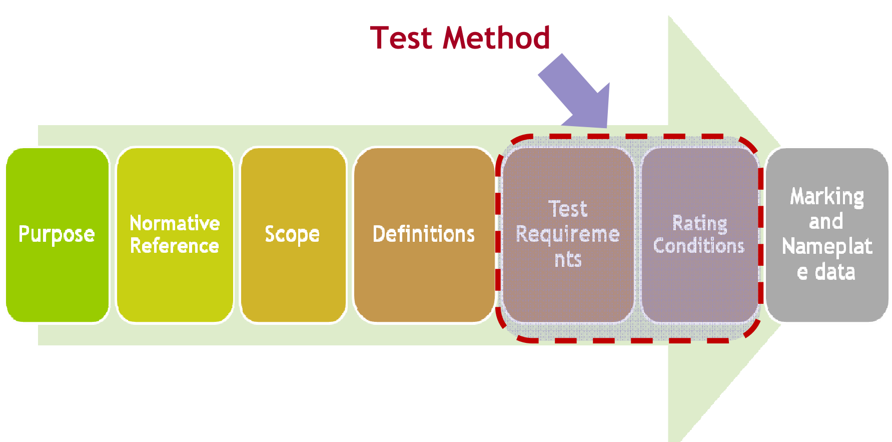

Figure 4-1 Typical structure of standards concerning test procedures and performance evaluation methods for chillers and heat pump

The advantage of using this approach lies in the fact that the procedures are developed step by step making easier the analysis and the definition of prescriptions for each working mode. Furthermore, a procedure developed in such a way might favour its feature standardization.

For the development of this test procedure, greater importance is given to definition of rating conditions and test requirements – as it can view from Figure 4-1 - since they represent the most crucial points of the whole procedure. Nevertheless, even if briefly, purpose, scope, normative references and definitions are specified too.

4.1 Purpose and Scope

The aim of this test procedure is to develop a common methodology for testing and evaluating the performance of existing sorption chillers at design and off-design conditions with the intent, on one hand, to mark the unit; on the other hand, to supply professional associations as well as final customers with reliable performance figures. In order to do this, test requirements, rating conditions as well as the minimum data to publish and to use for the unit marking have been individuated and, here, specified. A fundamental aspect to underline is that such procedure, like most of those focused in testing devices, prescribes to investigate the chillers only under stationary conditions. This means that effects of interactions between machine and external factors, like variable loads, variable climatic conditions, seasons and variable desorber inlet temperatures, are here left out. Although this might seem a limiting factor, actually it represents the starting point for the definition of transient and partial-loads tests since it defines the approach to use depending on the chiller working modes as well as the method for the performance evaluation - i.e. COP calculation. An example of this is represented by the *Integrated Part-Load Value* (IPLV) – i.e. the chiller's performance over a range of operating conditions – which is calculated as linear combination of coefficients of performance calculated at different percentages of chiller's full load.

Equation 4-1

$$IPLV = a \cdot COP_{100\%} + b \cdot COP_{75\%} + c \cdot COP_{50\%} + d \cdot COP_{25\%}$$

From the

Equation 4-1, it can be deduced that, regardless of how the part-loads are obtained, once they have been reached, for calculating the COP at different conditions the developed procedure shall be used.

Regarding the *Scope*, the procedure applies to all sorption chillers prototypes: direct and indirect fired; single, double or more effects; continuous, semi-continuous and batch mode; absorption and adsorption. Obviously, depending on the specific case, the equations for energies, powers and performances calculations have to be adapted, but always using as control volume that including the whole machine. So what happens within the chiller is not taken into account.

4.2 Normative References and Definitions

To develop such test procedure, the standards carried below have been used. They concern the methodologies for testing conventional and sorption chillers and represent a valid reference for the development of test procedure. In particular, they have been used for the definition of rating conditions, test requirements – i.e. measurement equipment, stationary conditions etc… - and evaluation methods. Some of their provisions have been included in the procedure and cited at the appropriate places in the present chapter.

In detail, the considered standards are:

- EN 14511-1:2004, Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling - Part 1: Terms and definitions.
- EN 14511-2:2004, Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling - Part 2: Test conditions.
- EN 14511-3:2004, Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling - Part 3: Test methods.
- EN 14511-4:2004, Air conditioners, liquid chilling packages and heat pumps with electrically driven compressors for space heating and cooling - Part 4: Requirements.
- ANSI/ARI Standard 560, 2000 Standard for Absorption Water Chilling And Water Heating Packages

Concerning the definitions of working modes of sorption chillers and their typology, the definitions given in the Chapter 2 are used. All the other definitions, concerning figures used in the procedure, are given in the present chapter.

4.3 Test requirements

Test requirements dictate the necessary conditions to perform tests under stationary conditions in a standardized way. They concern both the utilized methodology during the test execution and the instrumentation employed for carrying out measurements. Below the test requirements have been specified and differentiated, where it was necessary, on basis of chillers' working mode.

4.3.1 Equipment

4.3.1.1 Typology and Category

The choice of sensors to use for the test execution depends substantially on two factors: type of measurements necessary to determine of the quantities involved in the performance calculation – i.e. COP -; and the maximum permissible uncertainty associated with performance evaluation. In this test procedure, since a black box approach is used – i.e. the chiller is probed only externally -, the quantities to be measured are only the temperatures, static pressure differences, mass flows of the mediums with which the machine exchanges heat with external sinks/sources and the electric power and electric consumption employed for the electric performance evaluation. Thus, the typology of sensors has been chosen in order to guarantee the measurements of these quantities.

Concerning the category instead, it has been chosen so as to have 7% as maximum expanded uncertainty of COP obtained applying the error propagation law and a coverage factor equal to 2. Since the uncertainty of COP is given by the not linear combination of uncertainties type B, relative to instruments used for measurements, and uncertainties type A, relative to fluctuations of the measured values, the category of sensors is meant as uncertainty type B of measurements carried out with those specific sensors. For the choice of such uncertainty values, standards UNI EN 14511-3 and UNI EN 12309-2 have been considered. In particular, those specified for UNI EN 14511-3 have been used for the present test procedure since they guarantee a maximum expanded uncertainty of COP equal to 7%. In Table

4-1, the uncertainties prescribed by the present test procedure are listed and refer to UNI EN 14511-3.

It has to be précised that the measurement equipment is equal for all chillers, independently from their working mode.

Table 4-1 Uncertainties of measurement for indicated values

MEASURED QUANTITY (*)	UNIT	UNCERTAINTY OF MEASUREMENT
		UNI EN 14511-3
Temperature Inlet/Outlet	°C	± 0.1 K
Temperature Difference	°C	-
Volume Flow	m^3/s	± 1 %
Static Pressure Difference	Pa	± 5 Pa (Δp ≤ 100 Pa)
		± 5 % (Δp > 100 Pa)
Air		
Dry Bulb Temperature	°C	± 0.2 K
Wet Bulb Temperature	°C	± 0.3 K
Volume Flow	m^3/s	± 5 %
Static Pressure Difference	Pa	± 5 Pa (Δp ≤ 100 Pa)
		± 5 % (Δp > 100 Pa)
Concentration		
Heat transfer medium	%	± 2 %
Electrical Quantities		
Electric Power	W	± 1 %
Electrical Energy	kWh	± 1 %
Thermal Power (**)	kW	-
Chilling Capacity (***)	W	≤ 5 %

4.3.1.2 Placement

Regarding the placement of sensors, they shall be installed in such a way that the measurements obtained are directly attributable to the machine. This means that they shall be placed at inlets and outlets of the machines without any other components installed between them and the machine itself.

4.3.2 Stationary Conditions

The definition of *Stationary Conditions* – i.e. of the boundaries within which the machine operation is considered stationary – represents one of the most crucial points of the whole procedure since, right in it, the main cause of inapplicability of current standards lies.

The reason for which such conditions are characterized is for supplying the tests executor with a tool able to establish if the fluctuations of the main quantities representing the machine, recorded during a test, are attributable to the operation of the machine itself or to a bad test execution.

Right on the origins of these fluctuations, the reasoning used for the determination of new stationary conditions is based. In particular, the operation of a chiller working in batch mode has been analyzed, since it represent the most complicate and, at the same time, the most general case – see Figure 4-2. Nevertheless, it is necessary to precise that such figure doesn't show the pure batch operation –i.e. when only a unit, consisting of desorber and condenser, performs the whole thermodynamic cycle – but an operation more close to a semi-continuous one.

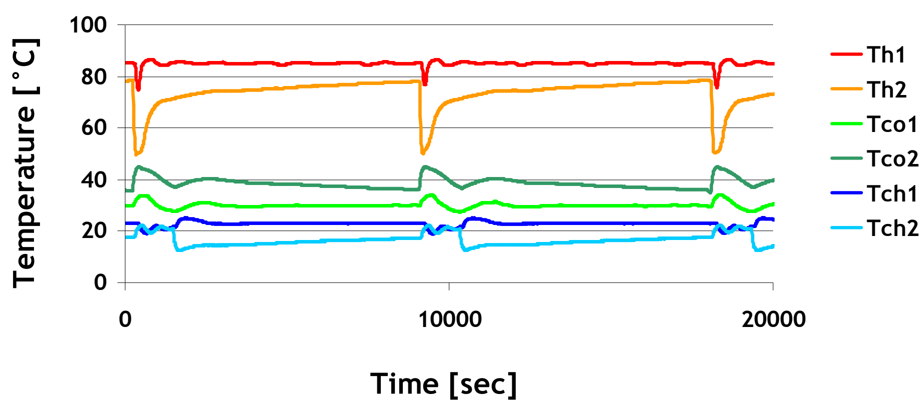

Figure 4-2 Temperature profiles vs. time of CW10 (ClimateWell) absorption chiller during its operation - Source: EURAC

Observing Figure 4-2, it is possible to note that, whenever a swap occurs, the chiller's characteristic quantities – in this case, the inlet and outlet temperatures - present sharp fluctuations. This is due to the fact that the swapping units, changing their working conditions, need of time for reaching the equilibrium at new conditions. Nevertheless, just this recurrence demonstrates that such fluctuations are attributable mainly to the specific operation of the chiller and only minimally to the regulation system.

Another aspect that can be observed concerns the trends of temperatures after the swapping time - i.e. the period, following a swap, during which the chiller doesn't deliver any chilling power. In particular, it's possible to observe that inlet temperatures, after this period, are quite stable around their own set value till the next swap, with only exception of the inlet temperature at desorber which starts to be stationary just after the swap. While the outlet temperatures show, depending on the specific circuit, increasing or decreasing monotonic trends as function of solution desorption and absorption indexes. In detail, the outlet temperatures at the desorber and at the evaporator increase as the rate of desorption and absorption process decrease; while the outlet temperatures at the condenser/absorber decrease. This leads to not have a constant behaviour of such temperatures around a certain value during the chiller operation. These last observations allow deducing that, while for inlet temperatures, after a certain mean time, the fluctuations depend only on the system regulation, for the outlet temperatures fluctuations and trends depend strongly on desorption and absorption processes and so, on the machine. The same kind of reasoning has been done for the other characteristic quantities of chiller, which are: pressures, static pressure differences and flow rates, for which it is possible to observe behaviours like those of inlet temperatures.

On the base of the analysis just carried out, the data collected during each operational cycle have been subdivided in two sub-intervals: *Sub-Interval A,* consisting of all data collected during each cycle time, i.e. the mean time between two consecutive swaps; and *Sub-Interval B,* consisting of data collected during each cycle time with the exception of those relative to the swapping period. The choice of such subdivision is based on the nature of the fluctuations before analyzed. More precisely, the *Sub-Interval B* contains all those data whose deviations from the set value are only due to regulation system; the *Sub-interval A* contains all data whose fluctuations are due to both the chiller operation, since it includes also the swapping time, and to regulation system.

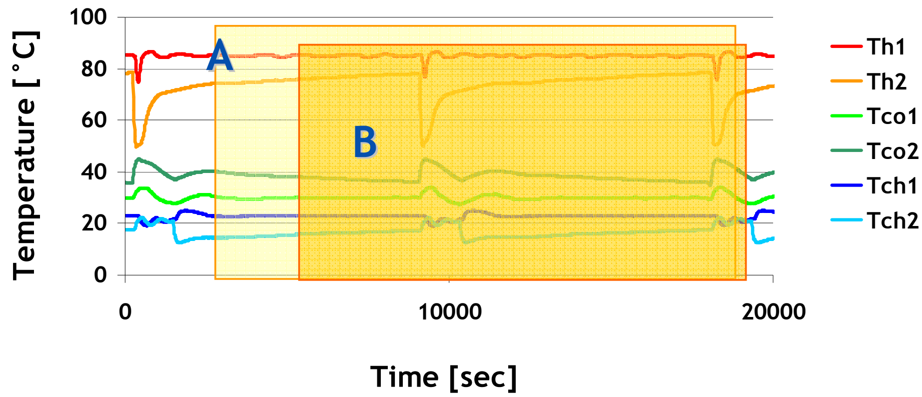

Figure 4-3 Sub-Intervals employed in the new stationary conditions defenition

Even if the whole analysis, included the determination of the two subintervals, has been carried out on a chiller working in batch mode, the results can be extended to all working modes, with opportune precisions obviously, and used for the determination of new stationary conditions.

Below, the permissible deviations which state stationary conditions are defined on the base of above-explained sub-intervals and differentiated for all working modes.

4.3.3 Continuous Chiller

The definition of stationary conditions for continuous chillers represents, perhaps, the most simple case since all existing standards prescribes test procedures and so the permissible deviations from the set values, only for these kind of chillers even if with some imprecision.

Nevertheless, using for continuous chillers the approach before explained, it is possible to say that, once reached set conditions, the fluctuations of their main features are essentially due to the regulation system. So, imagining such chillers as they were characterized by a single-cycle operation whose duration coincides with that for establishment of stationary conditions and whose swapping time is practically equal to zero, for them the two subintervals *A* and *B* coincide - as shown in figure - and are right equal to the whole cycle. This means that the permissible deviations are defined indifferently for both subintervals.

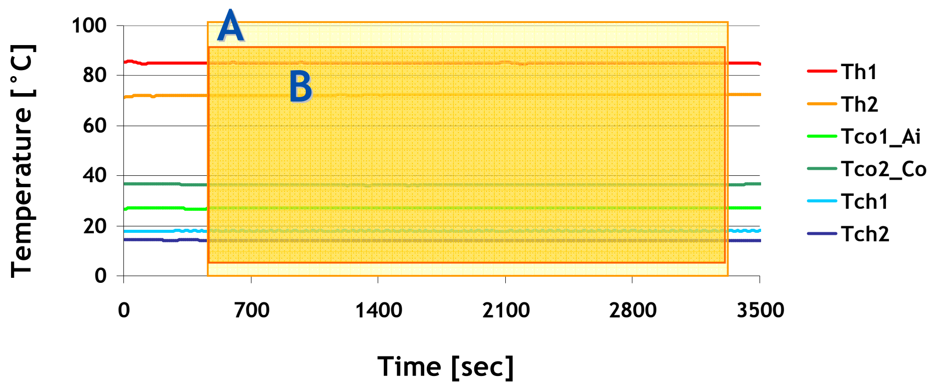

Figure 4-4 Temperatures profiles vs. time of a Continuous sorption chiller during its operation - Source: TU Berlin - and Sub-intervals A and B for stationary conditions definition

At this point it's possible to state that the stationary conditions for a continuous chiller are considered obtained and maintained when all the measured characteristic quantity that in this case are:

- Inlet Temperatures
- Outlet Temperatures
- Static Pressure Differences
- Volume Flows

remain constant without having to alter the set values for a minimum duration of:

- one hour

with respect of tolerances given in Table 4-2. Periodic fluctuation due to the regulation and control system are admissible but under condition that the mean values don't exceed the tolerances given in Table 4-2.

Table 4-2 Stationary boundary conditions for a Continuous chiller

Measure Quantity	*Permissible deviation of individual measured values from set values*	*Permissible deviation of individual measured values from set values*
Inlet Temperature	±0.2K	±0.5K
Outlet Temperature	±0.3K	±0.6K
Mass Flow	±2%	±5%
Static Pressure Difference	-	±10%

To be noted that for continuous chillers, the outlet temperatures are considered since they not depend on the working of the machine.

4.3.4 Semi-Continuous Chiller

The definition of stationary conditions for semi-continuous chillers is more complex since these chillers, like those working in batch mode, are characterized by a cyclic operation. Figure 4-5 shows how the presence of swaps determines sharp fluctuations of measured quantities especially just after the swap. Furthermore, it possible to observe that, also in this case, with the exception of outlet temperatures, the main chiller features reach, after a certain time, an equilibrium condition around their set values till the next swap. At this point, it is possible to individuate the two subintervals A and B that here are distinct. *Subinterval A* coincides with a single cycle and contains all data collected during it whose fluctuations are partially due to the machine operation and partially due to the regulation system; while *Subinterval B* contains the data colleted between the end of a swapping time of a certain cycle and the following swap, whose fluctuations are only due to the regulation system.

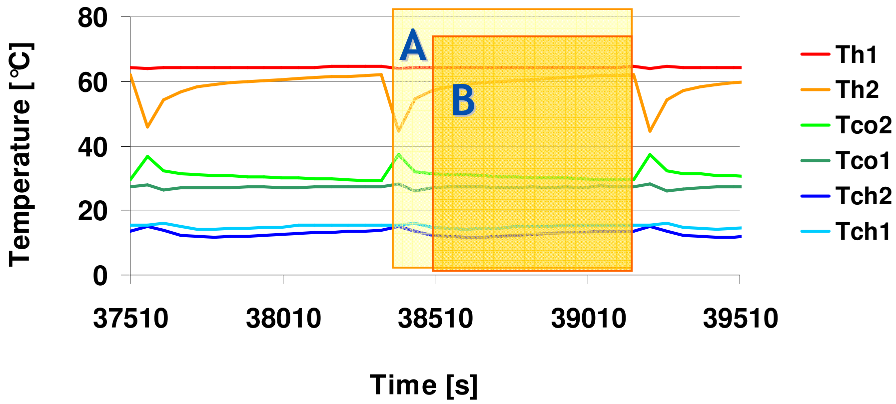

Figure 4-5 Temperatures profiles vs. time of a Semi-continuous sorption chiller during its operation - Source: LESBAT - and Sub-intervals *A* and *B* for stationary conditions definition

The stationary conditions are defined, with the help of the two subintervals A and B, for the following quantities:

- Cycle Time
- Inlet Temperatures
- Static Pressure Differences
- Mass Flows

Since such conditions lay down restrictions to the fluctuations of considered quantities independent from the specific machine operation, they are characterized only on subinterval B, with the exception of *Cycle Time* and the *Inlet Temperature* at desorber.

Regarding the *Cycle Time*, in order to achieve the stationary conditions, it shall be such that its standard deviation calculated on the analyzed cycles is lower than 10%, value obtained by the experience.

