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Solar Air Conditioning Potential for Offices in Lisbon

by

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ABSTRACT

The first section contains a critical review of solar cooling, focusing on commercially available technologies: absorption, adsorption and desiccant cooling. It is then followed by two case studies, an adsorption cooling example in Germany and a desiccant-cooling example in Portugal.

The second section consists of a comparison between adsorption and absorption cooling plants against a conventional vapour compression plant, using an office building in Lisbon as background.

The second section also describes a method to size the main components of a plant with solar heat driven chillers. It includes a modelling tool, developed as a spreadsheet that simulates the match between the cooling demand and the solar availability in a year, using hourly time steps.

The energy results from the model shows the technology is technically viable and the results are in line with what is found in literature; however the financial analysis indicates worse returns of investment than what is found in some case studies and reference publications. This is due to the high sensitivity of the model with energy costs and also uncertainty estimating the capital costs, reflecting the actual difficulties in finding accurate references from equipment suppliers.

The final results indicate absorption slightly more competitive than adsorption and more secure and easy to implement.

Keywords: Solar, Absorption, Adsorption, Desiccant, Air-conditioning, Offices, Lisbon, Lisboa, Portugal

I INTRODUCTION

This document covers solar assisted air conditioning technologies, and in particular its applicability for offices in the region of Lisbon. This is done by setting a comparison between conventional and solar assisted air-conditioning, covering its energetic, environmental and economical aspects.

The report is organized into two sections. The first section starts with a critical review on existing solar assisted air conditioning technologies. This critical review is written in an approachable manner, and it is intended to give the reader an introductory yet technical background on the subject. It is followed by two relevant case studies.

The next section consists in a study for the comparison of two solar air conditioning plants, specifically solar adsorption cooling and solar absorption cooling. These options are also compared against a conventional compression cooling plant.

This second section is divided into four chapters. Chapter III describes the demand side of the system, i.e. it describes the office building chosen as background for the study and the TRNSYS model used to simulate its annual thermal behaviour. This chapter also sets general considerations on the final cooling demand profile, and how it relates with the selected HVAC system (distribution sub-system).

Chapter IV describes technical aspects related with the model created for the purpose of this study. This model matches the building cooling demand with the solar plant supply, breaking down the solar generated and backup cooling.

Chapter V describes the methodology used for sizing the main equipments taking into account the energy savings and economics of the plant. It also describes the three options, presents the results obtained from the model, and adds general considerations on it, identifying possible future work.

Finally Chapter VI presents the results obtained and concludes the study, adding some general considerations on solar air-conditioning and its applicability in Lisbon. It is again written in an approachable manner, in the hope that this thesis may be more broadly disseminated.

II CRITICAL REVIEW

II.1 background

Solar assisted air conditioning is one of the only renewable energies applications where supply meets demand.

While some renewable technologies such as wind are fairly unpredictable, and other technologies such as natural ventilation, passive solar or solar hot water have their highest seasonal efficiency in periods they are not particularly required, solar air conditioning appears highly attractive since more cooling is needed when the solar intensity is strong and ambient temperatures are high.

In Mediterranean countries solar gains are considerable even during transitional seasons. Portugal is the European country with more hours of Sun, ranging between 2200 to 3000 hours per year [1]. It is also a country that imports about 85% of the energy it consumes, totalling over 4 billion Euros [2], and there should be a strategic interest in alleviating this dependency by moving to endogenous renewable sources.

Solar air conditioning will not solve the national problems on energy, but it could bring opportunities for local development in at least two industries that are well consolidated in Portugal: refrigeration and solar water heating. It would also bring some innovation to the construction industry.

There is a notion that the present cost of these equipments is still prohibitive, not only for the price of the solar cooling equipments and solar collectors themselves, but also for the cost of required backup systems. In fact, even though they cannot compete with conventional technologies, it is often forgotten the indirect benefits in terms of image and marketing this technology can bring. The concept itself is highly marketable – to generate cooling with the Sun.

II.2 existing solar cooling technologies

Cooling can be obtained from many different ways. The majority of them have a direct or indirect origin in the sun, such as fossil fuels. The expression “solar cooling” is usually restricted to when the solar radiation is the direct agent of cooling.

Figure II-1 [3] overviews the direct methods of solar cooling:

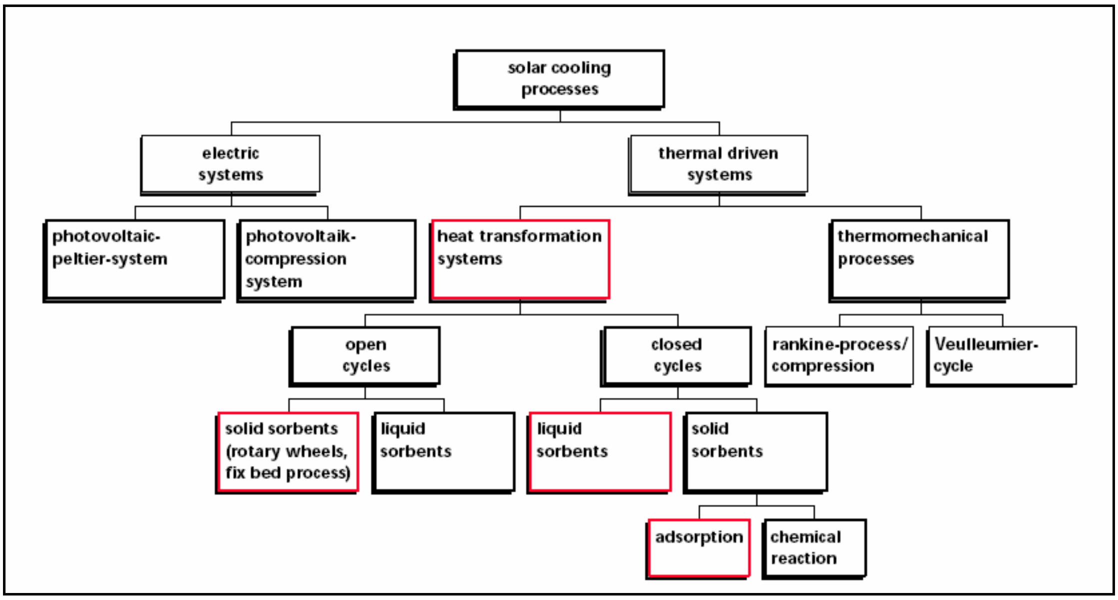


Figure II-1 overview of solar cooling methods [3]

The two main categories are the electric systems and the thermal driven systems.

The electric systems' category require the use of photovoltaic panels, and are not included in this study due to the high costs and low efficiencies, characteristic of these equipments.

Henning [3] divides the thermal driven systems into thermomechanical processes and heat transformation systems.

The thermomechanical processes are very experimental. An example is the rankine process compressor, where the cooling is obtained conventionally through a compressor. This compressor instead of being driven by electricity or a reciprocating engine is driven by a rankine cycle using pressurized vapour from solar concentrators. These clearly aren't technologies suited for mainstream applications and are out of the purpose of this study.

The heat transformation processes covers all the methods for generating cooling from a heat source. Its applicability in solar air conditioning is obviously dependent on a solar heat production.

The heat transformation processes are divided into open and close cycle processes.

The close cycle may use liquid or solid sorbents. If liquid, the sorbent will "dilute" the refrigerant and move from different compartments (absorption). If solid, the sorbent can't move, and its compartment will be cyclically subjected to heating and cooling in order to adsorb the refrigerant and then regenerate.

Absorption and Adsorption have an enormous potential to be used both in new build and in retrofit since the cooling media produced is chilled water, compatible with the majority of the existing air conditioning equipment. A more in depth study of these technologies is presented ahead.

The open cycles are in contact with the atmosphere and use water as refrigerant. The most frequent and commercially available technology is the open cycle solid sorbent, where the air is forced to pass in a rotating wheel (Figure II-2), typically split between the flow and the return of an air-handling unit. The "outdoor air" is dehumidified by action of silica gel (the solid sorbent), laid in the internal wall of the wheel, and then cooled adiabatically. The wheel is regenerated (the silica gel is released of its water content) with the return air that is heated by solar action. This technology is commonly called desiccant cooling, and is presented again and more detailed in the next sub-chapter.