Concerning the *Inlet Temperature* at desorber instead, since its behaviour is not influenced by the machine operation with the exception of fluctuations caused by swaps, the rapidity with which such temperature goes back to its set value just after a swap depend only on the efficiency of control and regulation system. In this regard, it has been formulated a condition which establishes that such temperature has to reach its set value so rapidly that the ratio of coefficient of performance calculated using the real inlet temperature at desorber divided by the coefficient of performance calculated using the set temperature value shall be higher than 0.95. From a mathematic point of view, such condition is formulated as follows:

Equation 4-2 $$\frac{COP_{real}}{COP_{set}} \geq 0.95$$

Where:

Equation 4-3 $$COP_{real} = \frac{\dot{m}_{ch} \cdot c_{p,ch} \cdot \Delta T_{ch,real}}{\dot{m}_{h} \cdot c_{p,h} \cdot \Delta T_{h,real}}$$

With:

Equation 4-4 $$\Delta T_{h,real} = T_{h,in,real} - T_{h,out,real}$$

And:

Equation 4-5 $$COP_{set} = \frac{\dot{m}_{ch} \cdot c_{p,ch} \cdot \Delta T_{ch,real}}{\dot{m}_{h} \cdot c_{p,h} \cdot \Delta T_{h,set}}$$

With:

Equation 4-6 $$\Delta T_{h,sel} = T_{h,in,set} - T_{h,out,real}$$

By substituting Equation 4-3 and Equation 4-5 into Equation 4-2, this last becomes:

Equation 4-7 $$\frac{COP_{real}}{COP_{set}} = \frac{\Delta T_{h,real}}{\Delta T_{h,set}} = \frac{T_{h,in,real} - T_{h,out,real}}{T_{h,in,setl} - T_{h,out,real}} \geq 0.95$$

Equation 4-7 is the extended formulas of this condition.

It is necessary to precise that the *Inlet Temperature* at desorber, besides to observe the condition specified by Equation 4-7, like all the other inlet temperatures, it shall remain constant in the subinterval *B* according to the tolerances prescribed in Table 4-3.

At this point it's possible to state that the stationary conditions for a *Semi-continuous* chiller are considered obtained and maintained when all the measured characteristic quantities which are:

- Cycle Time
- Inlet Temperatures
- Static Pressure Differences
- Mass Flows

remain constant without having to alter the set values for a minimum duration of:

- four cycles

with respect of tolerances given Table 4-3. Periodic fluctuation due to the regulation and control system are admissible but under condition that the mean values don't exceed the tolerances given in Table 4-3.

Table 4-3 Stationary boundary conditions for a Sami-continuous chiller

Measure Quantity		***Permissible deviation of individual measured values from set values***		***Permissible deviation of individual measured values from set values***	
		A	***B***	***A***	***B***
Generator	Inlet Temperature	COP>95%	±0.2K	COP>95%	±0.5K
	Mass Flow	COP>95%	±2%	COP>95%	±5%
	Static Pressure Diff.		-		±10%
Condenser/ Evaporator	Inlet Temperature		±0.2K		±0.5K
	Mass Flow		±2%		±5%
	Static Pressure Diff.		-		±10%
Cycle Time		±10%			

The choice to establish the stationary conditions on four cycles is due to two reasons. First, in this way each unit constituting the chiller performs two entire cycles; second, repeating four cycles for every working condition it is possible to slake the influence of the former working condition on the present.

Concerning the permissible deviations, the values used in the subinterval B are equal to those used for the continuous chillers, which refer to UNI EN 14511. This choice is dictated by fact that, in this subinterval, it is supposed that all measured quantities have reached the equilibrium around their set conditions and that the chiller behaves like a continuous one.

4.3.5 Batch Mode Chiller

Regarding the definition of stationary conditions for batch mode chillers, since they are often being let to work in semi-continuous mode the conditions specified in the previous paragraph can be applied identically to them. It is different instead, when the chiller works in pure batch mode, i.e. when a single unit performs the whole thermodynamic cycle, for which it is necessary to do some clarifications.

In this case, in fact, the unit for performing a full thermodynamic cycle needs of two operational cycles, i.e. two cycles during which it executes completely the desorption and sorption processes as shown in Figure 4-6. In semi-continuous chillers, these two operational cycles are usually distinguished by a swap between units. Thus, to establish if the chiller works under stationary conditions, the prescriptions concerning permissible deviations shall be verified on both working phases and on the involved characteristic quantities. In particular, during the desorption phase, the stationary conditions shall be verified for:

- Inlet and Outlet temperatures
- Static Pressure Differences
- Mass Flows

at desorber and condenser on only subinterval A – i.e. the interval consisting of all data collected during the desorption process and coinciding with the operational cycle - since subinterval B cannot be individuated in this phase. It can be defined as follows:

- Subinterval A consists of all data collected during the desorption process and coincides with the operational cycle

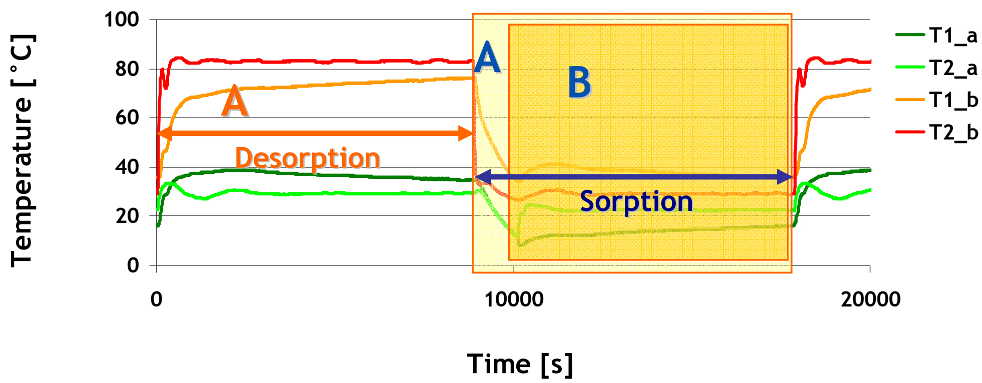

Figure 4-6 Temperatures profiles vs. time of a batch mode sorption chiller during its operation – Source: EURAC - and Sub-intervals A and B for stationary conditions definition

Concerning the stationary conditions during the sorption phase, they shall be verified for:

- Inlet and Outlet temperatures
- Static Pressure Differences
- Mass Flows

at evaporator and absorber on the subinterval A and B that here are defined as:

- Subinterval A consists of all data collected during the sorption process and coincides with the operational cycle
- Subinterval B consists of all data collected during the sorption process with the exception of those collected during the period during which any chilling power is produced. It coincides with the swapping time for those working in semi-continuous mode.

Anything can be say concerning the cycle time.

The restrictions used to establish if the chiller works under stationary conditions shall be applied on the two operational cycles and on the relative characteristic quantities involved.

Table 4-4 Stationary boundary conditions for chiller working in pure batch-mode

Measure Quantity		*Permissible deviation of individual measured values from set values*		*Permissible deviation of individual measured values from set values*	
		A	***B***	***A***	***B***
Desorption Phase					
Generator	Inlet Temperature	COP>95%		COP>95%	
	Mass Flow	COP>95%		COP>95%	
	Static Pressure Diff.				
Condenser	Inlet Temperature	COP>95%		COP>95%	
	Mass Flow	COP>95%		COP>95%	
	Static Pressure Diff.	-			
Sorption Phase					
Condenser/ Evaporator	Inlet Temperature		±0.2K		±0.5K
	Mass Flow		±2%		±5%
	Static Pressure Diff.		-		±10%

4.4 Rating Conditions

The rating conditions are those conditions at which the chiller has to be tested and evaluated for the final marking. As explained in Chapter 3, they depend only on technological limits of the systems usually coupled with this kind of machines to reject heat at medium temperature and to distribute the cold produced. This means that they are completely independent from the specific working modes, and therefore, they are defined in the same way for all chillers types.

For the definition of rating conditions, the following technologies have been considered as heat rejection systems:

- *Wet Cooling Tower*, which uses the evaporation of water to remove the process heat from the chiller and to cool the working fluid to near the wet-bulb air temperature;

- *Dry Air Cooler*, which uses air to remove process heat from the chiller and cools the working fluid to near the dry-bulb air temperature. Its efficiency is lower than that of Wet Cooling Tower;

- *Hybrid Air Cooler*, which represents a hybrid solution since it uses air to cool the working fluid to near the dry-bulb air temperature and sprays water to lower of some Celsius degrees the temperature of working fluid utilizing the latent heat of water.

The current literature developed on existing prototypes supplies as return temperatures from these systems, calculated for areas characterized by low *Relative Humidity,* the following values: 27°C for Wet Cooling Tower; 30°C for Hybrid Air Cooler and; 35°C for Dry Cooler. Such values have been used as rating conditions.

While as cold distribution systems, the following technologies have been considered:

- *Fan Coil*, which removes both latent heat and sensible heat from the room in which it is installed, providing also to the dehumidification;

- *Radiant floor*, which removes only sensible heat from the room in which it is installed without controlling the humidity;
- *Chilled Ceiling* that, like the radiant floor, can remove only sensible heat from the room in which it is installed without controlling the humidity.

For these systems, the current literature provides as design temperatures the following values: 7÷12°C for Fan Coils; 15÷18°C and 18÷23°C for Radiant Floor and Chilled Ceiling depending on the specific applications and on the dew point of the room in which they are installed. These values represent the inlet and outlet temperatures for each technology and they have been used as rating conditions too.

Table 4-5 summarizes the rating conditions so individuated.

Table 4-5 Rating Conditions written as inlet temperatures to the chiller

Driving Temperature	*Temperature Level*	*Evaporator*	*Condenser-Ab/Adsorber*
According to the manufactures specifications, to achieve 100% of full capacity	***Low Temperature***	**12°C** (7/12)	**27°C**
	Medium Temperature	**18°C** (15/18)	**30°C**
	High Temperature	**23°C** (18/23)	**35°C**

Analyzing Table 4-5, it is possible to note that only the inlet temperatures to the chiller are prescribed as rating conditions with the exception of that at desorber, for which, no value is provided. As explained in the Chapter 3, such temperature depends strongly on the working pairs used and on the specific driving technology chosen by manufacturer, for this reason it as well as volume flows and pressures shall be set according to the manufactures specifications for achieving 100% of full chilling capacity.

To be noted that fixing the inlet temperature at the evaporator, the outlet one is automatically determined by the technological limits of the distribution system employed. This means that for the evaporator also the outlet temperature is given as rating condition.

4.5 Test duration

Once the chiller has been installed and regulated for its normal operation, in accordance with the manufacturer's instructions, and the stationary conditions have been achieved, the tests can be performed.

In order to obtain a complete vision of chiller operation, which takes into account all meaningful information, it is necessary individuate adequate sample time and test duration.

For the choice of sample time, two aspects have been considered: first, the inertia of chillers, which is quite high; second, the minimum interval able to capture the smallest meaningful data and to minimize the effects of transient conditions. Among all sample times prescribed by the standards analyzed in Chapter 3, the most restrictive one has been chosen and it is equal to 10 seconds. It is the same for all working modes.

Concerning the test duration instead, the minimum time able to provide data representative, in faithful way, the chiller operation has been chosen. It differs for each working mode, and is equal:

- *0.5 hour* for Continuous Chiller

- 4 cycles for Semi-continuous Chiller

- 4 cycles for Batch mode Chiller

For Semi-continuous and Batch mode chillers, four cycles have been chosen as test duration because in these way the units constituting the chillers perform two entire thermodynamic cycle for each.

4.6 Rating Equations

To asses the chiller performances at specific rating conditions, general equations independent of working modes have been defined. They allow calculating the main quantities of interest as combination of punctual values calculated for each sample "*I*". Obviously, the data used for the evaluations are those collected during tests once stationary conditions have been obtained.

The assessment of chiller performances, in this test procedure, is carried out on two topics: thermal and electrical. Below the equations used for the calculations are written.

For the calculation of *Instantaneous Heat Rate* relative to sample "*I*", the following equation is used:

Equation 4-8

$$\dot{Q} = \dot{m}_i \cdot c_{p,i} \cdot \Delta T_i$$

Thus, the *Mean Heat Rate* can be obtained as mean of all instantaneous heat rates calculated for each cycle and for each sample carried out during the test.

Equation 4-9

$$\overline{\dot{Q}} = \frac{1}{n_{total,sample}} \cdot \sum_{j=1}^{n^\circ_{cycle}} \cdot \sum_{i=1}^{n^\circ_{sample/cycle,j}} \cdot \left(\dot{m}_{i,j} \cdot c_{p,i,j} \cdot \Delta T_{i,j}\right)$$

Obviously, for continuous chiller, the number of cycle is equal to 1 and the *Mean Heat Rate* coincides with the mean of instantaneous heat rates of the single cycles. The *Instantaneous Heat Rate* and the *Mean Heat Rate* are expressed in kW

The *Total Heat Transfer* is calculated as sum of trapezium areas whose basis are the heat rates calculated for the samples "*I*" and "*I+1*" and the height is the sample time – expressed in seconds - according to the following equation:

Equation 4-10
$$Q = \cdot \sum_{j=1}^{n^\circ_{cycle}} \cdot \sum_{i=1}^{n^\circ_{sample\,/\,cycle,j}} \cdot \left(\frac{\dot{Q}_{i+1,j} + \dot{Q}_{i,j}}{2} \right) \cdot \left(\frac{\Delta T_{i,j}}{3600} \right)$$

Equation 4-8, Equation 4-9 and Equation 4-10 allow calculating heating, cooling and chilling *Instantaneous Heat Rate*, *Mean Heat Rate* and *Total Heat Transfer*.

The thermal *COP* is calculated as the ratio of chilling capacity divided by the heating capacity – i.e. effective input to the chiller - expressed in kW/kW.

Equation 4-11
$$COP = \frac{\overline{\dot{Q}}_{ch}}{\overline{\dot{Q}}_{h}}$$

Concerning the electrical quantities, besides to estimate the chiller's input power for activating internal valves and for winning internal pressure drops, it's interesting to know which are the electrical consumptions and powers necessary for its operation. For this reason, the electrical powers required by external pumps for moving the heat transfer medium within the chiller as well as the performance indexes related to electric consumptions of systems used for heat rejection are here estimated.

For the calculation of *Instantaneous Electric Power* of the pump employed to win the pressure drops only due to machine, the following equation is used for each sampling "*I*":

Equation 4-12
$$\dot{Q}E_k = \frac{\Delta P_{i,k} \cdot \dot{V}_{i,k}}{\eta_{pump,k}}$$

Where:

- "*k*" indicates the specific circuit connected to the chiller, i.e. heating, cooling and chilling circuits;

- the $\Delta P_{i,k}$ is the *Static Pressure Difference* between the inlet and outlet of chiller relative to the circuit "*k*" and to the sample "*I*", measured in Pascal;
- η_{pump} is efficiency of pump necessary to win the pressure drops – i.e. ΔP_i - within the machine relative to the circuit "*k*".

This figure is expressed in Watt since its values are usually small.

Also in this case *Mean Electric Power* relative to circuit *"k"* can be obtained as mean of all instantaneous electric power according to the following equation:

Equation 4-13
$$\overline{\dot{QE}}_k = \frac{1}{n_{total,sample}} \cdot \sum_{j=1}^{n^\circ_{cycle}} \cdot \sum_{i=1}^{n^\circ_{sample/cycle,j}} \cdot \left(\frac{\Delta P_{i,k} \cdot \dot{V}_{i,k}}{\eta_{pump,k} \cdot 1000} \right)$$

It is expressed in kW.

While the *Electric Consumption* is calculated as sum of trapezium areas whose basis are the electric power calculated for the samples "*I*" and "*I+1*" and the height is the sample time – expressed in seconds - according to the following equation:

Equation 4-14
$$QE_k = \cdot \sum_{j=1}^{n^\circ_{cycle}} \cdot \sum_{i=1}^{n^\circ_{sample/cycle,j}} \cdot \left(\frac{\dot{QE}_{i+1,j,k} + \dot{QE}_{i,j,k}}{2 \cdot 1000} \right) \cdot \left(\frac{\Delta T_{i,j}}{3600} \right)$$

It is expressed in kWh.

At this point, it's possible to calculate the electrical performance indexes expressed in kW/kW. They are developed on three levels:

1. *Machine level* **-** *COPE₁* - in which only the electric power required by chiller for its internal operation is considered;

2. *Circuits level* - $COPE_2$ - in which, besides the chiller electric power, the electric powers of external pumps used for moving the heat transfer mediums within the chiller are considered;

3. *Heat Rejection level*- $COPE_3$ – in which also the consumption of cooler used for the heat rejection is considered.

$COPE_1$ is calculated as the ratio of chilling capacity divided by the electric chiller power according to the following equation:

Equation 4-15
$$COPE_1 = \frac{\overline{\dot{Q}}_{ch}}{\overline{\dot{QE}}_{chiller}}$$

$COPE_2$ is calculated as the ratio of chilling capacity divided by $\overline{\dot{QE}}_2$ - i.e. the sum of electric chiller power with the electric powers of pumps estimated for the three circuits - according the following equation:

Equation 4-16
$$COPE_2 = \frac{\overline{\dot{Q}}_{ch}}{\overline{\dot{QE}}_2}$$

Where:

Equation 4-17
$$\overline{\dot{QE}}_2 = \sum_{k=1}^{3(circuits)} \overline{\dot{QE}}_k + \overline{\dot{QE}}_{chiller}$$

$COPE_3$ is calculated as the ratio of chilling capacity divided by $\overline{\dot{QE}}_3$ - i.e. the sum of $\overline{\dot{QE}}_2$ with the electric power required by heat rejection system - according the following equation:

Equation 4-18
$$COPE_3 = \frac{\overline{\dot{Q}}_{ch}}{\overline{\dot{QE}}_3}$$

Where:

Equation 4-19
$$\overline{\dot{QE}}_2 = \sum_{k=1}^{3(circuits)} \overline{\dot{QE}}_k + \overline{\dot{QE}}_{chiller} + \overline{\dot{QE}}_{cooler}$$

To calculate $\overline{\dot{QE}}_{cooler}$ is employed a mean value of electric power, expressed in kW, necessary to reject 1 kW of heat at medium temperature. This value refers to the electrical consumption of Wet Cooling tower available in literature.

Equation 4-20
$$\overline{\dot{QE}}_{cooler} = \overline{\dot{Q}}_{co} \cdot 0.07$$

where 0.07 are the kW of the electric power per kW of heat to reject.

4.7 Marking and Nameplate Data

The marking of chiller is the final result of the test procedure till now developed. It shall contain general data as well as all rating conditions - in particular that one individuated by manufacturer as *"Design Condition"* – and the evaluations carried out at these conditions and all useful information for the final customer.

In detail, it shall include:

- General Data:

  a) Manufacturer's Name and location
  b) Model number designation providing complete identification
  c) Voltage, *V*, phase, and frequency, *Hz*
  d) Chiller Electrical Power, *kW* – average of electrical power consumption for all auxiliary components including solution and refrigerant pumps, purge, control panel, burner fan, ….

In case of direct fired chillers, it has to be specified:

  e) Fuel – usually *Natural Gas*

- Rating at "Design Conditions":

  a) Net Chilling Capacity, *kW* – evaluated as mean power -
  b) Maximum Chilling Capacity *kW* – the maximum instantaneous power -
  c) Total Energy Input *kW* – evaluated as mean power -
  d) Chiller Efficiency, *COP*
  e) *Electric Efficiency, COPE* – efficiency calculated using, at denominator, the sum of mean electrical powers of chiller, system water pumps and heat rejection system.

- Design Conditions:

a) Inlet temperature at Desorber – i.e. Feeding Temperature
b) Inlet temperature at Condenser
c) Inlet temperature at Evaporator
d) outlet temperature at Evaporator
e) Mass Flow at Desorber
f) Mass Flow at Condenser
g) Mass Flow at Evaporator
h) Desorber Water Pressure Drop
i) Condenser Water Pressure Drop
j) Evaporator Water Pressure Drop

In case of direct fired chillers or steam fired chillers, it has to be specified:

k) Fuel pressure entering fuel train
l) Stem supply pressure

The off-design conditions and the relative ratings can be represented graphically by Power curves and Performance Indexes curves – i.e. COP and COPE curves - like those represented in Figure 4-7. This allows supplying the final customers with many data concerning chiller performances at different working conditions.

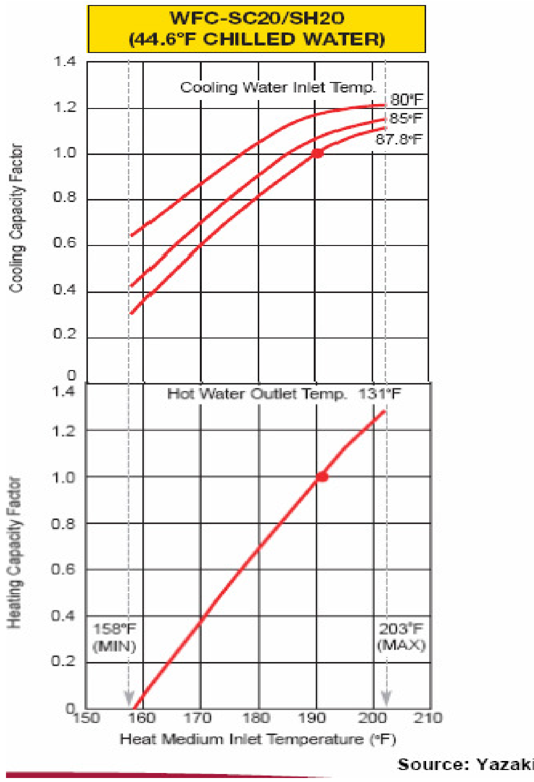

Figure 4-7 Power curves and COP curves for off-design conditions - Source: Yazaki

5 Test Facility

The tool used for the experimental investigation of considered sorption chillers and for the validation of the test procedure developed in the previous chapter, is a *Test Facility* for small-scale *Solar Combi Plus Systems* – i.e. systems using the solar heat for producing heating, cooling and DHW with a maximum of chilling capacity of 20kW - implemented at EURAC research in Bolzano, Italy.

It was born as result of a study carried out on the small and large scales sorption chillers currently present on the market, and on their particular application, the Solar Cooling, i.e. their combination with solar thermal collectors focused to the chilled water production used both for industrial purposes, e.g. food storing, and for summer air conditioning. Such study has been developed on three specific ambits:

- *Market* - in which a market analysis on air conditioning systems with a special focus on existing sorption chiller prototypes has been carried out;
- *On-going research activities* - in which the results coming out from monitoring activities on demonstration solar cooling plants, promoted, in particular, by International Energy Agency (IEA) through international projects like SHC Task38 and ROCOCO, and the installed test facilities for the experimental investigation of chillers have been analyzed;
- *Standards* - in which the norms concerning test procedures both for sorption and conventional chillers have been collected and analyzed;

with the aim to individuate the factors that mostly influence the performances of these machines meant both as single working units and as devices installed in specific plants. From this study, it has emerged that the low efficiency of these machines and of the solar cooling plants in which they are installed, especially the small scale ones, are due to:

- Lack of control strategies optimized as function of variable loads and of variable heat sources- i.e. solar heat source;

- Lack of standardized tools for the dimensioning of these plants and for the choice of auxiliary components;
- Low efficiency of sorption chillers in partial loads and not negligible transitory effects due to unsteadiness of heat source.

In order to overcome these weaknesses, the test facility has been designed following modularity and flexibility concepts, which allow obtaining several configurations simply by adding or removing single components from the main hydraulic system and reproducing different working conditions. In this way, the sorption chillers can be investigated under a number of reality-like working conditions– i.e. varying climatic conditions, cooling and heating loads - and system's configurations - storage volume and type on the hot and cold side, kind of heat rejection system, control strategy, etc –, allowing the individuation of optimized control strategies and energetic performance.

The present chapter concerns the presentation of the mentioned test facility. In particular the hydraulic scheme, the measurement equipment and the data acquisition system have been illustrated.

5.1 Hydraulic Scheme

The *Test Facility* here presented is a fully configurable energy system having the thermally driven chiller as a core component, at which a heat production system, heat rejection system and chilling/user system are connected – see Figure 5-1.

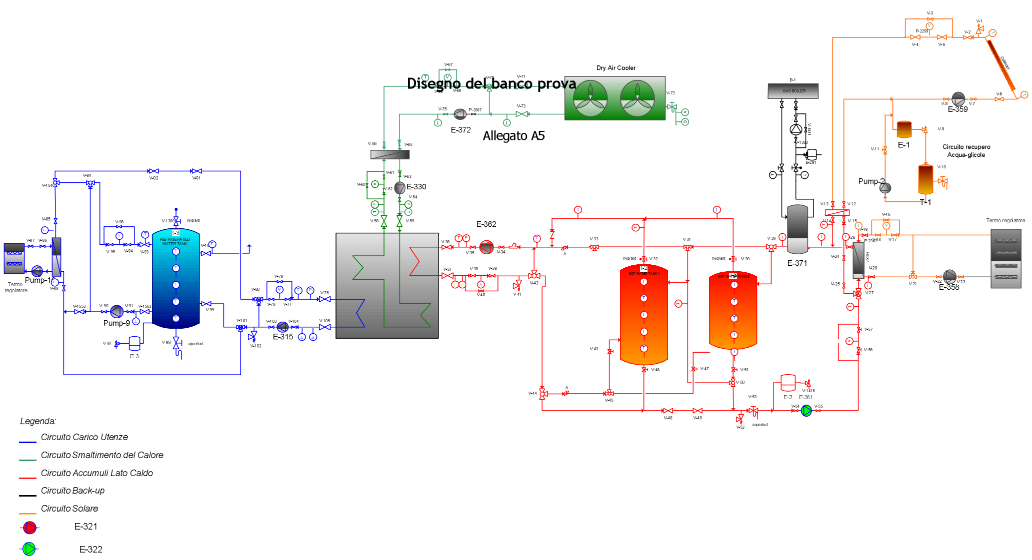

Figure 5-1 Hydraulic Scheme of *Test Facility*

5.1.1 Heat Production System

The heat production system consists of three heat sources:

- Thermal regulator
- Solar Collector field
- Gas boiler

which can be used singly or in conjunction. They are embedded in independent cycles and connected to the main cycle with a heat exchanger with the exception of gas boiler which, for legal prescriptions, allows vessels only – see Figure 5-2.

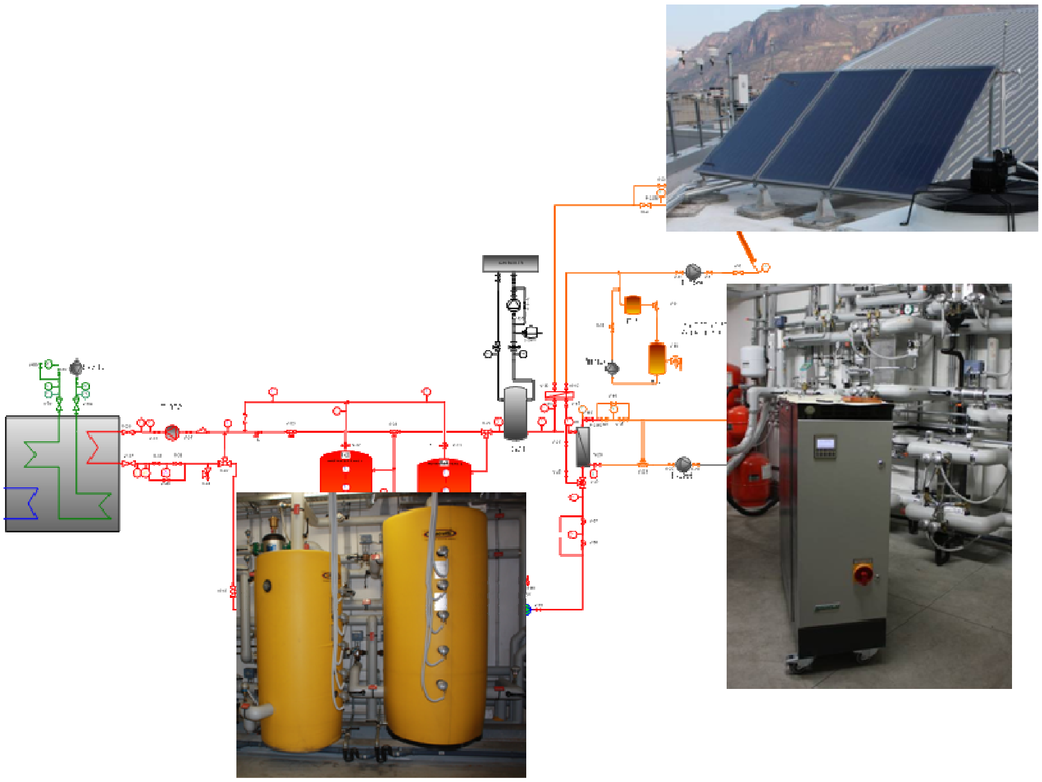

Figure 5-2 Heat production System

The heat is produced mainly by the thermal regulator, which consists of a series of electrical resistances able to reach 140°C with a maximum heating capacity of 40kW. It can replicate the heat produced by the installed reference collector field, consisting of three flat plate solar collectors, as well as the characteristic loads of different solar collector's type systems set up in different locations. Furthermore, it can produce heat at constant predefined temperatures. The gas boiler serves as back-up system which starts working when the temperature reached by solar collectors - or thermal regulator - is too low. In this way the real working conditions are reproduced. Water storages of 1000l and 500l are also installed. They can be connected both in series, obtaining a tall water column of 1500l completely stratified, and in parallel, obtaining a storage of 1500l. Furthermore, thanks to 3-way valves system, different configurations can be set up. It can be chosen to let use the produced heat directly by chiller or to store it in the different tanks for a later usage; or to send the working fluid coming from chiller desorber directly to the thermal regulator or to let pass it trough the tanks, in order to recover the heat. A 3-way valve, installed just before the chiller and activated by a PID controller, allows regulating the inlet temperature at chiller desorber instead. Variable pumps are also installed in order to work with different volume flows.

5.1.2 Heat Rejection System

The heat rejection system consists of two hydraulic circuits connected thermally by a heat exchanger: one is water filled and goes into the machine; and the other one is glycol/water filled and goes into the heat rejection device – see Figure 5-3.

As heat rejection system, a hybrid air cooler working as wet or dry cooling tower, depending on the targeted temperature, is used. It has a maximum cooling capacity of about 50 kW and consists of three ventilators working in cascade: the first one having the rotational speed regulated by inverter activated through a PID controller; and other two ventilators able to work only in ON/OFF mode. These three ventilators are activated as a function of required cooling needs. More precisely, when the first ventilator has already reached its maximum rotational speed, the other two start working in cascade. In case the thermal capacity is too low, even if the three ventilators are all working, sprinkler system spraying water on the fin of ventilators starts working as well. This enhances the cooling capacity and makes possible to cool the medium below the ambient temperature. The water used by sprinklers is produced by a reverse osmosis system is which provide to demonize it. Furthermore, a 3-way valve, installed just before this device and activated by a PID controller, allows regulating the temperature of the fluid entering into the chiller condenser.

Also in this case variable pumps are installed in order to work with different volume flows. The heat rejection system is partly installed within the laboratory and partly on the roof.

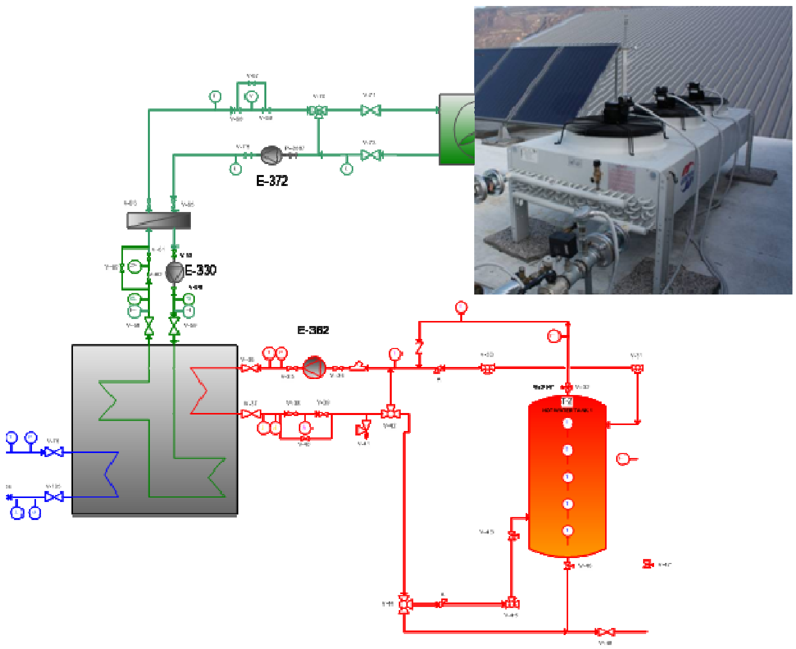

Figure 5-3 Heat Rejection System

5.1.3 Chilling/User System

The Chilling/User system consists essentially of a thermal regulator, able to reach 90°C with a maximum heating capacity of 20kW, and of cold water storage – see Figure 5-4. In this case the thermal regulator dissipates the produced cold by reproducing the return temperature from hypothetic distribution systems installed in different building types. It acts as the User. While the cold tank, stores the chilled water and together with the thermal regulator, simulates different cooling loads. Thanks to a 3-way valves system, cold produced by chiller can be stored in the tank or can be directly delivered to the virtual user. Furthermore, a 3-way valve, installed just before the inlet of chiller evaporator and activated by a PID controller, allows regulating the temperature of the fluid entering into the chiller.

Also in this case variable pumps are installed in order to work with different volume flows.

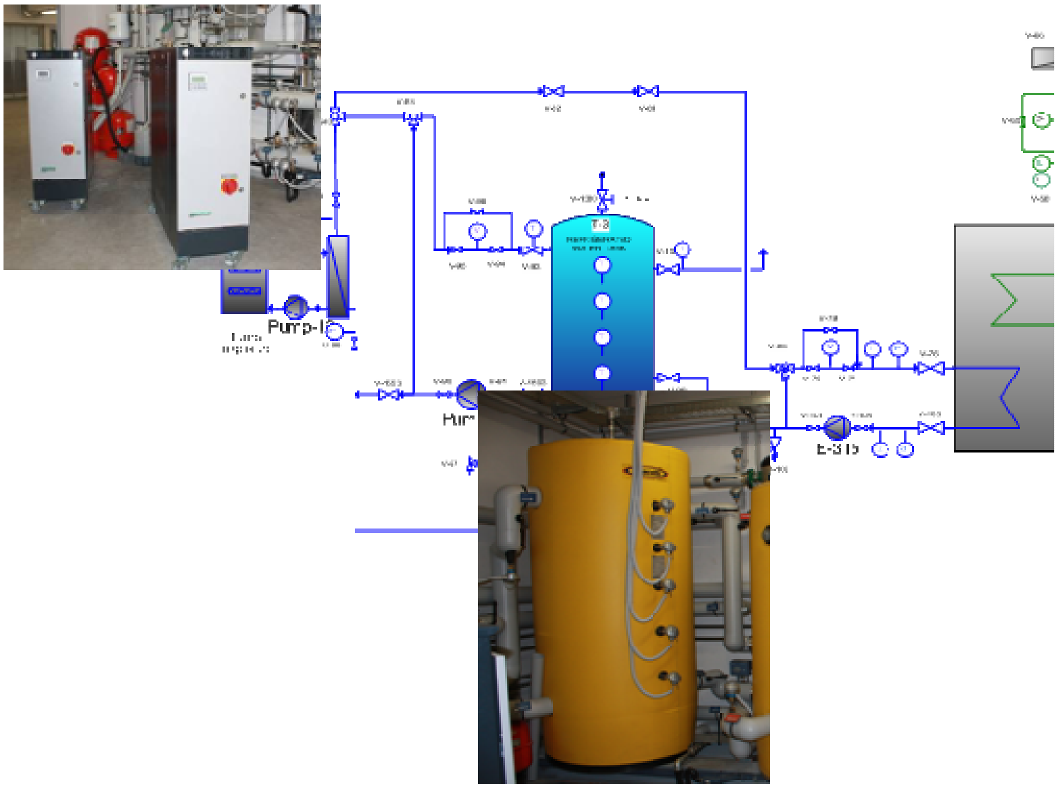

Figure 5-4 Chilling/User System

5.2 Measurement Equipment

To monitor all meaningful quantities of the test facility, a series of sensors for measuring temperatures, pressures, volume flows and electric consumptions have been installed. They have been selected according to the requirements prescribed by the developed test procedure, concerning typology and accuracy.

For the temperature measurements, 4-wires, Pt100, Class A thermo resistances have been chosen – see Figure 5-5. They have been placed at the inlets and outlets of all components installed in the test facility as well as in the storages, after the mixing points of working fluid and close to all 3-way valves in order to have temperature measurements for the whole test bench.

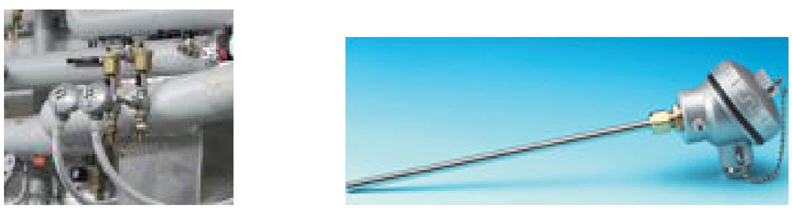

Figure 5-5 Thermo resistances Pt100 installed in the test facility – Source: TC Direct

For the pressure measurements, only six probes have been used and installed at the inlet and outlets of sorption chillers with the aim to measure pressure drops at internal heat exchangers of the tested sorption chillers.

Figure 5-6 Pressure probe installed in the test facility - Source: Siemens

For the volume flows measurements', electromagnetic flow meters, supplying highly accurate values – i.e. ±0.25% of measured value –, have been chosen – see Figure 5-7. They are 7 in all and have been installed at the three inlets of the sorption chiller, in the solar collector loop, in the secondary heat rejection loop – i.e. the circuit of the hybrid air cooler – in the two thermal regulators loops and, on the chilling side, in the portion of circuit between the cold storage and the thermal regulators loop.

Figure 5-7 Volume flow meters – Source: Siemens

Finally, to measure the electric consumptions of the chiller, hybrid air cooler, pumps and actuators of 3-way valves, five electric meters have been installed: three for measuring the electric consumptions of heating, cooling and chilling circuits; one for measuring the electric consumption of the chiller and one for the cooler.