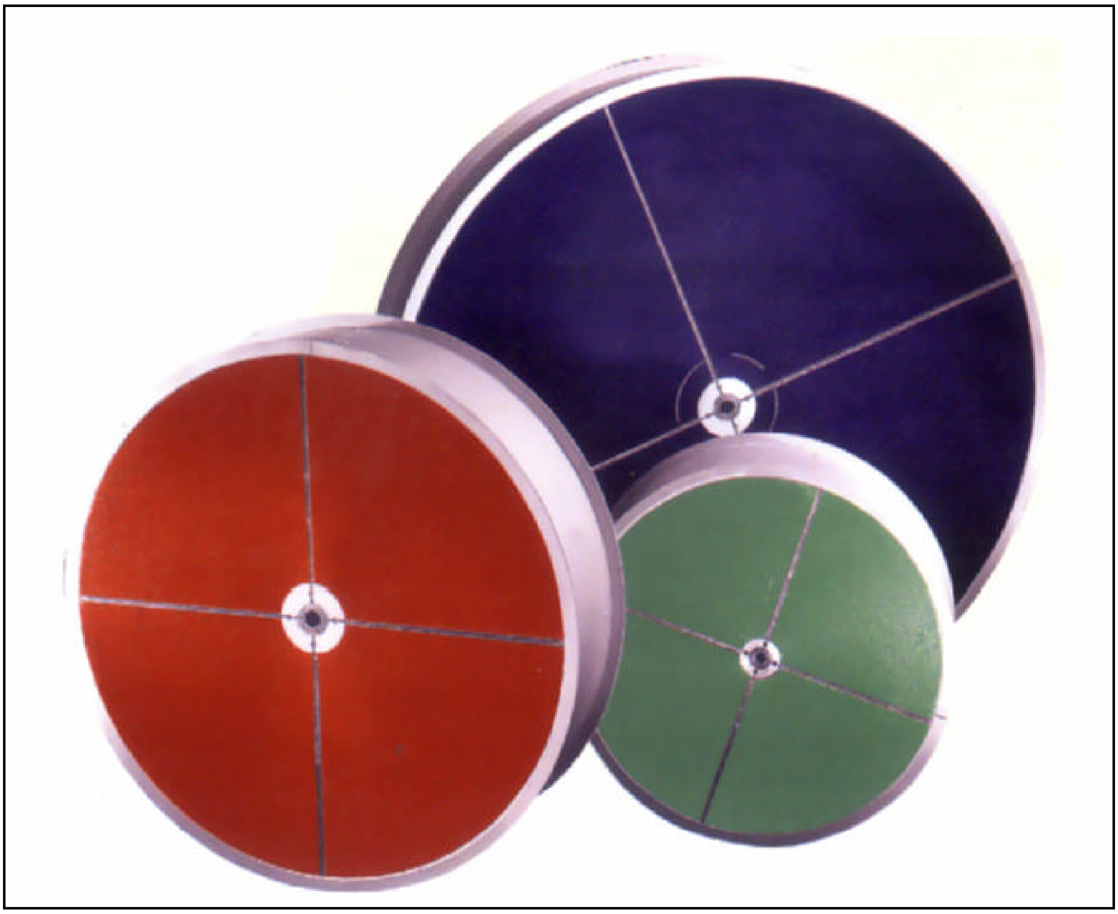


Figure II-2 desiccant wheel [4]

All the referred cooling technologies are driven by heat. In a solar air conditioning plant, this heat is obviously generated from a solar installation.

Solar air conditioning plants are divided in two sub-systems, the cold production sub-system and the heat production sub-system. In the next two sub-chapters, the most commonly available options for these sub-systems are described in more depth, along with an introductory description to compression cooling, used for reference in this project.

II.3 the cooling sub-system

compression cooling

This is the most common way of producing cooling, and can be found in fridges, residential and commercial air-conditioning units (chillers) and industrial refrigeration equipment.

Figure II-3 describes a vapour refrigeration cycle. The refrigerant evaporates at a low temperature in the evaporator.

This evaporation process requires heat that is taken from the evaporator, thus producing the “coolth”.

The vaporised refrigerant is then compressed from a low temperature and pressure in the evaporator to a high temperature and pressure in the condenser. The reverse process from the evaporator happens, and the refrigerant is condensed releasing heat that must be removed and dissipated.

An expansion valve closes the cycle by imposing a pressure drop on the refrigerant down to the pressure of the evaporator.

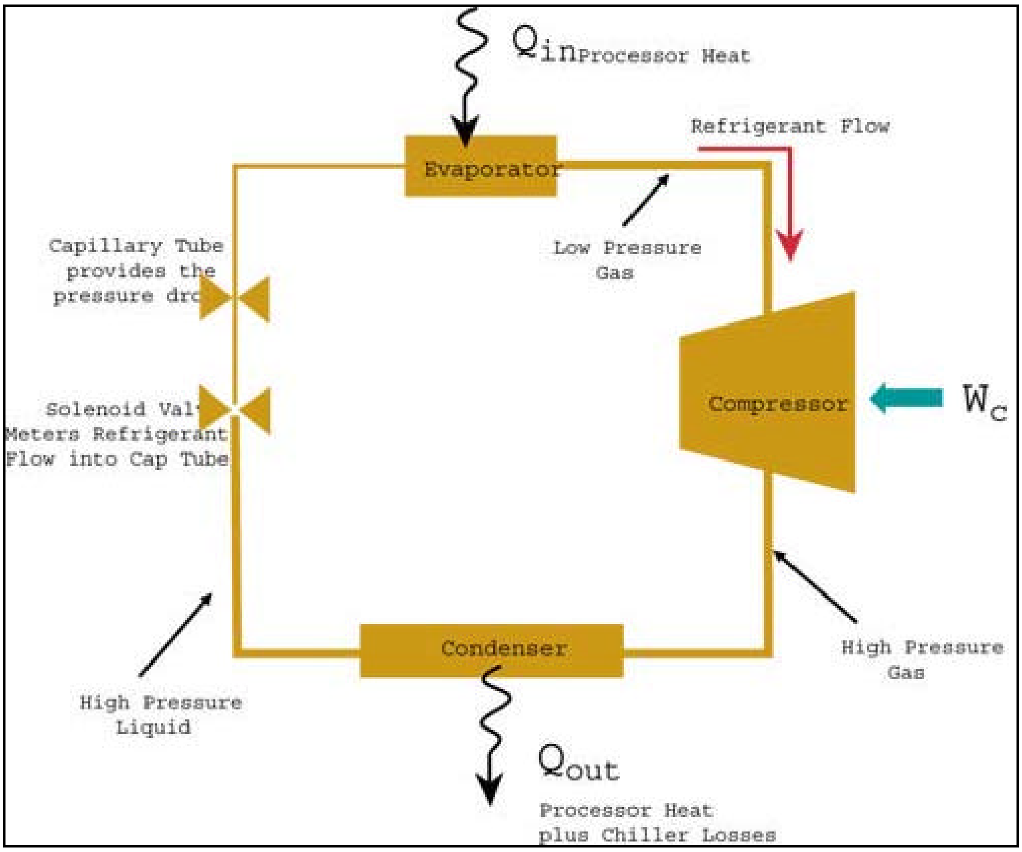


Figure II-3 compression refrigeration cycle [5]

The compressor is usually driven by an electric motor, but in large capacity chillers, sometime gas reciprocating engines are used.

The ratio between the cooling capacity (Q_c) and the electric power input (P_{el}) is called "coefficient of performance" (COP) and is a measure of the efficiency of the equipment:

$$COP = \frac{Q_C}{P_{el}}$$

The COP is dependent of several factors. The most important are:

- the layout of the components from the chiller itself (e.g. how well the evaporator and condenser are insulated from each other, thus preventing heat "short-circuits")
- the heat exchange capacity of the evaporator and the condenser

- the efficiency of the compressor.

A way to enhance the COP of a chiller is to low the temperature of the media that receives the waste heat from the condenser, thus increasing the heat transfer rate in the condenser. This is the philosophy behind ground source cooling, where the thermal inertia of the earth provides a steady and mild temperature throughout the year, avoiding the (bad) dissipation of the condenser's waste heat in hot summer days.

absorption cooling

Absorption chillers offer an exciting alternative to conventional compression chillers, since their main energy input is heat instead of mechanical power.

Absorption chillers come in a large variety of models and types, frequently requiring heat in the form of water vapour, or even direct fire. Applications with vapour frequently fit most waste heat rejection found in industrial processes such as CHP.

The COP is defined as in compression chillers but with the driving heat input (Q_{in}) replacing the mechanical/electrical input (P_{el}).

Traditionally absorption chillers are large machines with a large cooling capacity. However there is a growing trend in the market for low capacity absorption chillers. It is now possible to find small absorption chillers driven by hot water at temperatures that can be supplied by solar collectors.

Absorption chillers work much like conventional compression chillers, except that there is no mechanical compressor. Instead, the vaporised refrigerant leaves the evaporator to the absorber where is diluted by a solution (in the example of Figure II-4, the absorbent solution is Lithium Bromide).

The liquid solution is then pumped (pumps are more efficient than compressors), and then regenerated with heat (in the generator), so that the refrigerant is vaporised again, at a higher pressure and temperature.

It then goes to the condenser to release the contained waste heat.

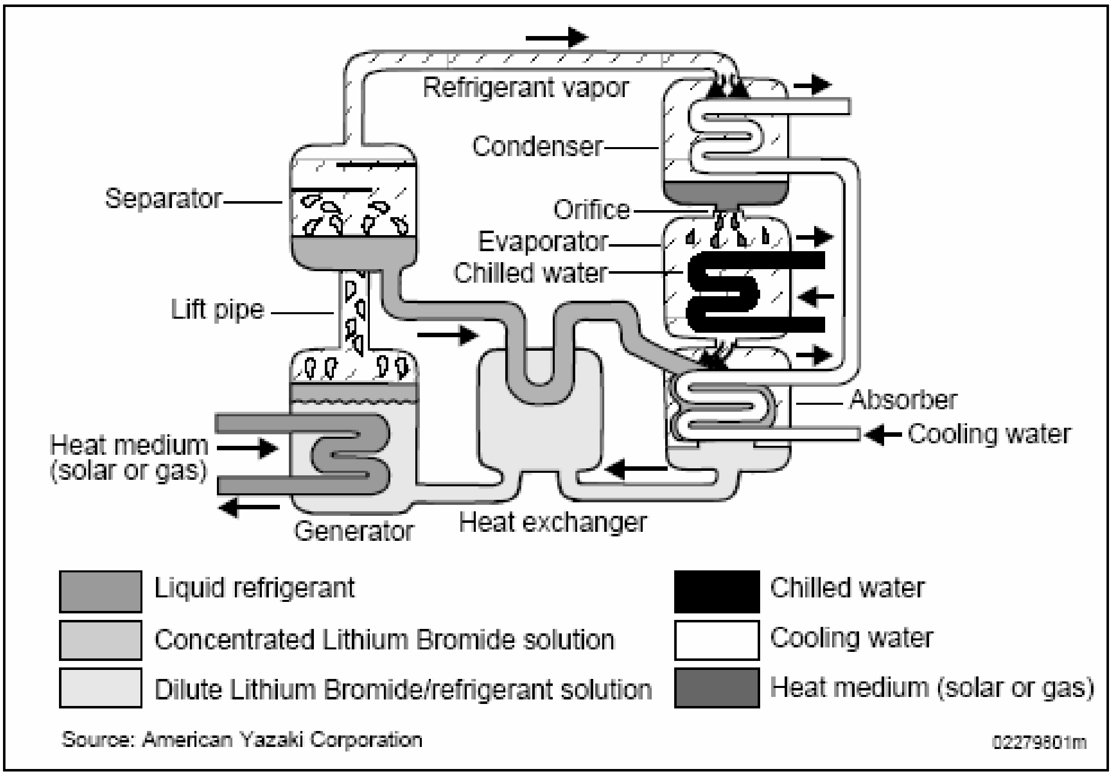


Figure II-4 absorption cycle [6]

Typically for chilled water temperatures less than 5ºC, ammonia is used as the refrigerant and water as the absorber. For typical air-conditioning applications (chilled water above 5ºC), the combination of water as refrigerant and lithium bromide (LiBr) as absorbent is more popular, while chillers using Ammonia as absorbent are more suited for industrial refrigeration, producing chilled water down to –10ºC [7].