Figure 5-8 Electric meters – Source: Vemer

Table 5-1 summarizes the main data of the sensors installed in the test facility, for which, the number of devices is specified. In particular the temperature sensors have been subdivided as function of their working range temperatures.

Table 5-1 Measurement Equipment

Measurand	*Sensor Description*	*Company*	*Measurement Range*	*Measuring Principle*	*Number of Devices*
High Temperature	4-Wire PT100 Class A	TC Direct	60 … 100 °C	Resistive	20
Medium Temperature	4-Wire PT100 Class A	TC Direct	10 … 50 °C	Resistive	8
Low Temperature	4-Wire PT100 Class A	TC Direct	0 … 30 °C	Resistive	15
Pressure	QBE2002-P10	Siemens	0 … 10 bar	Piezoresistive	6
Volume Flow	Sitrans F M Magflo MAG 6000	Siemens	0 … 10000 l/h	Electromagnetic	7
Electric Meter	Energy-230 D63A Class 1	Vemer	0… 0.1 kWh	-	5

Concerning the uncertainty of sensors – i.e. uncertainty Type B -, for pressure probes and electric meters, the values carried in their data sheet have been used, since it was not possible to do any calibration or achieve any information about this. While, the uncertainty of volume flow meters and thermo resistances has been calculated using the values obtained by calibration activities carried out on them.

In particular, for volume flow meters, the calibration sheet supplied by manufacturer Siemens has been used, since it is not easy to find the equipment necessary for their calibration. While, concerning the thermo resistances, each sensor has been calibrated at those temperatures at which it usually works. Therefore, the sensors have been subdivided in three groups - i.e. high, medium and low temperatures sensors. In this case, the uncertainty Type B has been calculated applying the law of error propagation for which:

Equation 5-1 $$U_B^2 = \sqrt{U_{Calibration}^2 + U_{\mathrm{Re}\,ference}^2 + U_{PXI}^2}$$

Where:

- U $_{Calibration}$ is the difference between measuring device and reference device or in absence of calibration, it is the specified uncertainty of the device (*Data Sheet*)
- U $_{Reference}$ is the uncertainty of calibration devices
- U $_{PXI}$ is the uncertainty of data acquisition system

In table, the uncertainty values of all sensors have been listed.

Table 5-2 Uncertainties of sensors installed din the test facility

Measurand	*Uncertainty Type B*
High Temperature	< 0.035 °C
Medium Temperature	< 0.035 °C
Low Temperature	< 0.035 °C
Pressure	< 0.1 bar
Volume Flow	< 0.25% of Full Volume flow
Electric Meter	Class 1

5.3 Monitoring and Control System

For the data acquisition and for the control of test facility, a PXI, produced by National Instruments, is used as hardware. It allows acquiring data in real time and communicating with the different devices of test facility- i.e. pumps, valves…- through chassis, system controller ans peripheral modules

From a software point of view instead, one developed in LabVIEW ambient is used. It is termed "*COSMO*" and was developed by Cardiff, a Spanish company. Figure 5-9 shows a screenshot of the main software window.

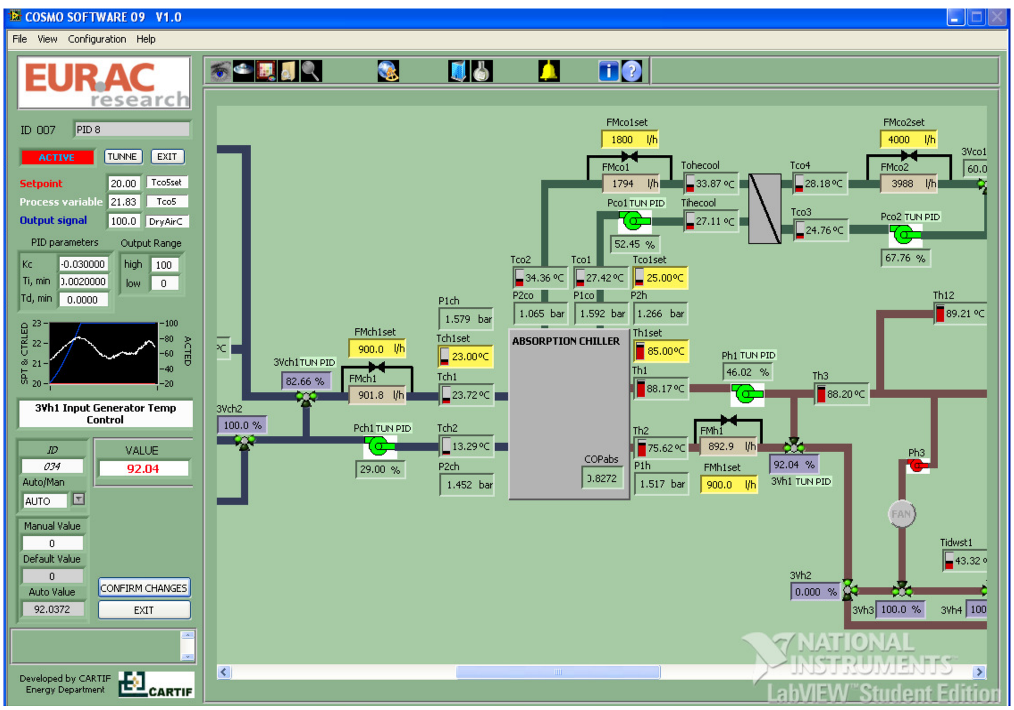

Figure 5-9 Screenshot of COSMO software

This software allows programming tests at different working conditions, monitoring and controlling the whole test facility and monitoring externally the chiller that is being tested. In order to do all these things, on the top of the main window, there are icons through which it is possible to manage variable, PID, etc.. - see Figure 5-10.

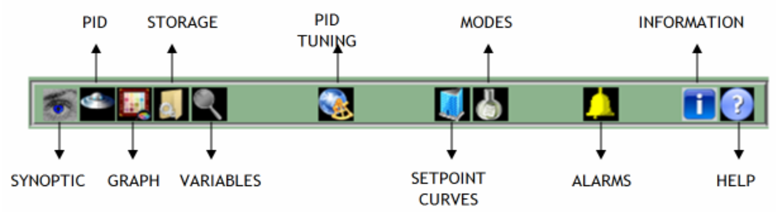

Figure 5-10 Main MENU-ICONS for handling Cosmo Software09

In particular:

- *SYNOPTIC*: is the window containing the whole test facility scheme. This window is active and allows setting different variables just clicking on the components figures;
- *PID*: is window in which it is possible to manage the PID controllers in general.
- GRAPH: is the window in which variables chosen by the user are plotted.
- STORAGE: allows storing the acquired data.
- VARIABLES: is the window in which it is possible to manage and define variables.
- PID TUNING: is the window containing the commands for PID tuning
- SETPOINT CURVES: is the window in which is possible to create curve with different set points, that the device to which they refer has to follow
- MODES: is the windows in which it is possible to define different system configurations with some prefixed values
- ALARMSINFORMATION: is the window containing all alarms.
- HELP: This panel has no function and can be ignored.

In the present work, COSMO software has been used for testing and monitoring externally the chillers at different working conditions.

6 Case Study: ClimateWell 10

The test procedure developed in the *Chapter 4* has been validated mainly on a prototype of ClimateWell 10 - Figure 6-1 -, the chiller chosen as core component for test facility previously introduced.

It represents the most emblematic case among the chillers currently present on the market because, even if it is classified as an absorption chiller – its working pair is a salt solution -, it behaves more like an adsorption chiller capable also to work in pure batch-mode and to serve as chemical storage. This peculiarity makes inapplicable most of the prescriptions of dedicated existing standards rendering it the ideal case for this test procedure.

The present chapter concerns with the results achieved from the application of the developed test procedure through the execution of tests on above mentioned chiller and with the comparison of such results with those obtained by applying backwards the test procedure on other two sorption chillers. The test files have been supplied by partner research institutes. In order to carry out an accurate analysis of considered chillers operation and to obtain meaningful evaluations, an introduction about their functional scheme, working phases as well as main working parameters has been done. It has done particularly for CW10.

Figure 6-1 ClimateWell 10 – Source: ClimateWell

6.1 CW10: Description and Working Principles

6.1.1 Functional Scheme

ClimateWell 10 consists of two identical under-vacuum units termed barrels – see Figure 6-2. Each unit includes a reactor, which works as desorber or absorber depending on the working phase, and a condenser/evaporator connected with it through a vapour channel. Vessels to store water and active salt, in solution phase and/or in solid monohydrate crystals, as well as pipes and pumps to move them among all parts of the barrel are included too. Both these units are interconnected trough a hydraulic switching unit consisting in 8 3-way valves, which control the flow from the external circuits to the two units, allowing their operation in quasi-continuous mode. These external circuits can be kept separate to each of the four heat exchangers, or they can have a common sink and thus a unique circuit for absorber and condenser. This last option was chosen.

As working fluid a salt solution is used, in which the absorbent is *Lithium Chloride* and the refrigerant – or solvent – is *Water*.

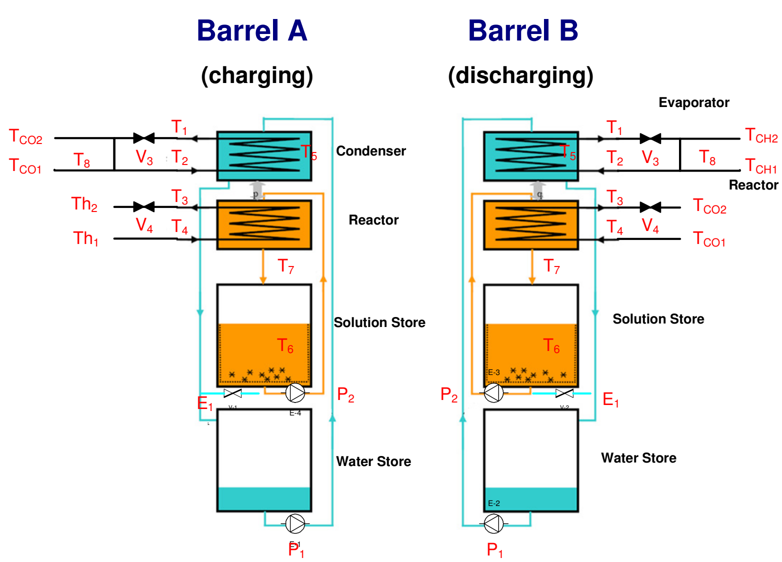

Figure 6-2 Functional Scheme of CW10- Source: ClimateWell

6.1.2 Working Phases

The machine operation is characterized by two working phases, i.e. *Charging phase* and *Discharging phase*, which are performed periodically by the two units.

During the Charging phase, the diluted absorbent solution is pumped from solution store to the reactor, where the refrigerant evaporates thanks to the heat introduced at high temperature – in solar cooling plants, it comes from solar collectors -. The produced vapour goes to the condenser where it condenses and flows to water storage. During this phase, the salt solution comes closer and closer to saturation, and when it reaches saturation point, further desorption results in the formation of monohydrate crystals that fall under gravity into the vessel. Here, they are kept separate by a ClimateWell patented technology that prevents them from clogging together.

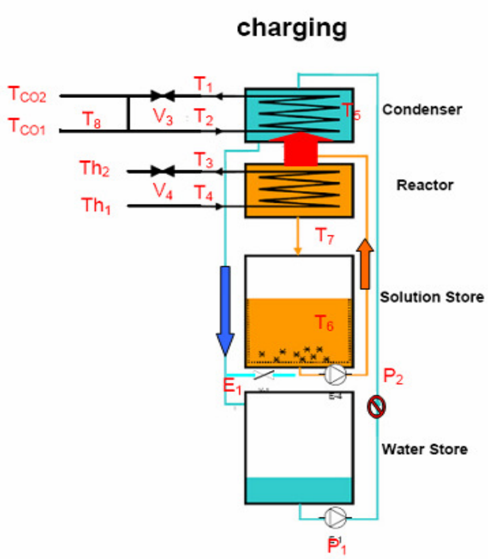

Figure 6-3 CW10: Charging Phase – Source: ClimateWell

In this way, high density energy is stored as mono-hydrate crystals and good heat and mass transfers are obtain as this occurs with solution. Once sufficient refrigerant has been moved to the condenser the discharge process can start.

During the *Discharging* phase, the process is reversed. The water is pumped from the water store to the evaporator where it evaporates and flows to the reactor - the heat for the evaporation is provided by the building when the chiller is used for the summer air conditioning -. In the reactor, the vapor of refrigerant is absorbed by the solution that becomes unsaturated. At this point, some monohydrate crystals are dissolved making the solution fully saturated again. In this way the solution is always saturated and the net result is a dissolving of the crystals into saturated solution that improves the absorption process.

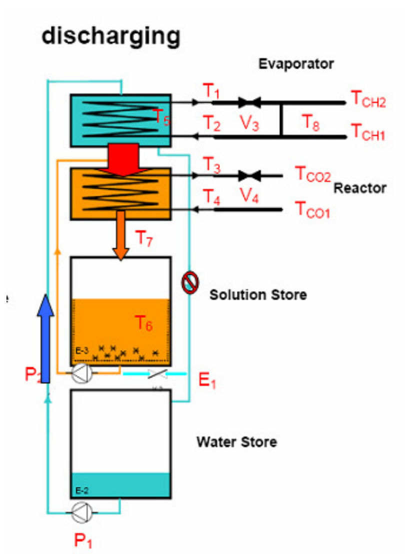

Figure 6-4 Discharging phase – Source: ClimateWell

The discharge process will continue as long as there is refrigerant left in the evaporator and the absorbent solution is sufficiently concentrated. When either of these conditions is not fulfilled the machine is empty and will swap to charging process again.

6.1.3 Parameters

The machine operation is regulated by an internal controller that switches the external circuits to the two barrels according to the specific swapping strategy and to initial settings. The machine allows, through an interface, the user choosing among a number of settings including the *Operation Mode* (meant as *Heating* or *Cooling* mode), and the *Swapping Strategies*.

There are eight different swapping strategies but the most important ones are:

- *NORMAL,* which is the strategy used for usual operation in installations with solar collectors. According to this swapping strategy, the machine works in semi-continuous mode and swaps the barrel when one barrel is fully discharged, *or* when the other is fully charged. In this swapping strategy, the priority is given to charging process.
- *FULL CYCLE,* is the strategy with which the machine works always in semi-continuous mode, but swaps the barrels only when *both* the discharging and charging cycle are completed. In this swapping strategy both processes have the same importance.
- *DOUBLE,* is the strategy that allows the machine working in pure batch mode. In this case, both barrels are charged and discharged at the same time. This results in a higher cooling power when in cooling and a higher charging power when in charging.

To establish when the barrels are charged or discharged, the internal controller checks the values of *Status Barrel Indicators*, which are:

- *Q,* the *Absorption Status Indicator* that indicates the “quality” of absorption process;
- *F,* the *Charging Level Indicator* that indicates the “quality” of desorption /charging process.

Other parameters are employed to ensure a proper operation of the machine in different working modes but, since they don’t concern the topic of the present work, they have been left out.

6.1.4 CWIC Software

To set the desired values of the different parameters as well as to monitor the machine internally, ClimateWell supplies users with a software named CWIC – i.e. *ClimateWell Internal Controller* – through which, they can interact with the machine. It is the mind of the Internal Controller and allows knowing, in real time, the status of internal valves and pumps and all internal temperatures both at inlet and outlet of heat exchangers. Furthermore, it allows also regulating and controlling the machine during its running through functions able to activate mechanical parts within the machine like, for example, the "*swap*" which determines the swap independently from the barrels status - see Figure.

Since it allows to monitoring and controlling the machine internally, it has been employed together with the software COSMO - introduced in the previous chapter - for the tests execution.

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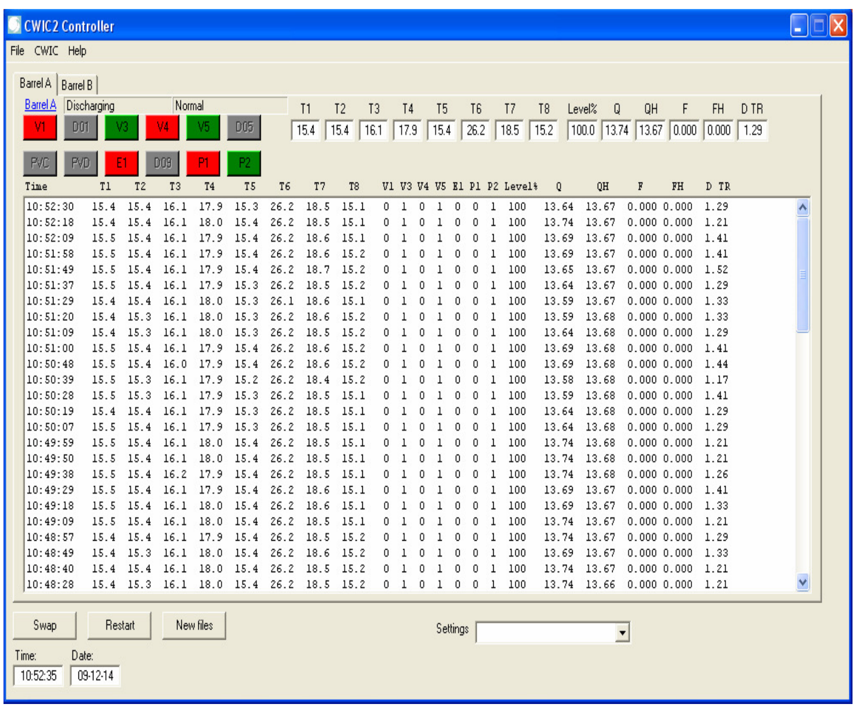


Figure 6-5 View of software CWIC used for setting and for monitoring internally the machine.

6.2 Run tests

ClimateWell 10 has been tested at its *Design* and *Off-Design* conditions, in accordance with the prescriptions defined in the developed procedure. In particular, a series of test has been carried out varying the inlet temperatures at desorber, condenser and evaporator systems. The mass flows have been set according to manufacturer instructions and maintained constant during the tests.

As inlet temperatures at desorber, the ones obtainable typically with solar collectors have been chosen among the range of feeding temperatures indicated by the manufacturer – i.e. ClimateWell. They are:

- 75°C (*low temperature*)
- 85°C (*medium temperature*)
- and 95°C (*high temperature*)

As inlet temperatures at condenser and evaporator instead, those indicated as rating conditions in Table 4.4 have been used, since they refer to the design temperature of the technologies usually coupled with this machine in solar installations. The only exception is represented by the condenser temperature relative to the wet cooling tower. According to the Table 4.4, it should be equal to 27°C. Nevertheless, in order to carry out tests with 75°C as inlet temperature at desorber, it's necessary to have a maximum of 25°C. This can be verified from Figure 6-6 representing the Dühring plot of solution Water/Lithium Chloride, in which it is observable that a ΔT_{equ} for charging process is about 50°C, i.e. the required difference of temperature between desorber and condenser; while for discharging process ΔT_{equ} is about 35°C.

Table 6-1 summarize the tests carried out on CW 10 and the relative rating conditions. Concerning the mass flows, the design values have been used and they are:

- 0.9 m^3/h at desorber
- 0.9 m^3/h at evaporator
- 1.8 lm^3/h at condenser

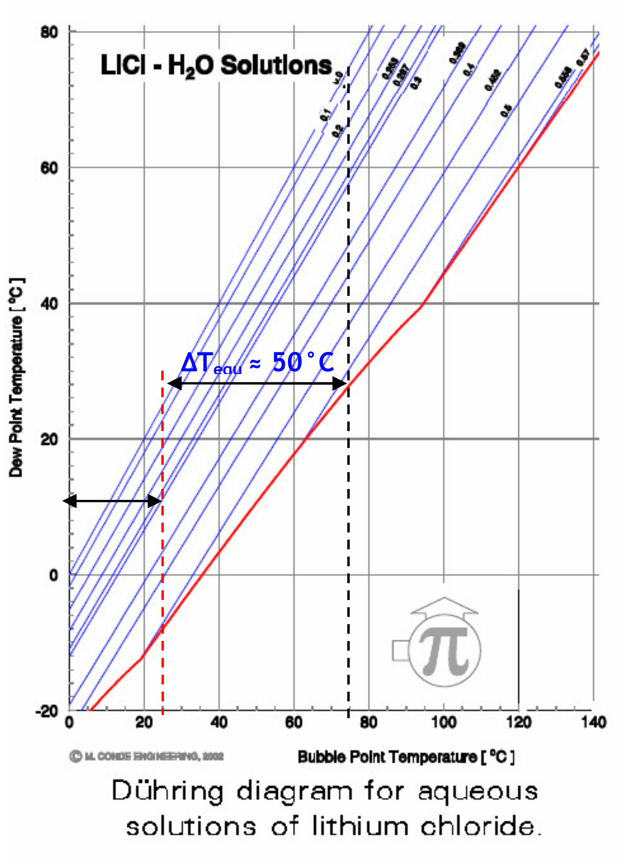


Figure 6-6 Dühring Chart for LiCl: water vapour pressure above the solution for varying mass fraction of sorbent and Solution Temperature.

Table 6-1 Rating Conditions used to test CW 10. The symbol "X" indicates the tests carried out on the machine

		Test								
		Condenser [°C]								
		25	30	35	25	30	35	25	30	35
Generator [°C]	75	X	-	-	X	-	-	X	-	-
	85	X	X	-	-	-	-	X	X	X
	95	X	X	-	-	-	-	X	X	X
			12			18			23	
		Evaporator [°C]								

Concerning the swapping strategy that usually applied for solar cooling plants has been used for running the tests – i.e. “Normal” strategy. Since it ensures a semi-continuous operation, the prescriptions defined for this working mode have been considered.

6.2.1 Methodology

To assess the chiller’s performances, the data collected during the four cycles after the achievement of stationary condition, have been used. Figure 6-7 shows the 8 cycles analyzed for each of the test: 4 cycles for the reaching of the stationary conditions according to the prescriptions relative to semi continuous chillers; and 4 cycles for the execution of the test itself and for the data collection.

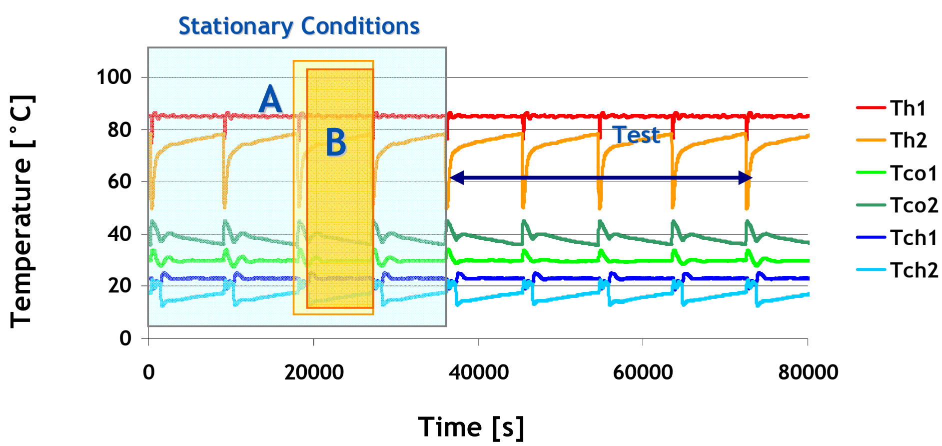


Figure 6-7 Cycles used for carrying out a test at specific rating conditions: 4 cycles for establishing the stationary conditions; and 4 cycles for performing the test itself.

For each of the cycles used to establish the stationary conditions, the subintervals A and B have been individuated, since the restrictions concerning the fluctuations of observed quantities are based on them. More precisely, the *Sub-Interval A* consists of all data collected during each cycle time, i.e. the mean time between two consecutive swaps; and *Sub-Interval B* consists of data collected during each cycle time with the exception of those relative to the swapping period.

Concerning the data used for the analysis, they are summarized in table and refer both "external" quantities – i.e. temperatures, pressures… - and internal quantities – i.e. status barrels indicators – of the machine.

Table 6-2 Data used for the chiller assessment obtained by CIWC and COSMO softwares

Data	*Generator*	*Evaporator*	*Condenser-Ab/Adsorber*
Mass Flow	X	X	X
Inlet Temperatures	X	X	X
Outlet Temperatures	X	X	X
Static Pressure Differences	X	X	X
Cycle Time		X	
Swapping Time		X	
FH Indicator - Barrel in Charging		X	
QH Indicator - Barrel in Discharging		**X**	

6.3 Analysis of the Results and Discussion

With the data achieved from the tests, an accurate analysis of the chiller performances has been carried out with the aim to obtain useful figures to insert in the final marking. In particular, the main quantities characterizing the machine (like temperatures, pressures, etc… as well as powers and COP) have been investigated at different working conditions.

Below the profiles of machine key features, obtained from the tests performed at its design conditions are plotted in order to illustrate the typical machine operation. Figure 6-8 shows the inlet and outlet temperatures of the three heat exchangers – i.e. desorber 85°C, condenser/absorber 30°C and evaporator 23°C -, from which it is inferred their strong dependence both on the machine semi-continuous operation and on the sorption processes. In particular, the inlet temperatures, after a certain period in which there are rough fluctuations due to the swaps, keep constant around their set values; the outlet temperatures rise as desorption and absorption processes proceed, coming more and more close to the inlet temperatures towards the end of each cycle. This is easily observable comparing trends of outlet temperatures with those of desorption – FHA and FHB - and absorption - QHA and QHB – indexes of the two barrels, which are plotted in Figure 6-9. In fact, when these two indexes increase reaching their maximum values at the end of each cycle, the outlet temperatures raise going close to the inlet temperatures. This results in decreasing heating, cooling and chilling powers along the cycle. Figure 6-10 shows these powers trends with the relative COP.

Particular attention has to be given to the chilling power, which is null just after the swap- see circled areas in Figure 6-8 and in Figure 6-9 -, since the machines require time to reach again the equilibrium at the evaporator. It is possible to ascertain this, also observing the corresponding temperatures at inlet and outlet of evaporator, which are equal for the whole swapping time - see dotted area both in Figure 6-8 and Figure 6-10. This has a negative impact on COP which, being null during this period, suffers a lowering of its mean value: in fact, it passes from an instantaneous value around 0.8, calculated when the machine delivers chilling power, to an average value of 0.6 calculated on the whole cycle. This reasoning is valid also for the chilling power, since its final value is calculated as average of all instantaneous powers calculated on the whole cycle including also those null.

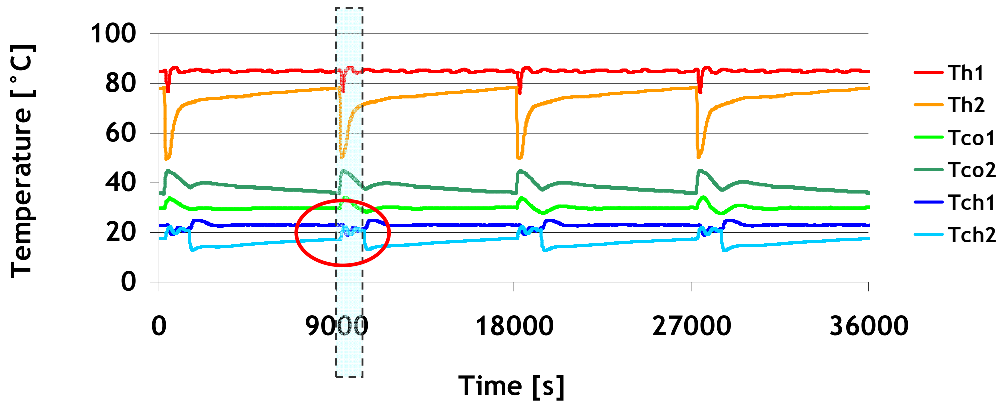


Figure 6-8 Inlet and Outlet Temperatures at desorber, condenser and evaporator vs. time recorded during a test carried out at machine design conditions: 85-30-23°C

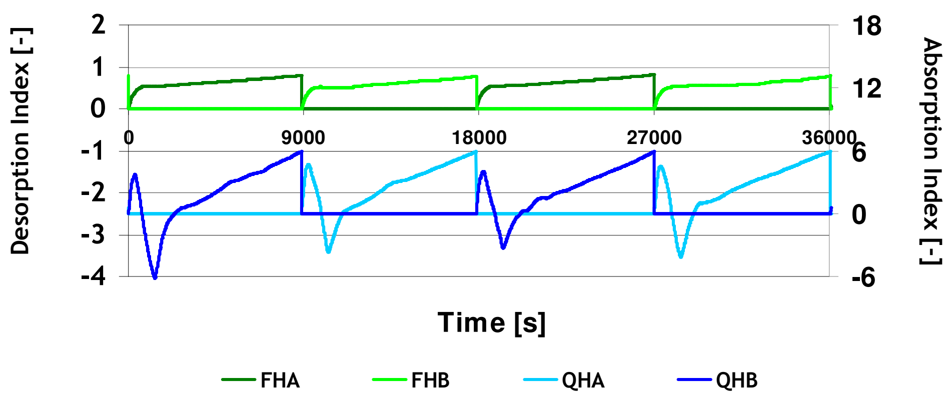


Figure 6-9 Desorption and Absorption indexes of the two barrels during the charging and discharging phases vs. time recorded with CWIC software during a test carried out at machine design conditions: 85-30-23°C

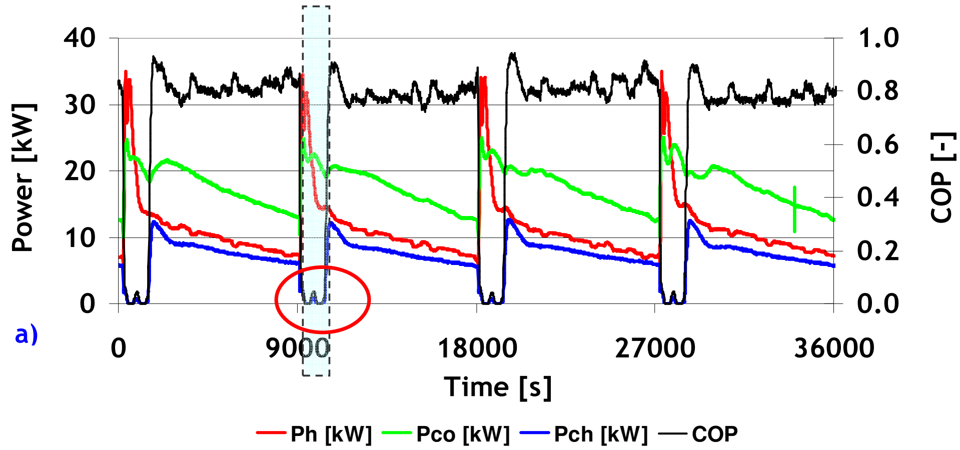


Figure 6-10 a) Heating, Cooling and Chilling powers vs. Time; b) COP vs. Time recorded during a test carried out at design conditions: 85-30-23°C

A similar behaviour can be observed also for the other machine features, i.e. volume flows, static pressures differences and electric power consumptions. In this case, the fluctuations of their values are due to the fact that, when the machine switches the external circuits to the two barrels, deviates the inlet flows in order to bypass the heat exchangers. This produces sharp reductions of volume flows followed by huge pressure drops which are reflected in increases of the localized electric power consumptions – see Figure 6-12

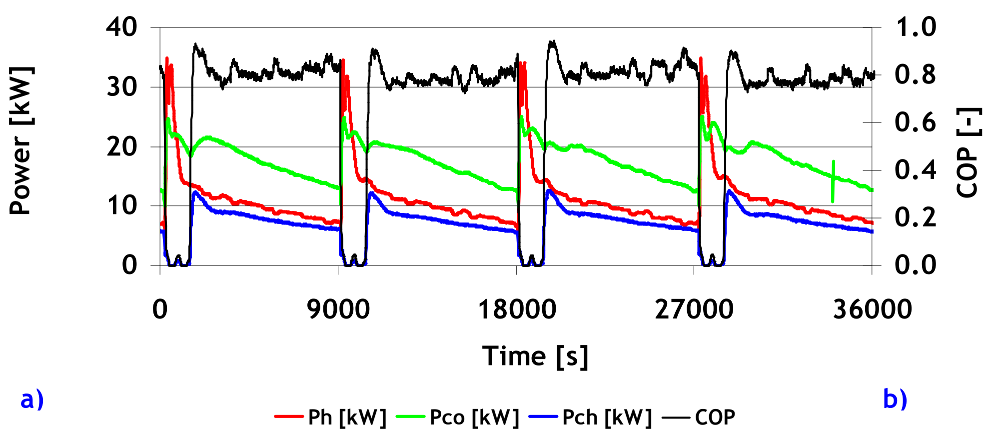


Figure 6-11) Electric powers vs. Time; b) COP2 vs. Time recorded during a test carried out at design conditions: 85-30-23°C

Another remark has to be done, as usual, with regard to the evaporator. In fact, during the swapping time, the inlet flow is reduced by nearly half until the machine has reached the equilibrium at the evaporator. This is due to valves partially closing, which produces localized pressure drops along the whole swapping time with a consecutive increase of electric power consumption. Obviously also the $COPE_2$, calculated as the ratio of chilling capacity divided by the electric consumption due to pressure drops within the machine and of external circuits as defined in Chapter 4.6, is influenced by these dynamics. In particular, just after the swap, it has a maximum value; then it is equal to zero for the whole swapping time since the chilling capacity is equal to zero; and it decreases until the next swap both due to the decrease of the chilling capacity and due to the localized energy consumption. All these remarks can be observed in the Figure 6-12 and Figure 6-13.

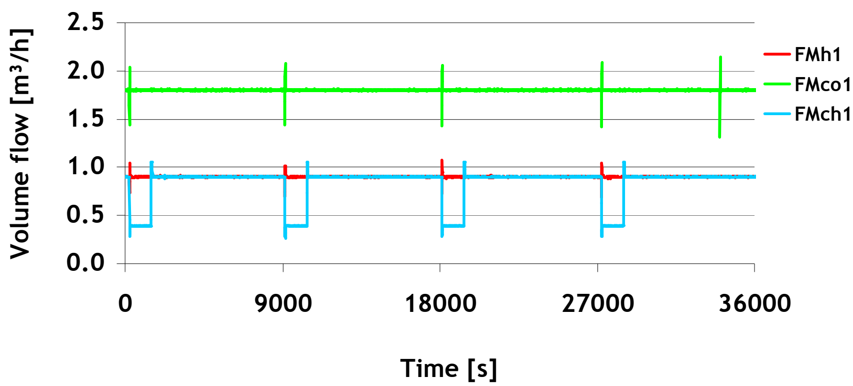


Figure 6-12 Volume flows at desorber, condenser and evaporator vs. time recorded during a test carried out at machine design conditions: 85-30-23°C

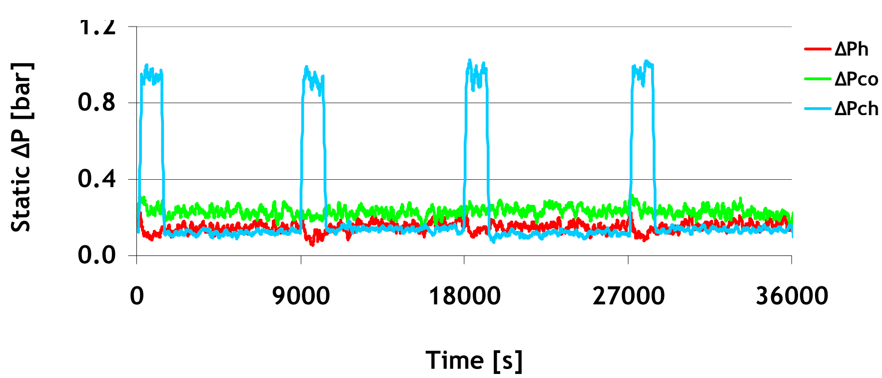


Figure 6-13 Static Pressure Differences at desorber, condenser and evaporator vs. time recorded during a test carried out at machine design conditions: 85-30-23°C

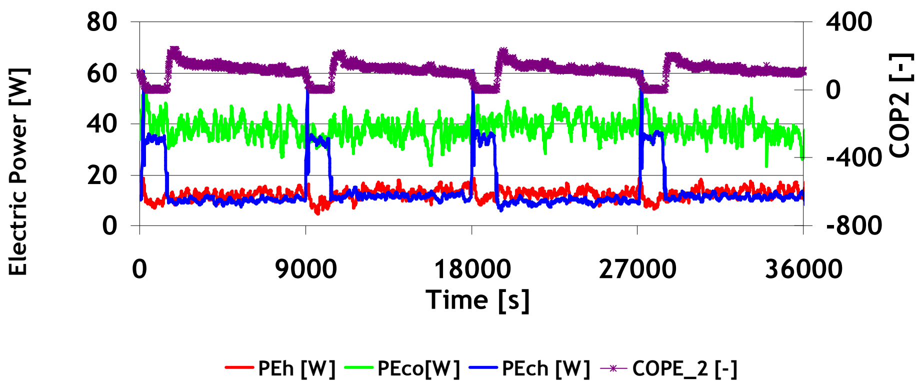


Figure 6-14 Electrical Powers consumption at desorber, condenser and evaporator and COP2 vs. time calculated for a test carried out at machine design conditions: 85-30-23°C

6.3.1 Analysis of the Chilling Capacity and thermal COP

At this point, the chiller performance, evaluated at different working conditions, are analyzed and compared with the aim to individuate those conditions at which the chiller is more performing. Obviously, for the calculation of the different figures, the equations written in chapter 4 are used.**Error! Reference source not found.** summarizes respectively: the chilling capacity - calculated as average of instantaneous chilling capacities on the whole test-, the peak of chilling capacity and the COP at different rating conditions.