The condenser's heat rejection is again critical to the COP of the chiller. On absorption machines, this heat is usually dissipated using low temperature water as media (between 30°C and 50°C), closed in a secondary loop with dry coolers, condensing towers or other.

A frequent problem in LiBr chillers is its crystallization due to excess temperature in the absorber. This may be caused by excess load, but more likely by the malfunction of external heat dissipation. In one hand, absorption chillers require a lower maintenance since they don't have compressors, but in the other, their controls need to be more sophisticated due to the security systems required (the crystallization may destroy the chiller).

Again the COP will be dependant on several factors, such as:

- Temperature of the chilled water generated – the COP increases with temperature, but it starts to be more constant after a certain temperature, as can be assessed in Figure II-5, taken from the specifications catalogue of an absorption chiller (York, [8])

- Temperature of the rejected heat water – dependant on the heat dissipation system conditions

- Temperature of the regeneration hot water – the COP drops significantly when the temperature gets lower. Typical design temperature of hot water chillers round the 80°C – 90°C, yet it is possible to low to 75ºC sacrificing the COP.

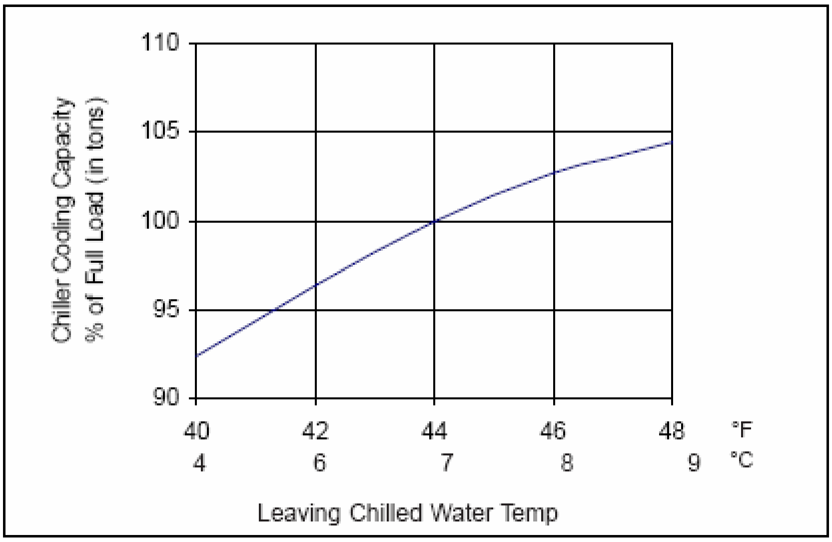


Figure II-5 example of variation on chiller capacity with chilled water temperature [8]

Double-effect chillers and even three and four effect chillers have a higher COP. Yet they are not considered for solar cooling since the driving temperatures lie in the range of 140°C to 160°C.

adsorption

Solar cooling technologies recovered adsorption chillers. From a "black box" standpoint, these chillers are similar to absorption chillers, since are both driven by heat. However, adsorption chillers can be driven by hot water at lower temperatures than absorption chillers, thus benefiting from a better efficiency on the solar collectors system because they are generating water at a lower temperature.

From a physical point of view, both technologies differ significantly since the sorbent used in adsorption is silica gel, a solid that cannot be compressed or pumped. Instead, each compartment containing the solid sorbent is alternately heated and cooled to adsorb and desorb the refrigerant in a periodic process.

An adsorption chiller consists of two compartments on which the internal surfaces are covered with silica gel; a highly porous solid that captures water vapour (adsorbs the refrigerant). The same compartment is then heated (regenerated) with the driving hot water at temperatures ranging from 55℃ to 95°C. The refrigerant at a higher temperature moves to the condenser where is condensed, resulting waste heat that must be dissipated. A throttle valve drops the pressure of the condensed water to the level of the evaporator. At that low pressure, it receives enthalpy from the chilled water and evaporates, moving on to the other compartment, with the regenerated silica gel, thus completing the process.

A cycle can take around 7 minutes [9] and it runs as follows (please refer to

Figure II-6):

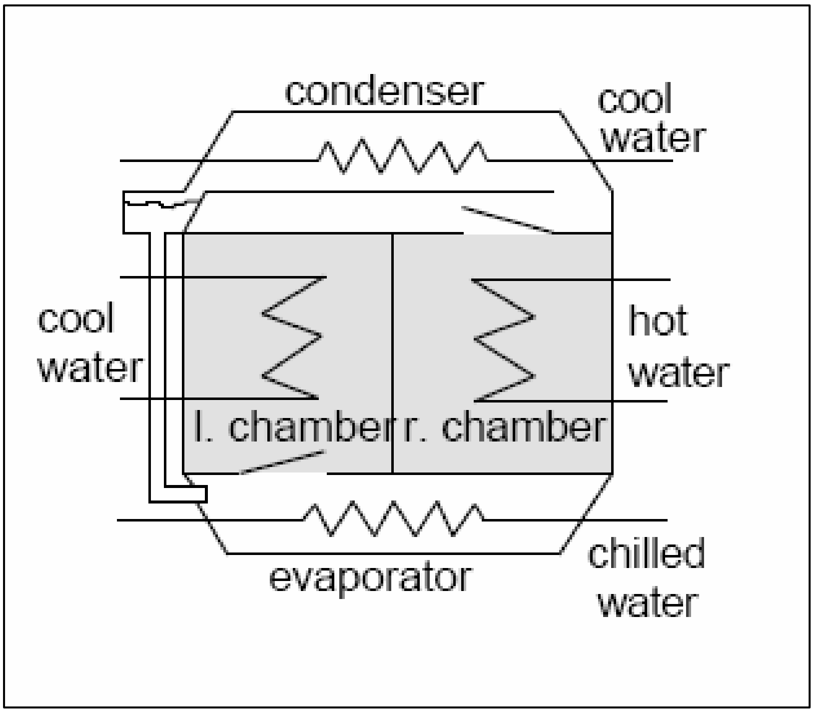


Figure II-6 adsorption chiller schematic [10]

1. The refrigerant goes to the evaporator, where is evaporated with strong vacuum, and produces chilled water.
2. The refrigerant moves to one of the compartments filled with recently regenerated silica gel, where it is adsorbed.
3. The cool/ hot water cycle inverts. Now heat is being supplied to the compartment, regenerating the silica gel. Back as in vapour form, the refrigerant is pressurized and goes to the condenser.
4. In the condenser, it condensates releasing waste heat (to be dissipated). The liquid refrigerant is then sprayed back to the evaporator completing the cycle.

There are few manufacturers of adsorption chillers. At the time of the writing of this report, only two Japanese companies Mycom (Figure II-7) and Nishiyodo manufactured adsorption chillers, with cooling capacities ranging approximately from 70 to 350kW [14].

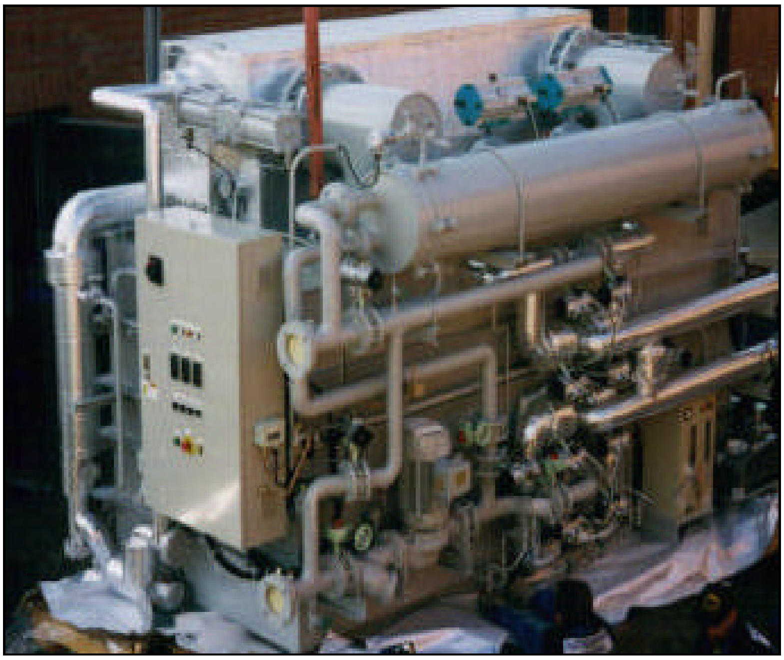


Figure II-7 Mayekawa Mycom ADR30 [10]

As opposed to absorption chillers, it is possible to operate with lower temperature limits without the danger of the sorbent's crystallisation. This allows an operation at temperatures down to 55ºC considerably lower than LiBr/H_2O chillers, bringing also a more stable operational behaviour.