Table 6-3 Chilling Capacity calculated at different rating conditions

Generator [°C]	Evaporator 12 °C / Condenser 25	12 / 30	12 / 35	18 / 25	18 / 30	18 / 35	23 / 25	23 / 30	23 / 35
75	4.19	-	-	5.67	-	-	6.05	-	-
85	2.02	3.17	-	-	-	-	8.24	6.76	5.04
95	2.50	0.81	-	-	-	-	4.84	6.84	5.99

Chilling Capacity [kW]; Condenser [°C]; Evaporator [°C]

Table 6-4 Peak of Chilling Capacity calculated at different rating conditions

Generator [°C]	Evaporator 12 °C / Condenser 25	12 / 30	12 / 35	18 / 25	18 / 30	18 / 35	23 / 25	23 / 30	23 / 35
75	7.86	-	-	10.99	-	-	13.16	-	-
85	7.30	7.11	-	-	-	-	13.61	12.86	10.93
95	9.71	3.65	-	-	-	-	11.47	11.72	9.87

Peak of Chilling Capacity [kW]; Condenser [°C]; Evaporator [°C]

Table 6-5 COP calculated at different rating conditions

Generator [°C]	Evaporator 12 °C / Condenser 25	12 / 30	12 / 35	18 / 25	18 / 30	18 / 35	23 / 25	23 / 30	23 / 35
75	0.54	-	-	0.64	-	-	0.63	-	-
85	0.15	0.37	-	-	-	-	0.60	0.60	0.60
95	0.13	0.13	-	-	-	-	0.30	0.44	0.54

COP [-]; Condenser [°C]; Evaporator [°C]

Examining the values listed in Table 6-3 and Table 6-4, it is easy to observe that CW 10 doesn't follow the typical behaviour of an absorption machine, for which the chilling capacity rises when the inlet temperature at desorber increases and the one at condenser decreases. It shows different trends depending on the specific performance figures and on the specific rating conditions at evaporator. In general, keeping constant all other boundary conditions, the worst cases occur for inlet temperatures at desorber equal to 95°C; while, the best cases occur for inlet temperatures at desorber equal to 85°C.

Analyzing the data more in detail, it can be noted that with 23°C at evaporator, the machine is more stable and well performing with respect to the other cases. With regards to inlet temperature at condenser, the chilling capacity follows the typical behaviour of on absorption chiller, i.e. decreases when the temperature at desorber increases (with the exception of the case 95-30-23°C which represents the best case for 95°C at desorber).

For the inlet temperatures at evaporator equal to 12°C, the machine presents an instable operation. In fact, the manufacturer doesn't suggest this temperature as working condition. The listed values, therefore, are indicative and affected by this instable operation However, the values of chilling capacity and its peak, related to most stable cases, occur for 75°C at desorber and 25°C at condenser, although the highest peak of chilling capacity occurs at 95°C as inlet temperature at desorber.

Concerning the COP, it behaves more stably for all rating conditions: it increases when the inlet temperature at desorber decreases and the inlet temperature at condenser increases.

The reasons of these anomalous behaviour have to be investigated in the charging and discharging modalities of the two barrels, which depend, strongly, on the working temperatures. In particular, the charging "speed" depends on the inlet temperatures at desorber and condenser; while the discharging "speed" depends on the inlet temperatures at evaporator and condenser - meant as absorber.

Figure 6-15, Figure 6-16 and Figure 6-17 show the charging and discharging "speeds" of the two barrels as function of inlet temperatures at desorber, condenser and evaporator respectively. In particular, Figure 6-15 shows the charging "speed" as a function of the inlet temperature at desorber – i.e. 75°C, 85°C and 95°C – while the inlet temperature at condenser and evaporator are kept constant at 25°C and at 23°C respectively. By observing the speed profiles plotted in the figure and their trend lines, it is easy to deduce that such speeds increase when the inlet temperatures at desorber increase, resulting in shorter cycle times.

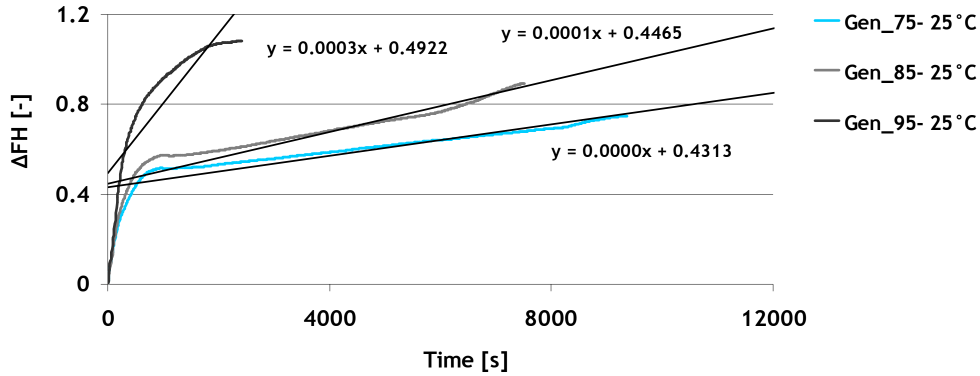


Figure 6-15 Charging "speed" of barrel vs. time calculated as function of inlet temperature at desorber

Similar remarks can be done for Figure 6-16, in which the charging "speed" is plotted as a function of inlet temperatures at condenser – i.e. 25°C, 30°C and 35°C – while the inlet temperatures at desorber and evaporator are kept constant at 85°C and at 23°C respectively. In this case, the charging "speed" increases when the inlet temperature at condenser decreases. Thus, to lower condenser temperatures correspond shorter cycle times.

In Figure 6-17 instead, the discharging "speed" is plotted as a function of inlet temperature at evaporator – i.e. 12°C, 18°C and 23°C-; while, 75°C and 25°C have been chosen as inlet temperatures at desorber and condenser respectively. Analyzing the speed profiles, it is possible to observe that when the inlet temperature at evaporator increases, the speed increases too. Nevertheless, this doesn't imply that the cycle time becomes shorter since, for the chosen swapping strategy, the charging process has more weight in the chiller operation. Furthermore it is possible to observe that at 12°C, the discharging process is quite unstable, if compared with the other ones. This is one of the reasons for the anomalies resulting at this temperature level.

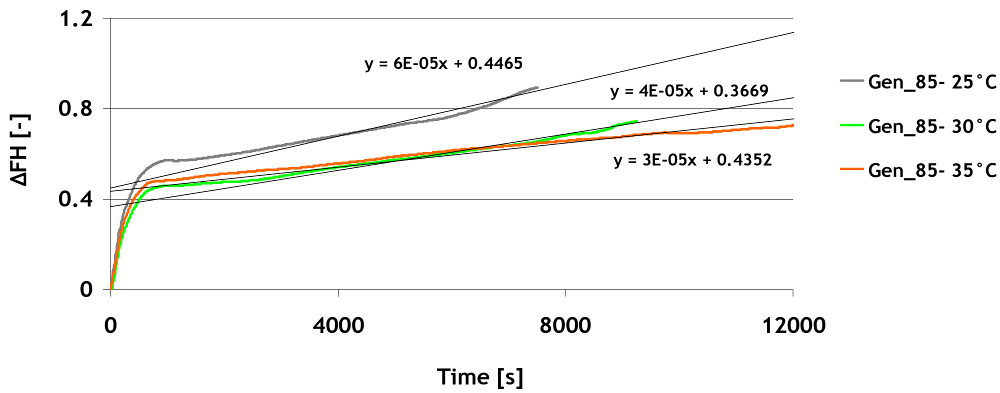


Figure 6-16 Charging "speed" of barrel vs. time calculated as function of inlet temperature at condenser

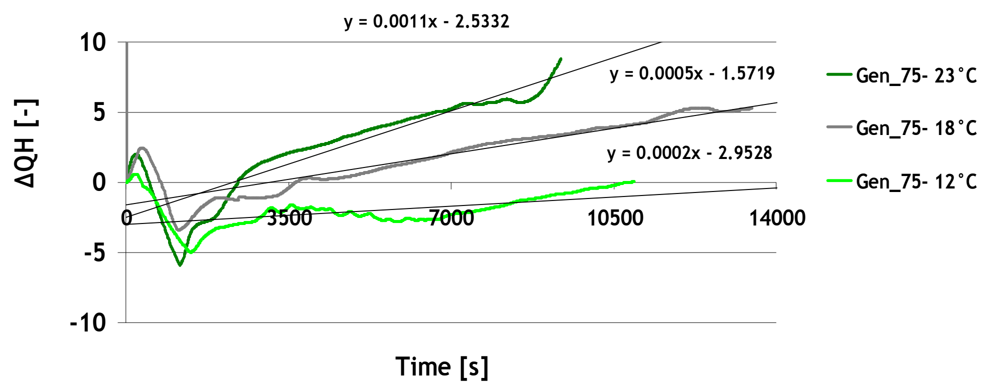


Figure 6-17 Discharging "speed" of barrel vs. time calculated as function of inlet temperature at evaporator

At this point, it is possible to analyze the influence that these quantities – i.e. charging and discharging speeds – and their interactions exert on the *Cycle* and *Swapping Times* and, indirectly, on the chillers performances. By observing the values listed in Table 6-6 it can be verified that the *Cycle Time* increases when the inlet temperature at desorber decreases and that at condenser increases. The swapping time is quite constant: it decreases lightly at low inlet temperatures at desorber.

Table 6-6 Cycle Time and Swapping Time calculated at different rating conditions

		Condenser [°C]								
Cyle Time		25	30	35	25	30	35	25	30	35
Generator [°C]	75	3.11	-	-	3.87	-	-	2.70	-	-
	85	0.68	1.88	-	-	-	-	2.02	2.51	3.39
	95	0.71	0.78	-	-	-	-	0.67	1.41	2.48
Swapping Time		25	30	35	25	30	35	25	30	35
Generator [°C]	75	0.33	-	-	0.29	-	-	0.32	-	-
	85	0.36	0.35	-	-	-	-	0.32	0.36	0.36
	95	0.35	0.35	-	-	-	-	0.36	0.36	0.35
		12			18			23		
		Evaporator [°C]								

An important aspect to be examined is the impact that the Swapping Time, i.e. the time during which the machine doesn't deliver any chilling power, has on the Cycle Time - see Table 6-7.

Table 6-7 Incidence of Swapping Time on Cycle Time calculated at different rating conditions

		Swapping Time/Cycle Time [%]								
		Condenser [°C]								
		25	30	35	25	30	35	25	30	35
Generator [°C]	75	10.62	-	-	7.55	-	-	11.74	-	-
	85	51.91	18.83	-	-	-	-	16.07	14.15	10.49
	95	49.08	44.68	-	-	-	-	52.67	25.16	14.24
		12			18			23		
		Evaporator [°C]								

From the values listed in Table 6-7, it is possible to observe that the bigger impacts occur at those working conditions usually considered ideal for common absorption chillers, i.e. 95°C and 85°C as inlet temperature at desorber and 25°C as inlet temperature at condenser. The worst case is, in fact, represented by 95-25-23 °C, for which the incidence is around 53%. This means that for more than the half of its cycle time, the machine, working at these conditions, doesn't deliver any chilling power. This results in low COP as well, since the machine absorbs heat at high temperature during the whole cycle. This explains also the difference between the peaks values of chilling capacities with respect to their mean values. Furthermore it explains why the COP calculated at 75°C, keeping constant all other parameters, is higher than that calculated at 95°C.

6.3.2 Analysis of the Electric Power Consumption and COPE

The last aspect to be analyzed concerns the electric power consumptions linked to the running of machine. This analysis, as explained in Chapter 4.6, has been carried out on three levels:

4. *Machine level - PE_1, $COPE_1$* - in which only the electric power required by chiller for its internal operation is considered;

5. *Circuits level – PE_2, $COPE_2$* - in which, besides the chiller electric power, the electric powers of external pumps used for moving the heat transfer mediums within the chiller are considered;

6. *Heat Rejection level- PE_3, $COPE_3$* – in which also the consumption of cooler used for the heat rejection is considered.

For the calculation of electric power and COPE at the first level, 100We have been used as electric power consumption attributable to the only machine. This value has been obtained calculating the average of the electric power consumptions recorded during all tests.

Concerning the calculation of electric power and COPE at the second level, they have been assess as function of the pressure drops and the volume flow collected during the test according to the Equation 6-1. While for the calculation of electric power and COPE at the third level, the electric consumption due to the heat rejection system has been calculated as function of heat to reject multiplied by reference electric power consumption - i.e. 0.07kW of electric power per kW of heat to reject.

The achieved results have been normalized with respect to a reference conventional chiller having a COP equal to 3.5. In order to do this, the specific values of electric power consumptions and COPE – i.e. related to 1 kW of chilling capacity – have been calculated at the all three levels and then divided by the specific COP and power consumption of the conventional chiller. The results of these evaluations have been plotted in Figure 6-18 and Figure 6-19. They allow observing both the electrical behaviour of the chiller at different working condition and its electrical performances compared with the conventional one.

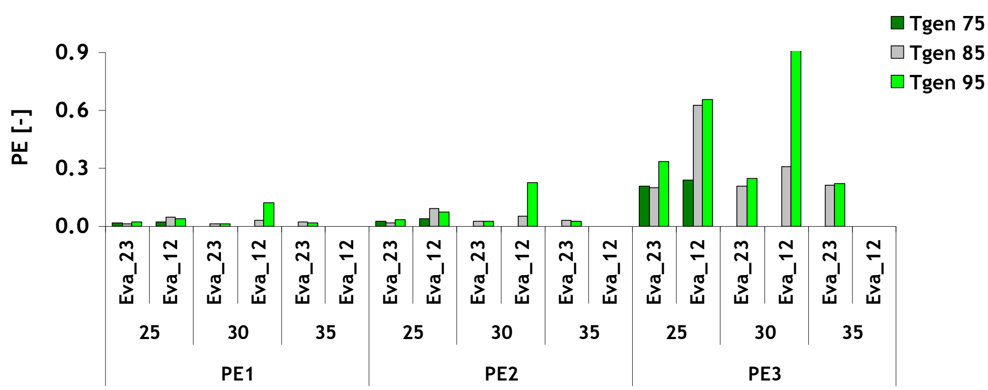


Figure 6-18 Electric power consumption assessed for different inlet temperatures at desorber vs. condenser and evaporator temperatures. The electric power consumptions have been evaluated on three levels.

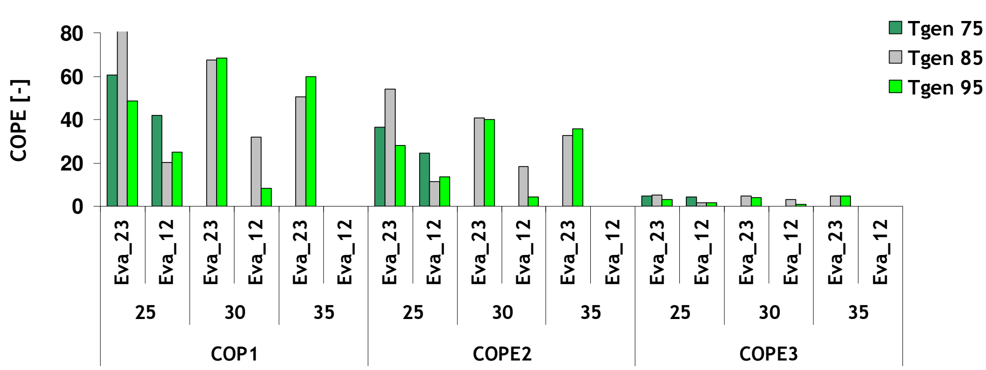


Figure 6-19 COPE assessed for different inlet temperatures at desorber vs. condenser and evaporator temperatures. COPE has been evaluated on three levels.

Analyzing the two graphs, it is possible to observe that at low inlet temperatures at evaporator – i.e. 12°C - correspond high electric power consumptions and, consequently, low COPE. This is due mainly to the instability of machine at these working conditions, at which it is characterized by short cycle times that generate huge pressure drops and increments of heat to reject; and by small chilling capacities that result in high specific electric power consumptions. Concerning the influence that the inlet temperatures at desorber and condenser have on these two figures - i.e. PE and COPE -, it is not possible to delineate a general behaviour. Nevertheless, it is possible to observe that with 12°C at the evaporator and 85°C at desorber, when the inlet temperature at condenser increase the electric power consumption decreases and the COPE increases; vice versa when the inlet temperature at desorber is equal to 95°C. While, with 23°C at evaporator, the electric power consumption increases tendentially when the inlet temperature at desorber increase and decreases when the inlet temperature at the condenser increases; vice versa for the COPE. The reasons of this behaviour have to be searched again in nature of the chiller operation and in particular, in the incidence of the swapping time on cycle time like for the thermal performances.

Nevertheless, it is possible to observe that, particularly with 23°C at evaporator, electrical performances CW10 are better than the conventional chiller.

6.4 Other Case Studies

The main result of the test procedure developed in Chapter 4 consists of the possibility to compare different chillers types. In fact, the use of a common methodology for their assessment allows achieving results that, since comparable, can be used for the selection of chillers to be installed in specific plants.

With this regard, it has been tried to apply the developed test procedure on other two chillers, i.e. on *Continuous* chiller, Suninverse - source: TU Berlin – and on *Semi-continuous* chiller, SorTech - source: LESBAT -. In this way, it is possible to validate the procedure also for chillers having different working modes from ClimateWell one, and to compare the achieved results both from an electric consumptions point of view and relatively to their use in solar cooling applications.

In order to apply the procedure to these two chillers, the data provided have been filtered according to the restrictions foreseen by the test prescriptions. Nevertheless, since it is not possible to intervene on the equipment used for performing the test as well as on the rating conditions at which they tested their chillers, only the prescriptions concerning the test methodology – i.e. stationary conditions, test duration, etc… - and the procedure for assessing the chiller performances have been considered and for the data analysis. In particular, for each test file, where it was possible, the intervals of data for the reaching and maintaining of stationary conditions, as well as those related to the test execution itself, have been individuated and treated as if they were belonging to real tests. Below, the two chillers are briefly introduced and the results achieved from the analysis of their performances are discussed. The chiller assessment has been done only at the rating conditions provided. Such results have been compared with those obtained for CW10 through an index created "ad hoc" to evaluate their performance in combinations with solar collectors.

6.4.1 Case Study A: Suninverse

6.4.1.1 Functional Scheme

Suninverse is a small scale absorption chiller – i.e. P_{ch}<20kW – produced by manufacturer SonnenKlima – see Figure 6-20/a). It is characterized by a continuous operation and has a chilling capacity of 10kW rated at its design conditions. The working pair is a solution of Water–Lithium Bromide, where the (distilled) water serves as refrigerant and Lithium Bromide, which is a very hygroscopic salt, serves as absorbent.

a) 

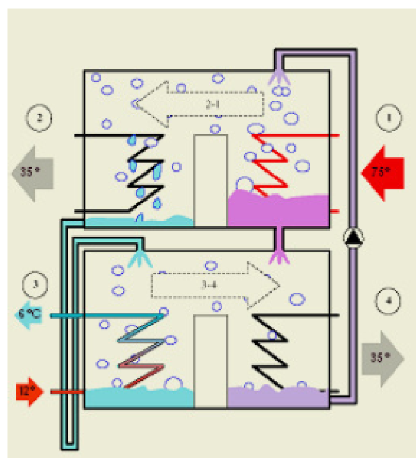
b) 

Figure 6-20 Suninverse: a) View of chiller; b) Functional Scheme – Source: SonnenKlima

From a constructional point of view – see the Functional Scheme in Figure 7 1/b -, it consists of the typical four components, i.e. desorber, condenser, evaporator and absorber, connected in series in order to guarantee a continuous operation. Each of them performs one of the four steps of thermodynamic cycle. In detail:

6.4.1.2 Tests and Analysis

The files containing the tests carried out on the Suninverse chillers have been supplied by TU Berlin and are 7 of all. They concern tests performed on the chiller at the rating conditions specified in Table 6-8, for which the sets of data representing the stationary conditions and the test itself have been individuated as shown in Figure 6-21.

Table 6-8 Rating Conditions of tests carried out on Suninverse by TU Berlin. The symbol "X" indicates the test carried out on the machine.

		Test						
		Generator[°C]						
		75	80	85	90	95	100	105
Condenser [°C]	27	X	X	X	X	X	X	X
		18						
		Evaporator [°C]						

To identify these two sets, it has been proceeded backwards by applying, on data already collected, the restrictions specified in Table 4.2, relative to continuous chillers. In particular, some filters have been employed to individuate those series of data which fluctuations from the set values fall in the permissible deviations. Once these data have been individuated, they have been subdivided in stationary conditions and in the actual test. As it can be noted, in this case, Subinterval A and Subinterval B coincide.

Concerning the sample time, 11 seconds have been used for collecting the data and, even if they are not exactly equal, in number, to those defined in the developed test procedure – i.e.10 seconds –, they can be considered useful since they allow observing all meaningful information about the chiller operation.

With the selected data, an analysis of the chiller performances has been carried out, by using the equations written in chapter 4, for continuous chillers, - i.e. with a number of cycles equal to 1. In particular, the Chilling capacity and the COP have been calculated and plotted in Figure 6-22. Analyzing this figure, it's possible to observe that Suninverse behaves like a typical absorption chiller, i.e. its chilling capacity increases when the driving temperature at desorber increases; while the COP increases until a certain point, after which, it starts to decrease since, the irreversibility raise more rapidly than the benefits.

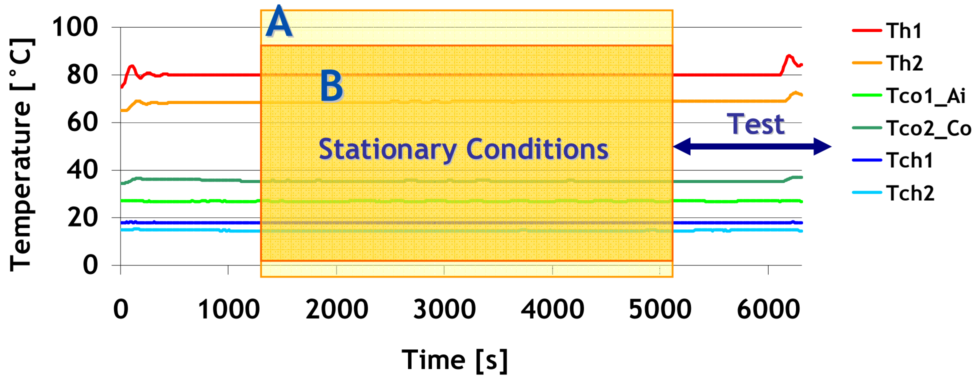


Figure 6-21 Stationary Conditions and actual Test individuated by applying backwards the restrictions for continuous chillers.

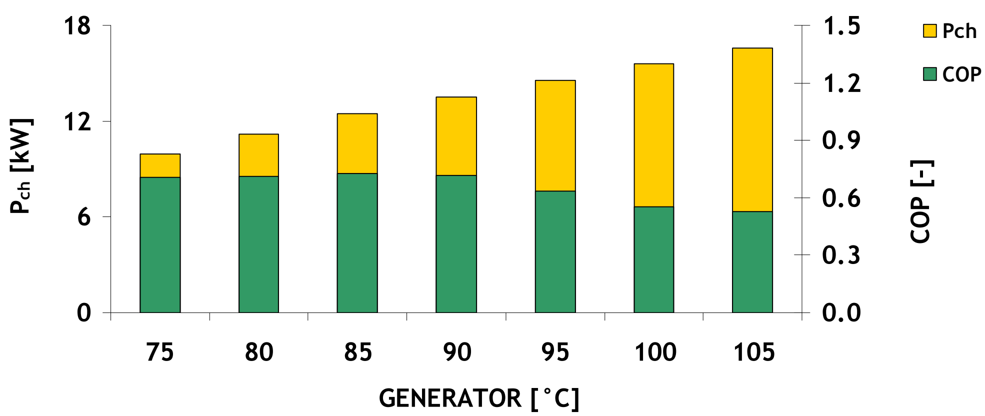


Figure 6-22 Chilling capacity and COP of Suninverse chiller calculated using the selected data.

This can be evicted also from Figure 6-23, in which it is possible to see that, with the increase of driving temperature, the driving heat raises more rapidly that the chilling capacity, resulting in a decrease of COP.

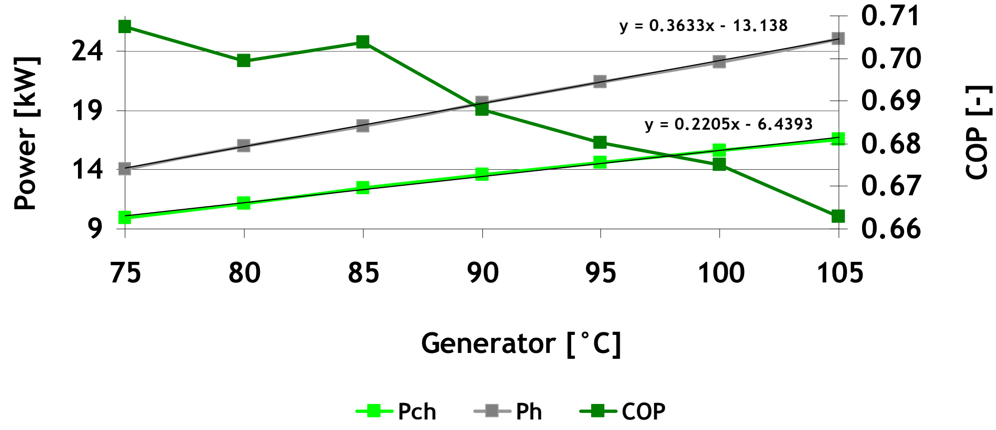


Figure 6-23 Chilling capacity, driving heat and COP vs. time

The electric power consumptions and related COPE have been also estimated. Like for CW10, the calculations have been carried out on three levels:

7. *Machine level*
8. *Circuits level*
9. *Heat Rejection level.*

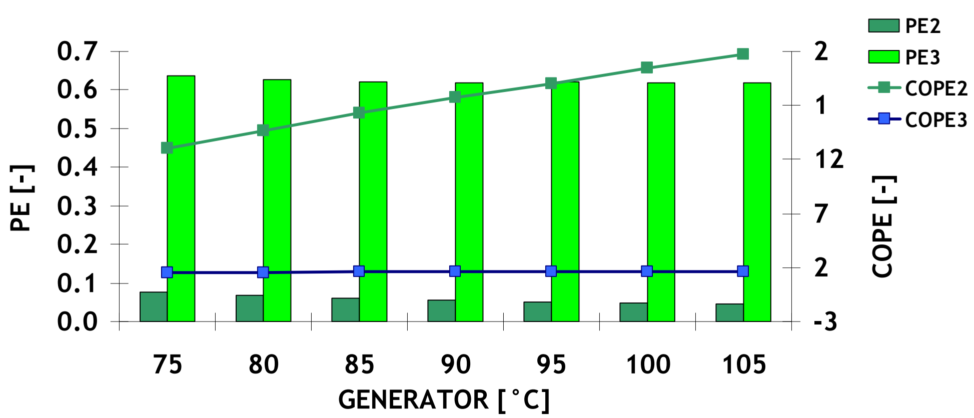


Figure 6-24 Electric Power Consumption and electric COPE vs. different inlet generator temperatures

For the calculation of electric power and COPE at the first level, since TU Berlin didn't supply any data concerning the electric power consumption, that one written in the data sheet of the chiller has been used. It is equal to 120We. For the calculation of electric consumption and COPE at second level, the measured pressure drops and volume flows have been used; while for the calculation of these figures at the third level, 0.07 kW of electric power per kW of heat to reject have been used as reference value consumption of heat rejection system

Also in this case, the achieved results have been normalized with respect to a reference conventional chiller having a COP equal to 3.5. In order to do this, the specific values of electric power consumptions and COPE – i.e. related to 1 kW of chilling capacity – have been calculated at the all three levels and then divided by the specific COP and power consumption of the conventional chiller. The results of these evaluations have been plotted in Figure 6-24.

Analyzing this figure, it is possible to observe that the electric consumptions and COPEs have a trend quite linear with the inlet temperature at desorber. In particular, the electric consumption decreases when the inlet temperature at desorber increases; while the COPE increases when this temperature increases, and this at both levels. This is due to the fact that at high inlet temperatures at desorber the chilling capacity increases and, consequently, the specific electric consumption decreases.

Nevertheless, as it has been seen previously, at high inlet temperatures at desorber, the thermal COP decreases significantly since the irreversibility increase. This results in bigger heat amounts to reject and thus, in higher electric consumptions related to the heat rejection system. This remark is can be view in Figure 7 5, where the COPE calculated at the third level is considerably lower.

6.4.2 Case Study B: ACS 08

6.4.2.1 Functional Scheme

ACS 08 is a small scale adsorption chiller produced by SorTech – see Figure 6-25/a). It is characterized by a semi-continuous operation and has a chilling capacity of 7.5 kW rated at its design conditions – i.e. 72-27-18°C. The working pair is the couple Water–Silica Gel, where the (distilled) water serves always as refrigerant and Silica Gel, which is a porous glass with high capacity of absorbing water vapour, serves as *adsorbent*.

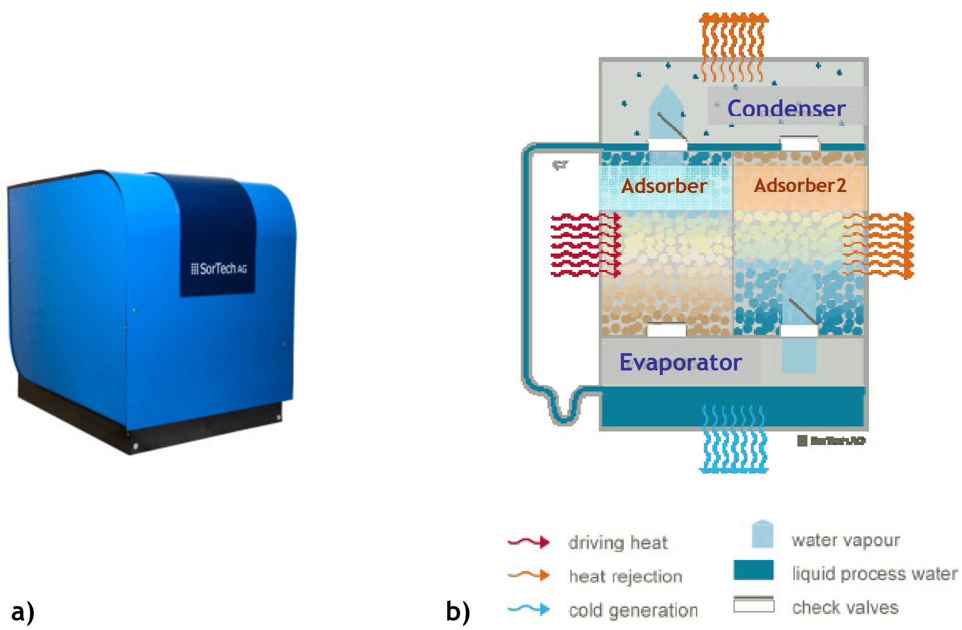


Figure 6-25 ACS 08: a) View of chiller; b) Functional Scheme - Source: SorTech

Internally, the chiller consists of four compartments: evaporator and condenser, and two boxes containing Silica Gel. These last two components exchange periodically the adsorber and desorber function; while the other two are kept fixed. Like CW10, a hydraulic switching unit, driven by an internal controller, provides for connecting the machine with the three external circuits – i.e. heat circuit, cooling circuit and chilled water circuit – by activating hydraulic components involved in the swapping between adsorber and desorber. The operation of machine is characterized by three working steps:

1. Desorption: during this phase, the adsorbent is dried by heat input producing water vapour, which flows in the condenser where it is liquefied under heat emission. When the material is dry enough, the heat input in the adsorber is stopped and the upper check valve closes.

2. Adsorption: after a cool down phase the reverse reaction and the evaporation of the liquid condensate starts. The lower check valve to the evaporator opens and the dry adsorbent aspirates water vapour. In the evaporator, the water evaporates and generates cold, which can be used for air-conditioning. During the adsorption process heat is rejected.

3. Return of condensate: during this final step, the liquid condensate is returned to the evaporator and the circuit is closed. In order to achieve a continuous cold production two adsorber's work in combination, i.e. one adsorber desorbs while the other adsorber generates cold by adsorbing in the meantime.

ACS 08, like CW 10, can work in reverse operation as heat pump connected to low temperature heating systems. Obviously, only its chilling operation is considered, since the heat pumps mode doesn't concern the topic of present work.

6.4.2.2 Tests and Analysis

In this case, the files containing the tests carried out on ACS 08 chiller have been supplied by LESBAT, a research centre located in Switzerland. They are 18 of all and concern tests performed at the rating conditions specified in Table 6-9. Also for them, the stationary conditions and the test itself have been individuated, by using the restrictions related to semi-continuous chillers. Nevertheless, it hasn't been possible to apply them in a proper way since LESBAT chose, as sample time for the data collection, a time equal to 55 seconds, which, not only doesn't allow observing all transitory effects following a swap, but doesn't allow clearly establishing the beginning and the end of the swapping periods (that, for adsorption chillers, is in the order of minutes).

Table 6-9 Rating Conditions of tests carried out on ACS 08 by LESBAT. The symbol "X" indicates the test carried out on the machine.

		Test					
		Condenser [°C]					
		25	27	30	25	27	30
Generator [°C]	65	X	X	X	X	X	X
	75	X	X	X	X	X	X
	85	X	X	X	X	X	X
		15			18		
		Evaporator [°C]					

Despite this inconvenient, it has been tried to individuate for each cycle the Subinterval B, which consists of set of data collected between the end of swapping time and the next swaps, and to apply backwards on it and on the subinterval A some filters according to the prescriptions for semi-continuous chillers. In this way, the cycles concerning the stationary conditions and for the test execution have been identified as shown in Figure 6-26.

The assessment of the chiller performances has been carried out elaborating the selected data trough the equations written in Chapter 4, adjusted for semi-continuous chillers – i.e. considering a number of cycles equal to 4. The achieved results have been listed in Table 6-10 and Table 6-11 and plotted in Figure 6-27 and Figure 6-28.

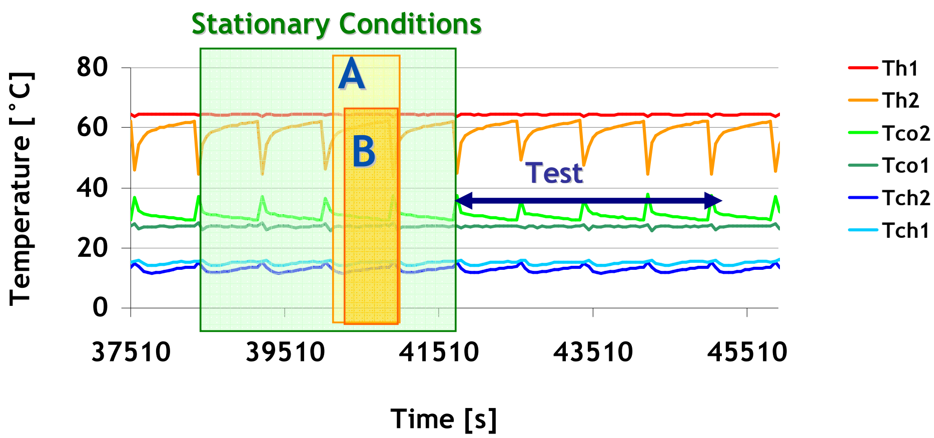


Figure 6-26 Stationary Conditions and actual Test individuated by applying backwards the restrictions for semi-continuous chillers.

Table 6-10 Chilling Capacity calculated at different inlet desorber and condenser temperatures

		Chilling Capacity [kW]					
		Condenser [°C]					
		25	27	30	25	27	30
Generator [°C]	65	5.81	4.92	3.49	6.92	5.90	4.46
	75	5.71	6.34	5.03	8.05	7.54	6.14
	85	8.18	7.35	5.90	8.01	8.10	7.17
		15			18		
		Evaporator [°C]					

Table 6-11 COP calculated at different inlet desorber and condenser temperatures

		COP [kW]					
		Condenser [°C]					
		25	27	30	25	27	30
Generator [°C]	65	0.51	0.49	0.44	0.56	0.54	0.52
	75	0.43	0.47	0.44	0.60	0.51	0.50
	85	0.44	0.43	0.39	0.53	0.52	0.41
		15			18		
		Evaporator [°C]					

Analyzing these two tables, it possible to deduce that the chilling capacity increases when the inlet temperatures at desorber and evaporator increase, and a decreases when the inlet temperature at condenser increases. The COP decreases when the inlet temperature at desorber increases, which means that irreversibility grow more rapidly than the chilling capacity, when the inlet temperature at condenser increases; and decreases when the temperature at evaporator increases. Thus, ACS 08 shows the typical behaviour of adsorption chillers, for which the best working conditions with regard to chilling capacity are: 85°C at desorber, 25÷27 °C at condenser and 18 at evaporator; while concerning the COP, the best working conditions are 75-25-18°C (quite close to its design conditions).

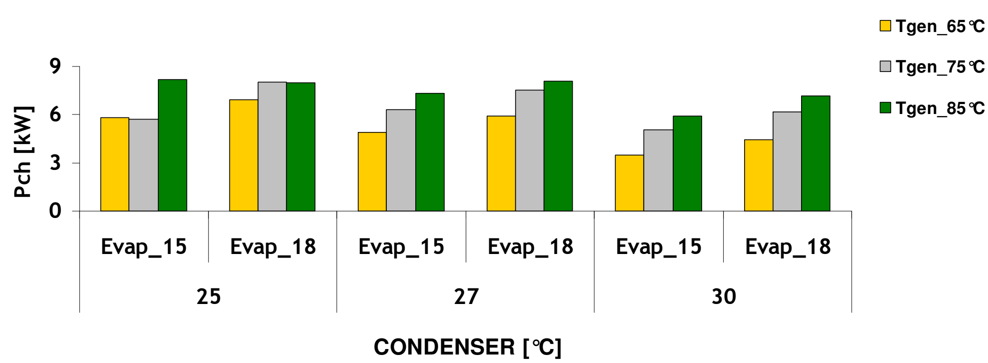


Figure 6-27 Chilling Capacity vs. different inlet temperatures at desorber, condenser and evaporator.