The negative aspects are the higher price and a larger construction volume and mass.

desiccant cooling

In desiccant cooling, the cooling medium is air instead of water. It is usually associated with a double deck air-handling unit, where the inlet air is dehumidified via a desiccant rotating wheel that adsorbs the water, again in silica gel laid in the internal walls of the wheel. The dehumidified inlet air is consequently cooled via a cooling coil, a heat recovery unit or adiabatic cooling system, to a lower temperature close to saturation. Meanwhile the return air from the building is heated through a heating coil (with e.g. solar hot water ranging from 50ºC to 100ºC) regenerating the rotating (6-12 rpm) desiccant wheel. Figure II-8 [X] shows the schematic of a desiccant cooling system, where the water is cooled with a heat recovery unit (regenerator), and direct and indirect adiabatic cooling (evaporative air coolers).

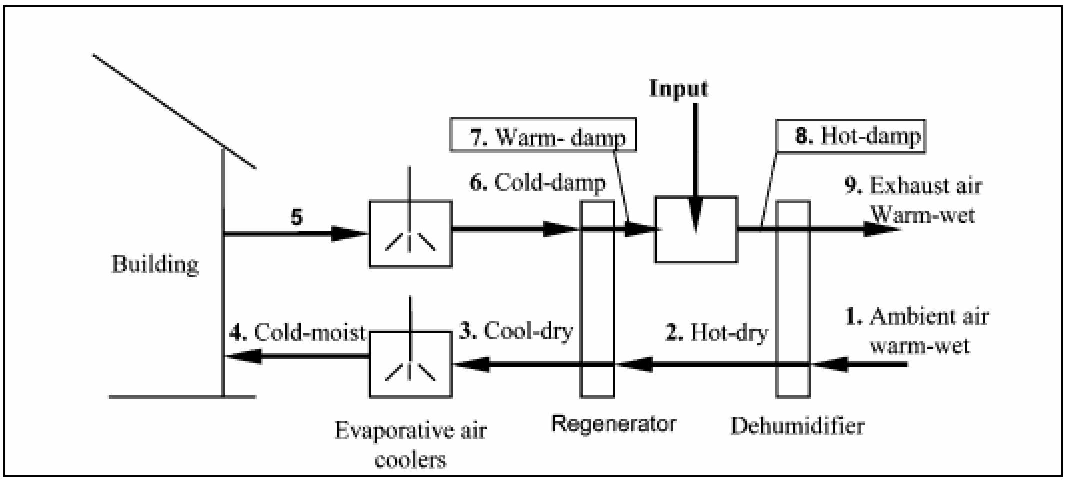


Figure II-8 schematic of desiccant cooling [11]

Figure II-9 shows an example where the cooling is obtained only with the heat exchanger rotor.

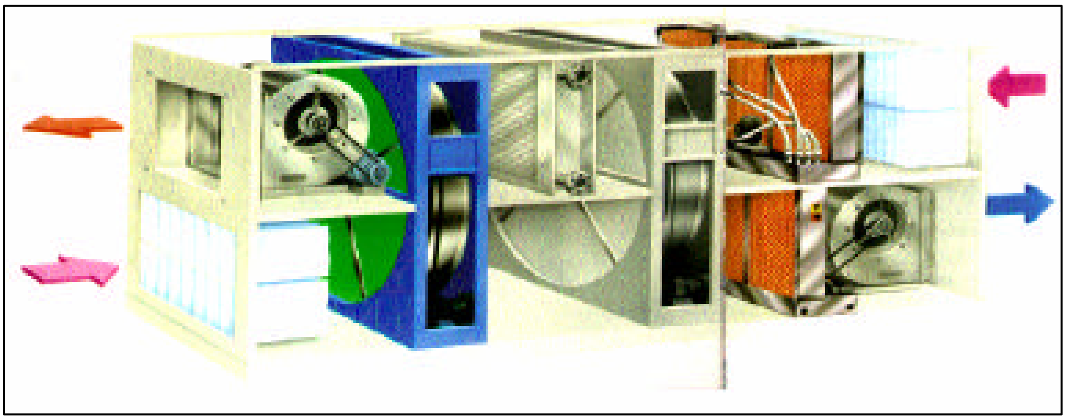


Figure II-9 air handling unit with desiccant wheel and heat exchanger rotor [3]

The COP for desiccant cooling systems is defined by:

$$COP = \frac{\dot{m}\left(h_{in} - h_{out}\right)}{\dot{Q}_{reg}}$$

Like in thermally driven chillers it consists in a ratio between the cooling power and the heat supplied to the system. However, because the cooling agent is air, the cooling capacity is calculated by the difference of enthalpy times the mass flow rate.

These systems are perhaps the lowest cost solar cooling technologies. However, and for warm climates such as the Portuguese, it cannot provide on its own the required cooling power for buildings.

Because the cooling agent is air, with a lower Cp than water, it would require very large ducts for distributing the required quantities of cooling. This solution would be unviable due to the space constraints on the HVAC of buildings.

II.4 heat production sub-system

introduction

In solar air-conditioning, all processes involve some sort of regeneration process that is done with solar heat.

For close cycle processes (absorption and adsorption), the ideal medium is through solar hot water.

Desiccant cooling can also use solar hot water, but it can also be regenerated with hot air heated by an air solar collector. It is interesting to note that outside air may be used for regeneration of the desiccant wheel, rather than the return air. This can be particularly interesting for Lisbon, where the ambient air temperatures are frequently above the traditional return of T26ºC/RH50%.

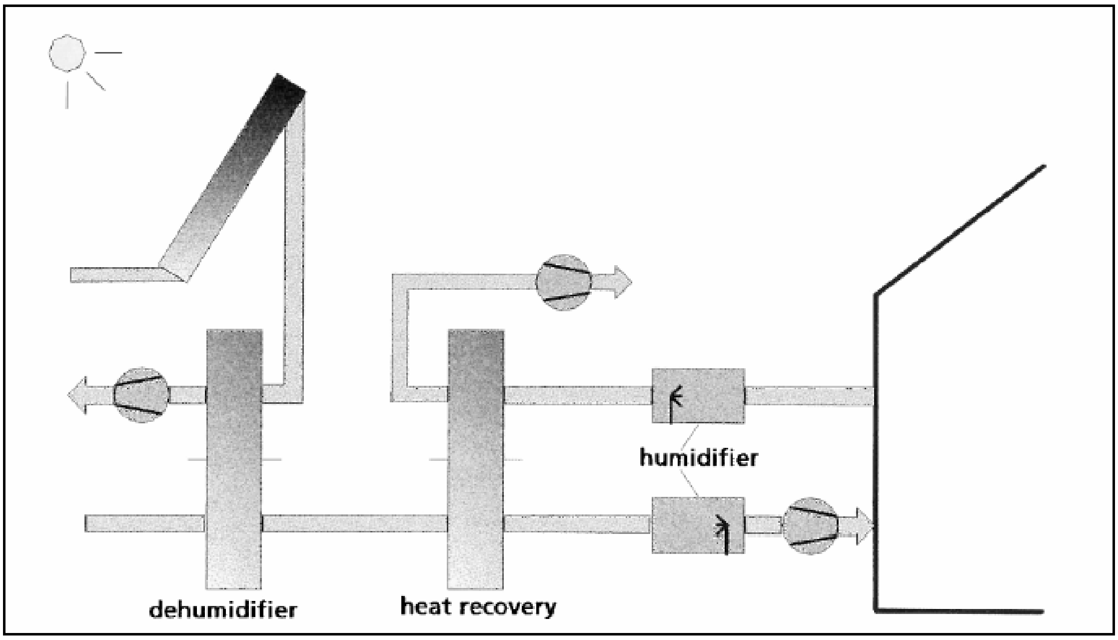


Figure II-10 schematic example of desiccant cooling system using outside air and air solar collectors [12]

solar collectors

The collectors' efficiency drops as the working temperature gets higher. To work at the temperatures required for solar air-conditioning, the collector' heat losses must be as small as possible.

Due to the temperatures involved, solar collectors' efficiency should not be calculated by a normal linear relation, but by a polynomial expression of 2nd order to account with the losses by radiation to the sky. It is used the expression [9]:

$$h = k(\Theta) \cdot c_0 - \frac{(T_{av} - T_{amb})}{G_p} \cdot c_1 - \frac{(T_{av} - T_{amb})^2}{G_p} \cdot c_2$$

Where k(T) is the incident angle modifier, T_{av} is the average fluid temperature, T_{amb} is the ambient air temperature, and G_p the total global radiation incident on the collector surface. The remaining constants are indicated in the following Table II-1, which presents diverse typical parameters for the most usual types of collectors:

	Evacuated tube collector 1	Evacuated tube collector 2	Flat plate collector 1	Flat plate collector 2	Stationary CPC collector	Roof integrated collector
Gross area	1.181	2.762	2.567	2.612	2.69	6.082
C0	0.612	0.601	0.690	0.696	0.556	0.704
C1	0.54	1.44	2.61	3.26	1.30	3.03
C2	0.0017	0.0033	0.0098	0.0062	0.0195	0.0096
? (?T=50K)	57.3 %	50.0 %	49.7 %	47.3 %	41.3 %	48.5 %
? (?T=70K)	55.4 %	45.4 %	40.3 %	37.4 %	32.2 %	38.0 %
€/m^2	643	555	234	234	223	175

Table II-1 solar collector typical parameters [9]

Figure II-11 example of compound parabolic collectors in Portugal [13]

Figure II-12 example of evacuated tube collectors in Tibet [14]

II.5 backup

The solar resource is not always available and all plants need to be backed up with and auxiliary system.

Either it is possible to backup a solar air-conditioning plant with chilled water generated by a compression chiller, or with a boiler heating up the hot water for the thermal process.

Concerning greenhouse gas emissions, the ratio of the carbon factors for electricity and natural gas is around 2. The ratio of the COP of compression chillers and heat driven chillers is largely superior, therefore backing up with a compression chiller is better in terms of CO_2 emissions.

On the other hand, a boiler can provide a important addition to a solar plant since it may be used to boost the hot water to a temperature than is usable by the thermal chiller.

As an example, considering a chiller requires water at 90°C, and the return to the collectors is at 85°C. If the temperature at the collectors is 88°C, then the chiller won't work unless the boiler is used to boost the temperature by two degrees. The return will then be 85°C, and the collectors will have a slightly better efficiency.

III CASE STUDIES

This chapter finishes the background research with two case studies on solar assisted air conditioning successful implementation, in Germany and in Portugal.

III.1 Malteser's hospital – Kamenz, Germany

This 210-bed hospital in Germany was taken as a demonstrational project under the EU's 5th framework program for RTD – part ENERGY (Thermie-Program). Hence there was a broad dissemination of results until the plant started to be run by the hospital's technical services. It is a €3M project, which was 60% funded. The estimated payback with the funding is 10 years [10].

Figure III-1 – Malteser's Hospital St. Johannes in Kamenz [10]

Due to the innovativeness of some applied technologies, the commissioning was held for over a year, facing several technical problems. These were monitored and are well documented in [10].

The scheme consists in a trigeneration plant. The main driver is a phosphor acid fuel cell, sized to meet the electrical base load of the hospital (200kWe – 220kWth). The heat is recovered for ambient heating, backed up with natural gas boilers (total heating of 1.8MWth) or for adsorption cooling (cooling capacity of 105kWth). Compression chiller and ice storage backs up the adsorption chiller during cooling demand peaks (total cooling of 220kWth). The heat storage that feeds the adsorption chiller and the heating plant is also supplied by a small solar hot water installation. Figure III-2 represents schematically the main components of the plant.

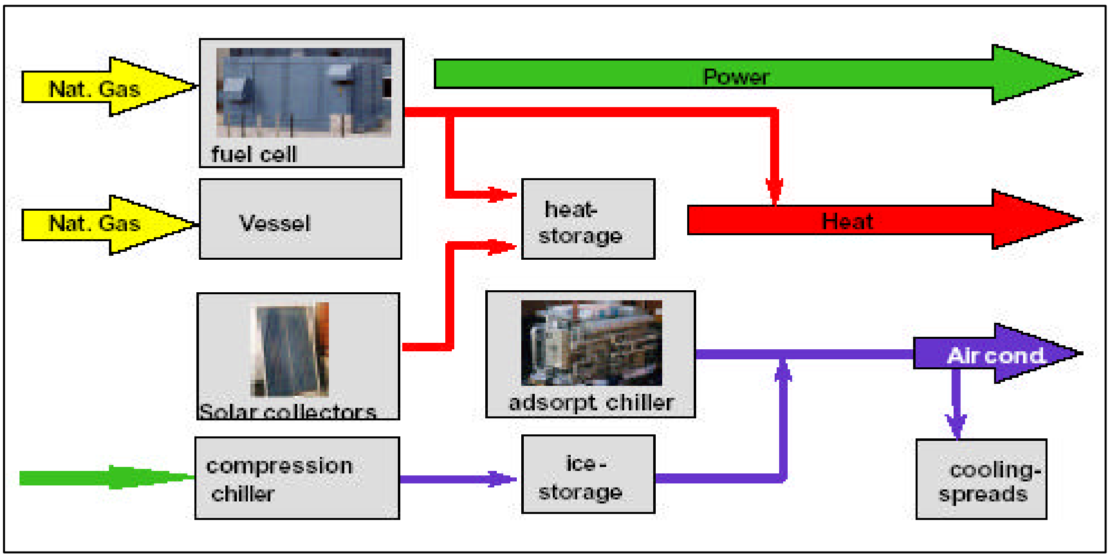


Figure III-2 schematic of Malteser's trigeneration scheme [10]

The main driver of the thermal chiller is the waste heat from the fuel cell. The solar collectors only complement the heating (for cooling) demand. This case study was included in this report since this is one of the few commercial adsorption chillers running in Europe.

The adsorption chiller is manufactured in Japan by Mayekawa and consists in the model Mycom ADR30. It operates with silica gel as sorbent and water as refrigerant. The adsorber and desorbers are operating as reversible chambers, since the sorbent is solid. The cycle period is 10 minutes. The large heat exchange surfaces allow a more "thermodinamically ideal" mode of operation, and since there is no crystallisation danger, the machine can operate at lower temperatures. At nominal conditions, Mycom ADR30 requires 175kWth of hot water at 80ºC to produce 105kWth of chilled water (COP of 0.60).

In Malteser's plant, the chiller operated sometimes at heating temperatures around 65°C, i.e. in a temperature range that is not usable for absorption chillers. It was observed that the performance was about 85% of the nominal performance indicated in the catalogue (COP 0.6). It was however verified at different conditions fluctuations between 0.51[1] and 0.65 were obtained. This is due to the fact that the hot water was sometimes delivered at a temperature higher than the nominal. It was also observed that the performance rose continuously since start-up and during the monitoring probably because inert gases were slowly being eliminated from the silica gel.

---

[1] It is indicated 0.56 in the report, however it seems to be an error since 85% of 0,6 is 0.51

The adsorption chiller was completed at 24/05/00, at its first run at 28/06/00 but it only became fully functional on the 14/07/00. The commissioning period of nearly 50 days reflect the problems of new technologies when opposed against the one day commissioning for the compression chiller.

III.2 ATECNIC factory's office – Sintra, Portugal

The company ATECNIC, a Portuguese manufacturer of air-conditioning equipment, installed a solar assisted air-conditioning plant in their Sintra's factory offices [3].

The plant was commissioned in December 1999, and was funded by the EU's Thermie-Program.

Figure III-3 shows the façade of the factory, where the air-handling unit can be seen on the right and the array of solar collectors (total of $72m^2$) to the left.

Figure III-3 façade with solar collectors and ahu [3]

The plant consists in a desiccant system. The double deck air-handling unit is equipped with a Robatherm rotating desiccant wheel (silica gel), which reduces the humidity ratio of the income air. The air is then pre-cooled with the action of a rotating heat recovery unit (exchanging heat to the return air, at a lesser temperature), and then humidified (adiabatically cooled) by the action of nozzle type air humidifiers in the supply. ATECNIC's AHU is pictured in Figure III-4.

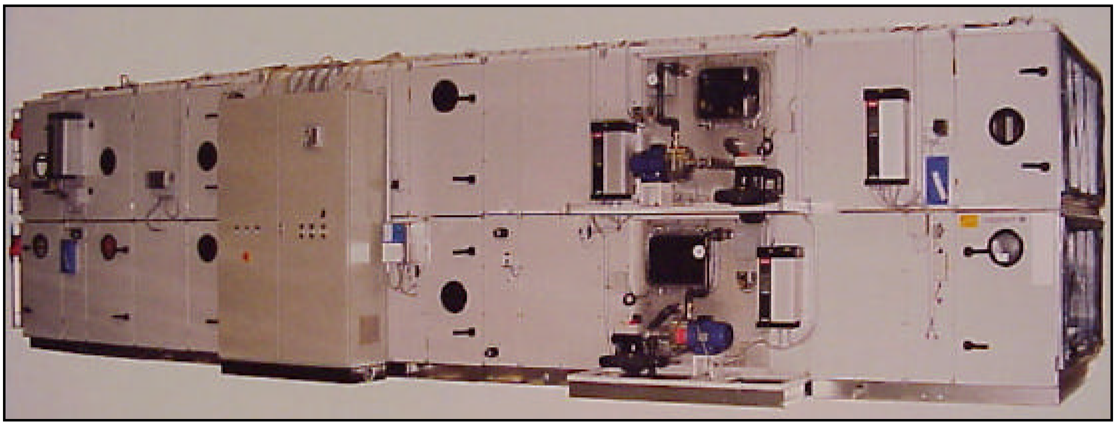


Figure III-4 ATECNIC's ahu [3]

The regeneration of the desiccant wheel is done via a heat exchanger coil in the return air. Two bypasses are included, one in the supply air stream along the desiccant wheel (to allow free cooling) and one along the regeneration air heat exchanger and desiccant wheel, to avoid pressure drops when the wheel is not working.

The air-handling unit works with variable airflow, up to a maximum of 9600m^3/h. There is an increased consumption in the AHU's fans, caused by the augmented pressure loss of all the new components. The maximum electric load is 15kWe, and the maximum cooling power is 75kWth. At these conditions, the COP, as defined by the ratio between the cooling capacity and the regeneration heat, is 0.78.

The heat production sub-system is based on 72m^2 of Compound Parabolic Collectors (CPC) with low optical concentration ratio.

CPC's are a family of collectors, where the incident radiation in concentrated in a small collection area running along the focus of a parabolic mirrored surface (Figure III-5). The ratio between the collector aperture area, $A_{aperture}$, and the collector absorber area, $A_{absorber}$, is called the (optical) concentration ratio:

$$C = \frac{A_{aperture}}{A_{absorber}}$$

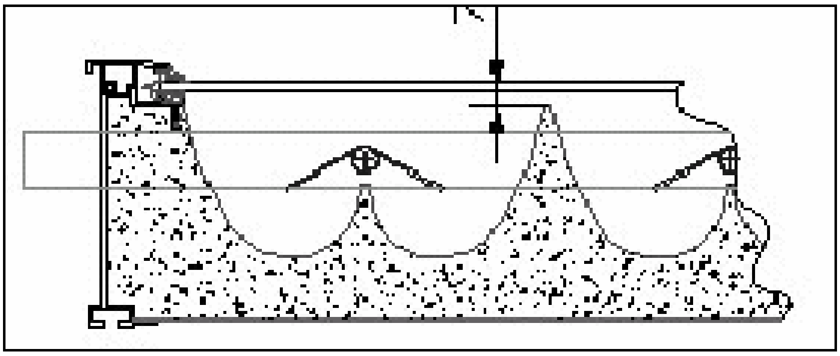


Figure III-5 schematic of CPC collector [11]

The collectors of the installation were manufactured by SETSOL, now extinct. A similar technology is now found in AOSOL collectors. The circulating media on the collectors is anti-freezing fluid, and is connected to a $3m^3$ buffer tank with a plate heat exchanger.