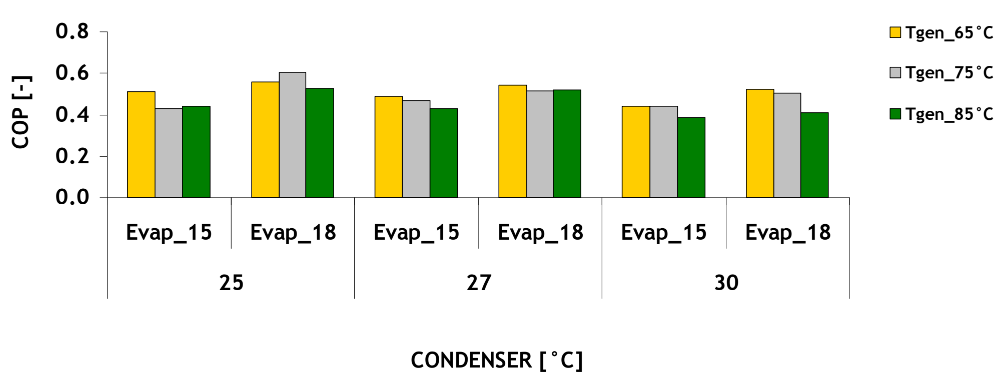


Figure 6-28 COP vs. different inlet temperatures at desorber, condenser and evaporator.

Since from the analysis carried out on CW 10 in the previous chapter, the chiller has showed an anomalous behaviour due to the impact of the swapping time on cycle time, it could be interesting to compare this influence with that of ACS 08. Nevertheless, the sample time chosen by LESBAT doesn't allow calculating this figure for ACS 08 since is to large, so making impossible the comparison. This underlines the importance of having a common test procedure also for collecting data during the test execution.

Also in this case, the chiller has been assessed from the electrical point of view. In particular the electric power consumptions – i.e. PE- and the electrical COP – i.e. COPE – have been evaluated on the three levels: 1) machine; 2) machine plus the pressure drops at the three external circuits; 3) machine, plus circuits plus electric consumptions due to the heat rejection systems. For the calculation of PE and COPE at first level, since LESBAT didn't supply data concerning electrical figures, the electrical power consumption written in the data sheet has been used, which is equal to 7W. While for the calculation of PE and COPE at third level, the value referring to the electrical consumption of Wet Cooling tower available in literature has been used, which is equal to 0.07 kW of the electric power per kW of heat to reject.

The results achieved from the elaborations have been normalized with respect to a reference conventional chiller having a COP equal to 3.5. and plotted in Figure 6-29 and Figure 6-30 related to inlet temperature at evaporator equal to 15°C and 18°C respectively.

Analyzing the graphs, it is possible to observe that both PE2 and PE3 increase when the inlet temperature at condenser increases, and decrease when the inlet temperature at desorber increases .While the COPE2 and COPE3 decrease when the inlet temperature at condenser increases and increase when the inlet temperature at desorber increases. This reflects the conclusion achieved with the analysis made on the thermal figures for which, when the inlet temperatures at desorber increases, the irreversibility increase and so also the heat rejection increases; while when the inlet temperature at condenser increases, even if the machine works worse, the heat rejection decreases and so also the electric consumption decreases.

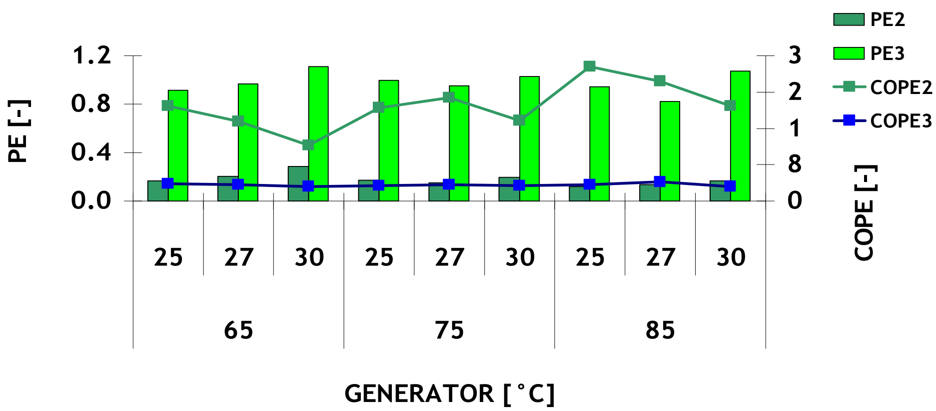


Figure 6-29 Electric Power Consumption and electric COPE vs. different generator and condenser temperatures. The inlet temperature at evaporator is set at 15°C.

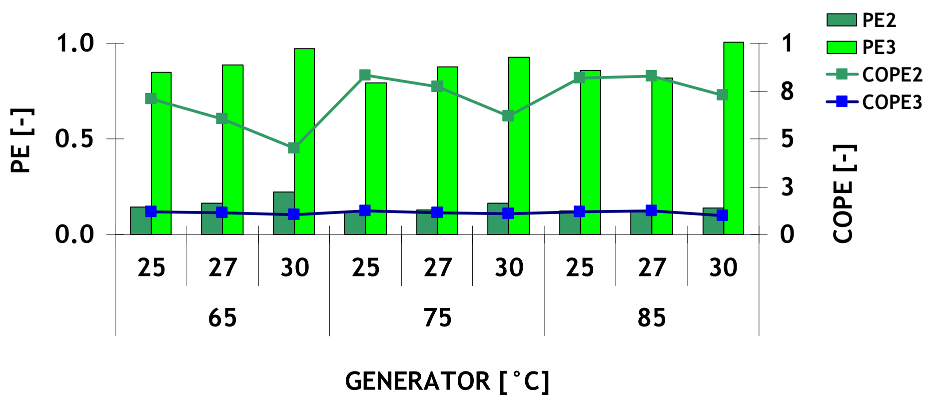


Figure 6-30 Electric Power Consumption and electric COPE vs. different generator and condenser temperatures. The inlet temperature at evaporator is set at 18°C.

6.5 Comparison in Solar Applications

At this point, since the three chillers, i.e. CW10, Suninverse and ACS 08, have been assessed applying the same procedure, the achieved results concerning their performances can be compared.

With this regard, going back to that which represents the most attractive application for these machines, an index capable of evaluating the "quality" of their combination with solar collectors has been ideated. In particular, it has been estimated the incidence that the temperature difference between the inlet and outlet of a sorption chiller desorber, connected directly with a solar plant, has on the collector efficiency. More precisely, the collector efficiency depends on solar irradiation, ambient temperature and on average manifold temperature according to the following equation:

Equation 6-2

$$\eta_{coll} = c_0 - c_1 \frac{T_m - T_{Amb}}{G} - c_2 G \left(\frac{T_m - T_{Amb}}{G} \right)^2$$

Where:

- c_0 is a dimensionless conversion factor
- c_1, c_2 are loss coefficients measured in W/m²K factor
- T_m is the average manifold temperature measured in K
- T_{Amb} is ambient temperature measured in K
- G is the solar Irradiation measured in W/m²

From this equation, it is easy to deduce that, maintaining constant all other equation quantities, the collector efficiency increases when average manifold temperature, T_m, is low.

The average manifold temperature is the arithmetic average of the inlet and outlet temperatures of collector. In case of solar cooling installations, where the desorber of sorption chiller is "connected" with the solar plant, these two temperatures are proportional to those at inlet and outlet of desorber under the condition to not have other components between the chiller and the solar field like, for instance, a water tank.

Taking this last remark as starting point for the evaluation of incidence of the ΔT at chiller desorber on the collector efficiency, an index, termed *Solar Index,* has been defined as follows:

Equation 6-3

$$I_{Solar} = \frac{T_{desorber,in} - T_{desorber,out}}{T_{desorber,in} - T_{Amb}}$$

Where:

- $T_{desorber,in}$ and $T_{desorber,\ out}$ are the inlet and outlet desorber temperatures respectively measured in K [°C]
- T_{Amb} is ambient temperature measured in K [°C]

According to this index, for a fixed $T_{desorber,\ in}$,, at higher $\Delta T_{desorber}$ corresponds higher collector efficiency. This implies that for a certain heat rate required by sorption chiller, $\dot{Q}$, it can be used a lower collector area or a lower collector volume flow. While, for a fixed $\Delta T_{desorber}$, at a lower $T_{desorber,in}$ corresponds higher collector efficiency. This implies that for a certain heat rate required by sorption chiller, $\dot{Q}$, it can be used a lower collector area or a lower collector volume flow.

For the comparison of considered chillers, the *Solar Index* has been calculated assuming an ambient temperature, T_{Amb}, equal to 25°C and using , for each chiller, the estimation of their ΔT at desorber when its inlet temperature is equal to 75°C, 85°C and 95°C respectively. The values have been listed in Table 6-12. In, particular it has been used to compare the chillers at those points – i.e. working conditions – in which their performances are quite close.

With this regard, the COPs and the chilling capacities of each chiller, evaluated at different working conditions have been plotted together in 3-D Graphs, shown respectively in Figure 6-31and in Figure 6-32.

They have been obtained in MATLAB by the interpolation of the available data and for this subject to approximations especially, for the case Suninverse, for which the test data are few.

Table 6-12 Solar Index evaluated for each of considered chillers at different inlet desorber temperatures

		Solar Index [-]					
		SK		*CW*		*ACS08*	
		ΔT	Isolar	ΔT	Isolar	ΔT	Isolar
GENERATOR [°C]	75	10.07	0.19	9.12	0.17	7.03	0.13
	85	12.71	0.20	13.09	0.20	7.88	0.12
	95	15.38	0.21	15.58	0.21		

To understand how use the defined index, the intersection points of the plotted surfaces have to be taken into consideration. For instance, in figure representing the COP as function of inlet temperatures at generator and condenser, the intersection point between CW10 and ACS 08 can be considered. It occurs at 85°C at generator and 30 °C at condenser. In this case, according to the solar index, the combination of CW10 with solar collector is to prefer respect to that one with ACS08. In this way, the considerations have been done only on the base of COP; to have a more complete vision, the chilling capacity of chiller has to be verified at the same working conditions.

The same can be done for the intersection points at 75°C of the chilling capacity of CW10 and Suninverse for which, according to the solar index, is preferable to use this last chiller, i.e. Suninverse, for solar applications.

Nevertheless has to be précised that the available data are not sufficient for doing this analysis accurately and that, in the real cases, for the chillers selection many other parameters have to be taken into account.

The intent of this index is, in fact, to evaluate the quality of the combination between a specific chiller and a solar collector field meant as exploitability of the solar source based only on the temperature levels.

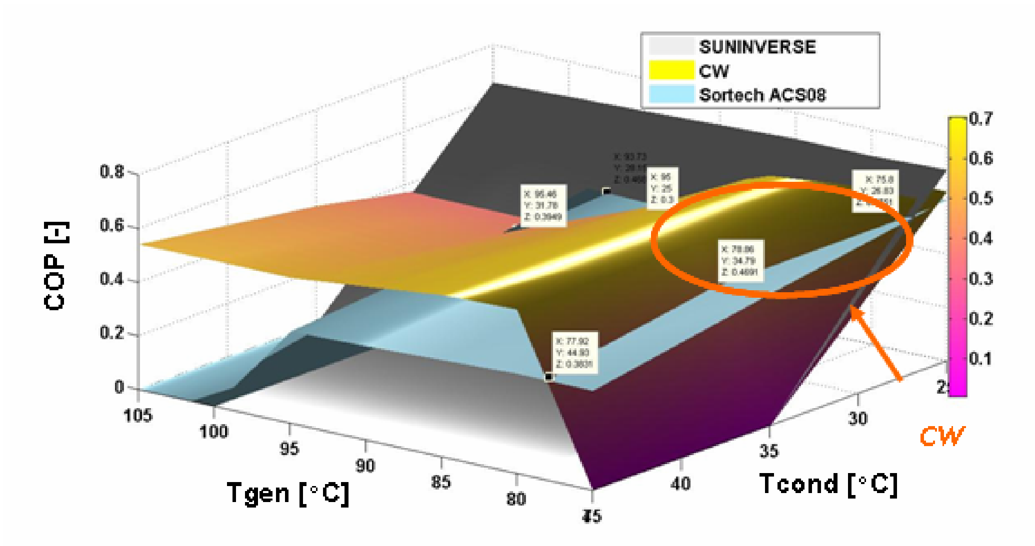


Figure 6-31 COPs of Suninverse, CW10 and ACS 08 vs. desorber and condenser Inlet temperatures

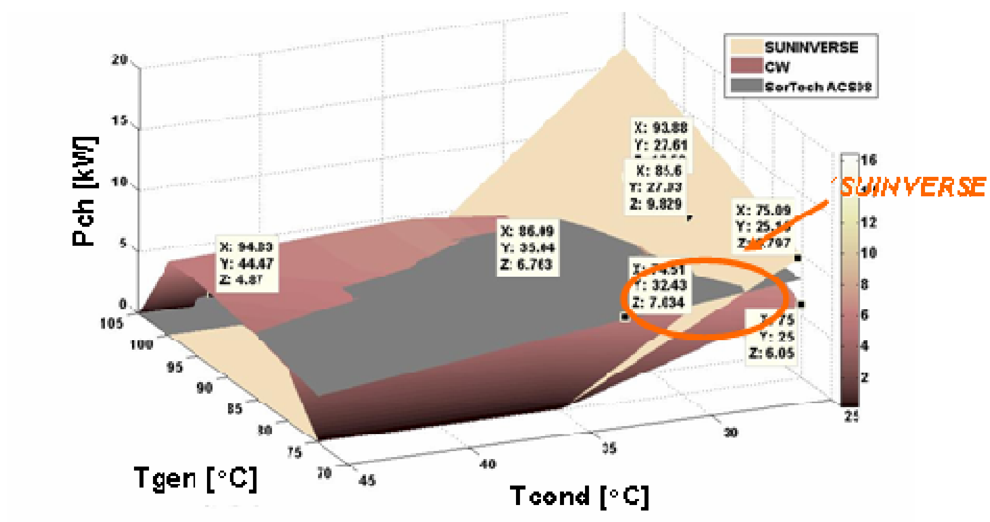


Figure 6-32 Chilling Capacity of Suninverse, CW10 and ACS 08 vs. desorber and condenser Inlet temperatures

1 Conclusion

Up to now, thermally driven chillers have been classified only on the basis of the number of effects (single or double effect), driving technology (direct or indirect fired) and sorbent physical properties (liquid: absorption; solid: adsorption); the few dedicated existing standards for their testing and evaluation are centred on this classification. The introduction of new technologies on the market has brought, in many cases, to the inapplicability of standards’ prescriptions and to the inadequacy of above-mentioned parameters used for their classification.

For this reason, in the present work, a new way to classify thermally driven chillers based on their working mode, i.e. continuous, semi-continuous and batch mode, has been identified and a new test procedure has been developed.

To validate the test procedure, an experimental facility was designed and installed at EURAC Research in Bolzano, suitable for testing sorption chillers with nominal cooling capacity up to 20 kW. A prototype of ClimateWell CW10 was used to the purpose, which was extensively tested in the laboratory. With regard to this chiller (operating internally in steady condition even when boundary conditions are stationary), it has been proved the inadequacy of the formulations and the prescriptions reported in the existing standards; new one and more general have been defined. The latter were used to assess the performance of the mentioned chiller, which have been compared with the ones relative to two other machines tested by partner research institutes.

The test procedure developed takes into account stationary conditions with regard to the temperatures and mass flows at the inlet of the machine. This is not the case if Solar Combi Plus systems are regarded though: in this application the inlet temperatures are continuously varying, mainly as a function of the sun radiation and the ambient temperature. The performance of the chiller is also strongly affected by those fluctuations, due to their notable thermal inertia. Therefore, further test procedures should be developed in the future for chillers to be used in Solar Combi

Plus applications. Unsteady, reality-like boundary conditions should be considered that reliably simulate meteorological conditions of representative locations placed on the market addressed.

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Appendix A

Table Appendix A - 1 Heat Transfer Medium for heat pumps

HEAT TRANSFER MEDIUM		
EVAPORATOR	*CONDENSER*	*CLASSIFICATION*
Air	Air	Air/Air heat pump
Water	Air	Water/Air heat pump
Brine	Air	Brine/Air heat pump
Air	Water	Air/Water heat pump
Water	Water	Water/Water heat pump
Brine	Water	Brine/Water heat pump

Appendix A - 2 Temperature levels in Standard Rating Conditions for Heating Mode

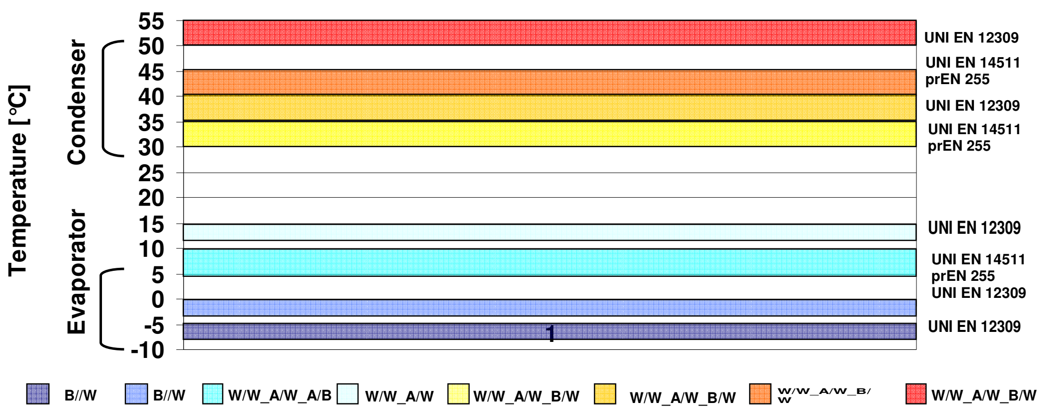

Table Appendix A - 2 Standard Rating Conditions for units working in heating mode

Standard Rating Conditions- Heating Mode												
	EN 14511-2 /prEN 255			UNI EN 12309-2								
Absorber/Condenser:	W/W	B/W	A/W	W/W			B/W			A/W		
				T1	T2	T3	T1	T2	T3	T1	T2	T3
Entering Temperature	40	40	40	50	35	50	50	35	50	50	35	50
(for floor heating or similar application)	**30**	**30**	**30**	-	-	-	-	-	-	-	-	-
Leaving Temperature	45	45	45	NS	NS	NS	NS	NS	NS	NS	NS	NS
(for floor heating or similar application)	**35**	**35**	**35**	-	-	-	-	-	-	-	-	-
Water Flow Rate	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS
Water-Side Fouling Factor	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS
Evaporator:												
Entering Temperature	10	0	7(6)	10	10	15	0	0	-5	7	2	15
(for floor heating or similar application)	**10**	**0**	**7(6)**	-	-	-	-	-	-	-	-	-
Leaving Temperature	7	-3	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS
(for floor heating or similar application)	**7**	**-3**	-	-	-	-	-	-	-	-	-	-
Water Flow Rate	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS
Water-Side Fouling Factor	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS	NS
Environmental Conditions:												
Dry Bulb Temperature	15°C to 30°C			15°C to 30°C heated area 0°C to 7°C not heated area								
Electrical Power Supply	NS			NS								

Legend
W/W= Water to Water
B/W = Brine to Water
W/B= Water to Brine
A/W =Air to Water
A/B =Air to Brine
NS = Not Stated