IV THE DEMAND SIDE

On this chapter, it is initiated the section dedicated to study the potential for solar cooling in Lisbon. This chapter deals with the building side, i.e. the demand side of the solar cooling plant. This is particularly relevant since the distribution of loads must follow the solar availability in order to minimise storage. This match between supply and demand is one of the major advantages of solar cooling technologies.

IV.1 background

An office building was selected for this study, since they represent a growing trend in the Portuguese construction industry along with housing. Office buildings in Lisbon are expected to be air-conditioned due to seasonal high temperatures. This is normally done with electric chillers.

For new build offices, the increase of power requirements must be met by the electricity utilities. It is being verified in Portugal that this increase in consumption is being met by building new natural gas power stations. This is either done with public funds through a City Council's investment, or under PFI contracts. In the end, the company that runs the office will always be paying for either the maintenance of the PFI contract (through local taxes) or extended energy bills, thus incurring in a drop of competitiveness.

The Great London Authority's decision on granting planning permission for new build developments, under the condition of having 10% of the energy consumed generated on-site from renewable sources, is not a "eco-façade/green marketing" decision, but a very clever way of converging the construction industry to Kyoto. It reduces the need of creating new power stations to meet new demand, thus safeguarding emission rights. This economical competitiveness is then passed to utility companies and (hopefully) to customers. Utility companies are not prejudiced in the long term, due to new opportunities in Carbon trading, and only the construction industry may suffer on the immediate since they must support the investment cost of the renewables. But on the other hand they have the opportunity to renew the image of an industry that is generally viewed with suspicion.

Solar air conditioning will not solve any of these problems, but it shows an alternative to conventional solutions.

IV.2 building description

The selected building consists in a 6-storey, L-shaped block used for offices. Its architectural layout and internal loads represent a good practice design, with insulation and solar protection slightly above building regulations.

Ideally, a solar air-conditioning plant should be designed for a "best practice" office building, since the plant would be designed to lower loads and would therefore require a smaller investment.

However, and since there is also an interest in studying this type of technologies in retrofitting existing offices, the energy efficiency measures were kept to a realistic minimum.

Each of the 6 floors is divided into three zones, with different ventilation and air conditioning controls:

- Zone East – air-conditioned and ventilated, gross floor area of 264m^2
- Zone Central – free running, gross floor area of 121m^2
- Zone South – air-conditioned and ventilated, gross floor area of 220m^2

Heating is considered negligible due to the high internal gains found in office buildings. Zone East and Zone South are considered office spaces, and Zone Central, service area with stairs and toilettes (without air-conditioning).

The following Figure IV-1 represents the simplified schematic for one floor:

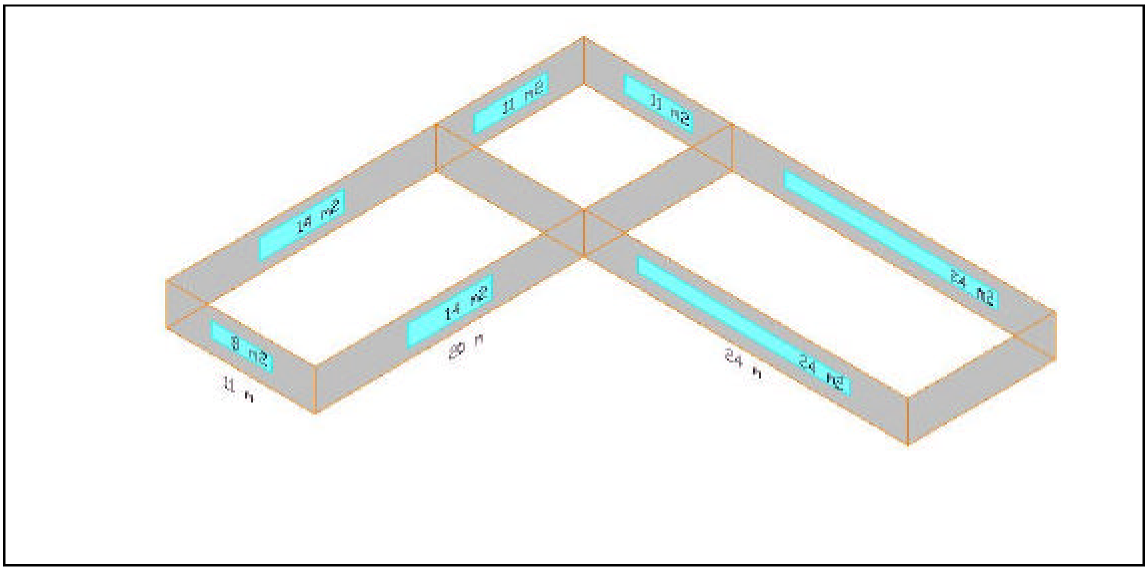


Figure IV-1 floor schematic

IV.3 building data

The office building is sited in Lisbon. As an office, it is open from 8h to 20h, during weekdays. All zones have 0.6 air changes per hour on external infiltrations and 1 air change per hour on ventilation.

The weather was simulated using an *energy plus* weather file for Lisbon, consisting in computer –generated data based on weather statistics measured at the Portela Airport.

For zones East and South, the occupation density is 1person/10m^2, seated and doing light work (150W/person). Each person has a computer (140W/person). The Central Zone, used only as a passage, has 5 persons walking (1.3 m/s - 305W/person).

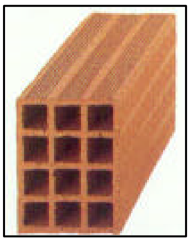
The lighting load is 17W/m^2 (30% convective) and it is assigned to an on/off control depending on the external horizontal radiation (setpoint on 1600kJ/h.m^2). The same control system controls also the internal shades, existent in all windows.

The walls consist in:

external walls:

Consists in ceramic brick with 15cm thickness (tijolo de 15), with external insulation (polystyrene – "wallmate" of 5cm). The internal and external layers are 2cm of mortar (reboco), with a light colour (solar absorptance internal and external set as 50%).

The convection coefficient adopted are the TRNSYS default coefficients [15] in TRNSYS units, 11kJ/h.m^2 internally and 64kJ/h.m^{2}K exteriorly (respectively 3.05 and 17.78 W/m^{2}K).



brick/

“tijolo de 15”

External wall	e	?	K
	[mm]	*[W/m.K]*	*[W/m2.K]*
Mortar	20	1.15	57.50
“Wallmate”	50	0.03	0.60
Brick	150	0.54	3.60
Mortar	20	1.15	57.50
U - value[2]			0.46

Table IV-1 external wall components

interior walls:

The internal walls consist in a single 11cm ceramic brick’s layer (tijolo de 11), with 2cm mortar on the “interior” and “exterior” layers painted with a light colour (solar absorptance internal and external set as 50%).

The convection coefficient used is 11kJ/h.m^{2}K for both sides (3.05 W/m^{2}K).

brick/

“tijolo de 11”

Internal wall	e	?	K
	[mm]	*[W/m.K]*	*[W/m2.K]*
Mortar	20	1.15	57.50
Brick	110	0.54	4.91
Mortar	20	1.15	57.50
U - value[3]			2.45

Table IV-2 internal wall components

[2] Including convection coefficients; TRNSYS values

[3] Including convection coefficients; TRNSYS values

windows

The windows have double glazing (U-value = 2.7 $W/m^2.K$, and G-value = 77.7%) and an aluminium frame (15% total area). All windows have internal shading, which is assigned to an on/off control depending on the external horizontal radiation (setpoint on 1600kJ/$h.m^2$ – 450W/m^2).

IV.4 the TRNSYS model

This sub-chapter intends to explain the calculations and assumptions that are behind the loads' calculation. It is impossible to give a full overview on TRNSYS and the models used.

For estimating the demand profile for the case study, it was decided to use thermal modelling software. The first simulation tool adopted was ESP-r, and some early models were designed. However, it was decided to change for TRNSYS version 15 since it was being used (and still is) in several research projects regarding solar air-conditioning for its flexibility in modelling energy plants/ systems.

A model of the building was created, using a single central floor as reference. The floor itself is a middle floor, being considered adiabatic to the ceiling and ground.

TRNSYS (TRaNsient SYstem Simulation program) is more of an equation solver than a specific building modeller. It allows simulating complex systems, selecting default models representative of the system's elements and establishing relations between these default models.

There is a large variety of these default models (in TRNSYS called TYPES) representing external data (such has weather climate file-readers), calculation methods (e.g. solar radiation processors, control systems, etc.) or sub-systems (multi-zone building data, overhang shading, absorption chillers, etc.).

It is an extremely powerful and flexible tool, since it allows creating any complex system, by connecting the relevant variables from one model as inputs to other model.

On the other hand its interface is not very friendly, requiring a deeper knowledge of the building's physics or system behaviour while creating the model.

The building model, as any TRNSYS project, is a group of TYPES whose relations define the complex model. Figure IV-2, describes the relations between all the TYPES that form the model, and it is followed by an explanation on the models used:

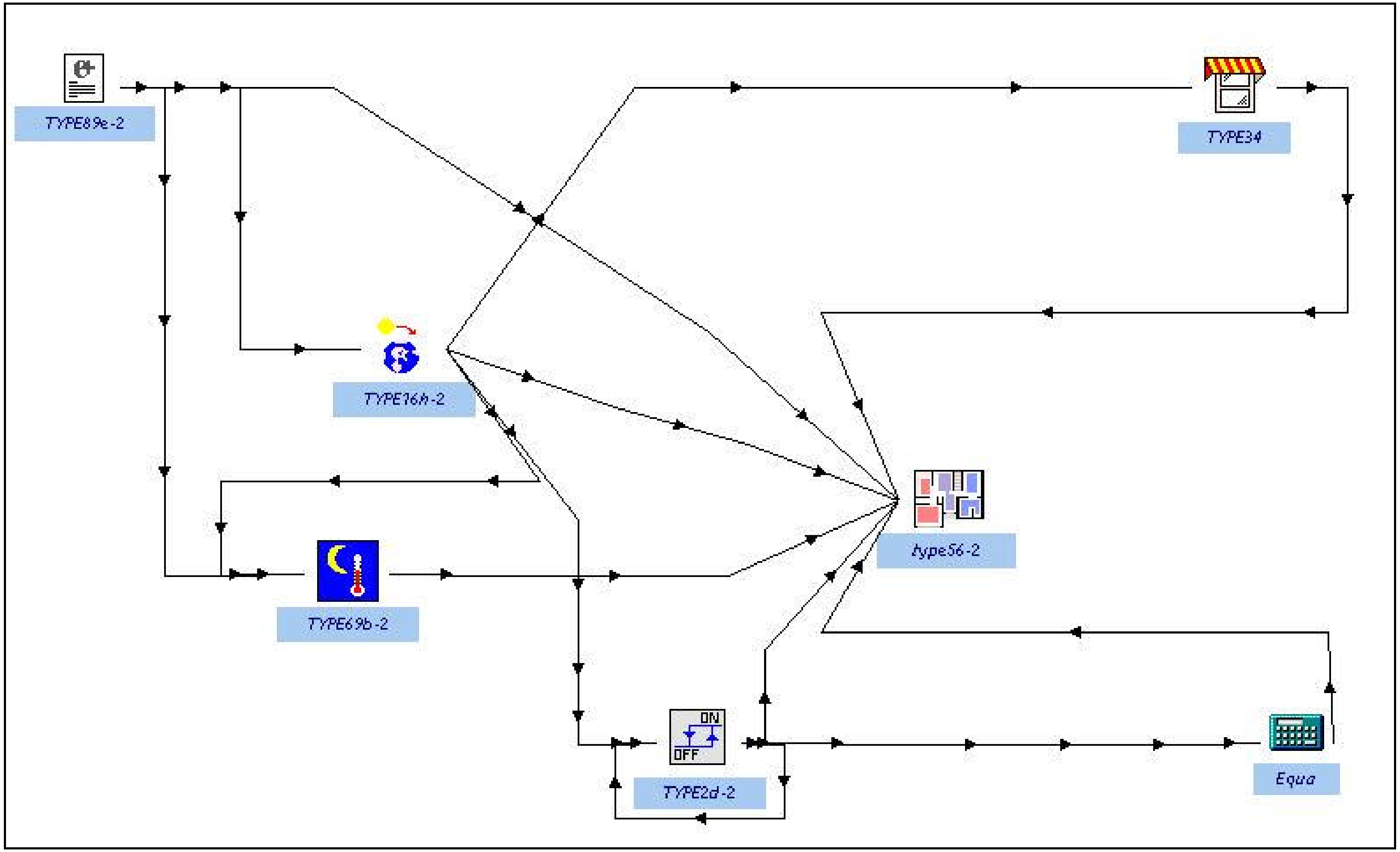


Figure IV-2 TRNSYS model

TYPE 89$_{e\text{-}2}$

The Type 89 is the generic name for a *Formatted Weather File Data Reader*. Particularly, this Type 89$_{e\text{-}2}$ reads the data from an external file with the *Energy Plus* format (.epw).

TYPE 16$_{h\text{-}2}$

The TYPE 16$_{h\text{-}2}$ is a *Solar Radiation Processor*. The one used in this project has the *Global Horizontal Radiation* and *Direct Horizontal Radiation* variables as inputs (from Type 89) and calculates the *Total Radiation* and *Beam Radiation* (normal radiation) on each of the five surfaces used on this project (Horizontal, Vertical facing North, Vertical facing East, Vertical facing West, Vertical facing South).

The relevant parameters are the latitude (around 38ºN for Lisbon) and a correction on the shift in solar time angle since the time (hours) of Energy Plus format are in local standard time (-7.97º).

TYPE 69$_{b\text{-}2}$

The Type 69 is a simple model that determines a *fictive sky temperature* (output) for long-wave radiation exchange models that will be required further on.

The required inputs are the *Diffuse Radiation on the Horizontal* (as calculated by the radiation processor, TYPE 16) and TYPE 89´s *Direct Normal Radiation* (horizontal), *Dry Bulb Temperature* and *Dew Point Temperature*, as extracted directly form the Energy Plus Weather File.

TYPE 56$_{\text{-}2}$

The TYPE 56 (version 2) is the *Multi-Zone Building Model* and can be considered the “heart” of this overall model. It describes a multi-zone building in terms of fabric, internal gains, schedules, ventilation, and infiltration, as common in other thermal simulation packages.

Due to the complexity and quantity of parameters required, a support program, PREBID (Figure IV-3), is used to arrange all the parameters into external files that are read by TYPE56.

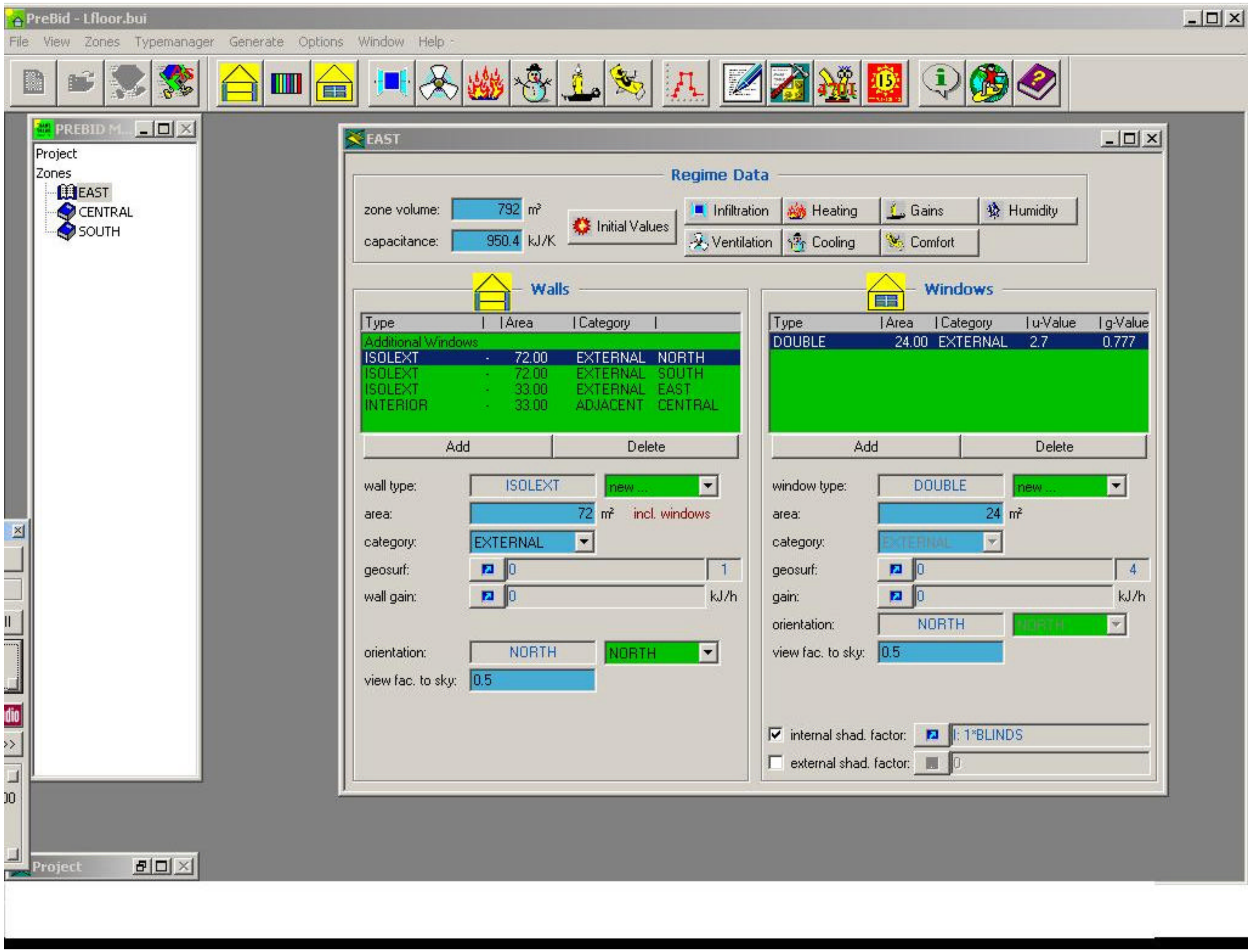


Figure IV-3 snapshot of PREBID interface

On PREBID, all the zones are specified in terms of geometry, fabric materials, internal loads, schedules of operation, etc., but PREBID doesn't interpret the relative position of elements, particularly the shadowing in an L-shaped building.

However, it is possible to assign an external input to any TYPE 56/PREBID parameter. Therefore, three external functions were introduced, EWS, BLINDS and EXTILUM. EWS to calculate the *External Shading Factor* of the South Window on the East Room (which is shadowed by the wall of Zone South), BLINDS, to calculate the *Internal Shading Factor* of all windows, and EXTILUM, to simulate the *Control Strategy* of the lighting system, under the internal gains.

Besides these external functions that will be detailed further on, TYPE 56, has a multitude of inputs from TYPE 89 (*Dry Bulb Temperature*, *Percent Relative Humidity*), from Type 69 (*fictive sky temperature*) and TYPE 16 (*Incident Angle*, *Beam Radiation* and *Total Radiation* for each of the five surfaces used in this project).

TYPE 34

Type 34 simulates *Overhang and Wingwall Shading*. It is used to generate the *External Shading Factor* used as the TYPE 56's external function input EWS (East room, Window South).

The *External Shading Factor* ranges from 0 (no shading) to 1 (total shading).

Type 34's inputs are all from TYPE 16 (Window's *beam radiation, horizontal diffuse radiation, total horizontal radiation, solar azimuth angle* and *solar zenith angle*) as well as its own parameters describing the wingwall's relative positions (see also Figure IV-4 wingwall schematic and Table IV-3 wingwall dimensions):

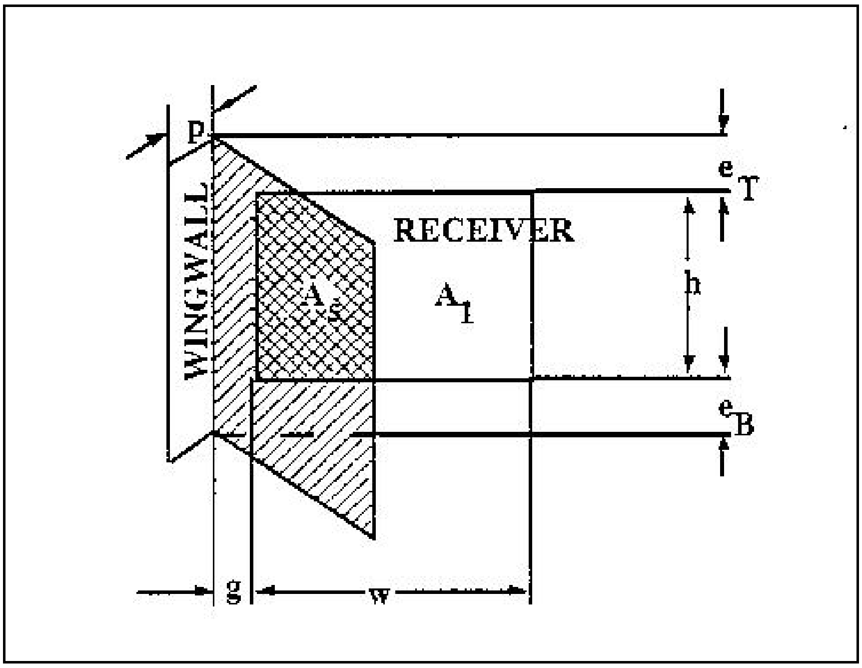


Figure IV-4 wingwall schematic [13]

w (m)	h (m)	g (m)	p (m=	e_T (m)	e_B (m)
24	1	1	1	3	3

Table IV-3 wingwall dimensions

TYPE 2$_{d-2}$

TYPE 2 is an *on/off differential controller* ($_{d-2}$ stands for the version), and is used to generate the TYPE 56's function EXTILUM, that controls the internal lighting on the building.

TYPE 2 has its only input from TYPE 89's *total horizontal radiation*, and its output is the *Output Control Function,* either 0 or 1.

It uses a setpoint of 1600kJ/h.m^2 – above that value, EXTILUM is "0", all lights are off; below that value EXTILUM is "1", all lights are on.

EQUA

Equa/ Equations Editor is the generic TRNSYS name for models made by the user, which use only simple equations.

Equa was used on this project to calculate the function BLINDS, the third of TYPE 56's external function, that controls the internal blinds. It uses the same parameters as EXTILUM, yet it is its complementary (above 1600kJ/h.m^2 BLINDS=1, below, BLINDS=0).

IV.5 results of building model and sensitivity analysis

As it was expected, the heating loads were found negligible.

For Lisbon's base case, the sum of the East zone and South zone cooling loads is given by the following profile (cooling power in kW vs. hour of the year):

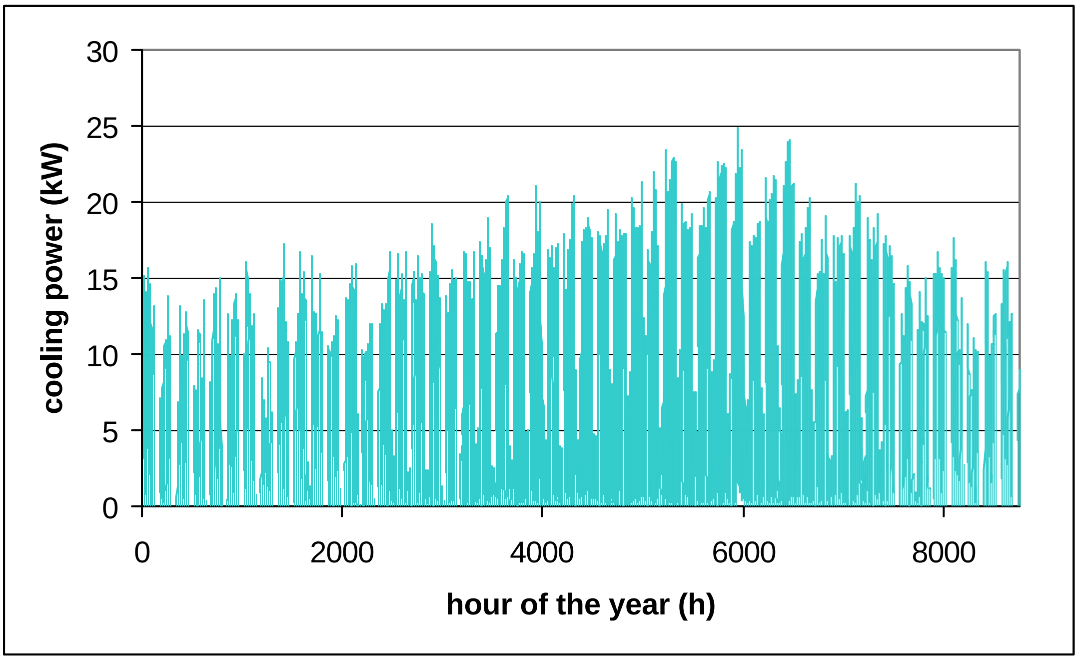


Figure IV-5 central floor annual cooling profile

A sensitivity analysis was performed to evaluate changes in different parameters. Two weather files were tested, Lisbon and Porto Santo in Madeira Islands. Porto Santo is in the middle of the Atlantic and benefits from the Gulf Stream; hence it is a more wet and mild climate.

The sensitive analysis included a variation on the East and West windows of the South zone, with a reduction of 30% and 50% of the area, and also a variation over the base case with a reduction to 2cm of “wallmate” on the walls’ insulation.

The results are summarised in Table IV-4:

		Maximum temperature in Zone Central (ºC)	Maximum cooling load in Zone East (W)	Maximum cooling load in Zone South (W)	Annual energy consumption in Zone East (MJ)	Annual energy consumption in Zone South (MJ)
Lisbon	Base case	35	13047	12058	73327	72618
	EW win50%	34,9	13033	10939	73162	59929
	EW win30%	34,8	13027	10543	73065	54971
	Insulation 2cm	34,8	12877	11940	65484	64659
Porto Santo	Base case	33,3	15375	12076	87387	84010
	EW win50%	33,1	15372	9813	87242	71181
	EW win30%	33,1	15371	9004	87135	66213
	Insulation 2cm	32,9	14780	11413	80499	76777

Table IV-4 sensitivity analysis results

These comprehend only the load to be removed locally with e.g. fancoils (sensible load) for a single floor.

The variation of the cooling load with the reduction of the windows area is consistent with experience. However, the reduction of the cooling load with the insulation thickness can only be explained to the relatively low infiltration of outside air, very high internal gains, and to the fact that there is no night ventilation.

As expected, the heating loads were negligible due to the typically large internal gains found in offices and the moderate external temperatures and the good levels of insulation of the model.

IV.6 considerations on the HVAC installation

The total cooling load that the refrigeration plant will have to meet and especially the shape of the cooling profile is not only a function of the building's cooling requirements, but from the interaction of this demand with the building's HVAC installation and controls.

The real shape of the demand profile is extremely complex to determine, and more important, highly dependant on the specifics of the HVAC system.

For this project, which is trying to find answers to generic offices, the building's HVAC system was simplified to independent fancoils in each of the two AC zones. It was also assumed that the control was made per zone, and was "perfect", i.e. with an immediate response to changing conditions. The cooling setpoint selected for the model was 25ºC.

The ventilation was assumed as one air-change per hour and is included on the cooling profile obtained with TRNSYS. It was assumed it would be provided by a single AHU that would bring the outside air temperature to the setpoint temperature of the zone.

Since the temperature of the chilled water (7ºC) on the fancoils is below dew point, condensation occurs. This means that a room with a 30kW requirement for (sensible) cooling will actually need more capacity since condensation is literally dropping chilled water down the drain (latent). The rate of condensation will depend on the room ambient temperature and moisture content. Constant inlet air conditions were assumed on the fancoil (T=26ºC; RH=50%), to keep calculation simple. For these conditions the Sensible to Latent Ratio (SLR) is about 0.85 [16].

The air-handling unit is the only HVAC component that requires a heating load, since the outside ventilation air shouldn't be supplied at winter at low temperature. Free cooling opportunities were not considered in order to keep the model simple. It was also assumed that the minor heat requirements from the air-handling unit could be met either by re-circulation of the extracted air or even with the chillers rejected heat due to the low temperatures accepted at the AHU's coils.

This concludes that the TRNSYS cooling profile is strictly the sum of the sensible loads for Zones South and East and are not counting with the latent load, and therefore must be multiplied by 1.18 (1/0.85).

IV.7 cooling demand profile

It was assumed in the last sub-chapter that the total cooling load for each floor would consist on the TRNSYS calculated load multiplied by 1.18 to meet the latent losses on the fan-coils. To estimate the total building load, the load obtained for the middle floor should be multiplied by six to take into account all floors. This assumes the roof of the top floor and the floor of the ground floor will be adiabatic. Again, this assumption doesn't influence much on the results, since these are easily super insulated, and add a significant simplicity by keeping the TRNSYS thermal model just to one floor.

The total cooling load from the building model is resumed in the following table:

	Max Cooling Load (kW)	Specific Cooling Load (W/m2)	Monthly Average Maximum (kW)
Zone South	78.0	49	-
Zone East	72.0	55	-
Total	149.4	51	98.4

Table IV-5 Summary of the building's cooling load

As previously mentioned, the first two result columns come directly from the building's thermal simulation, taking into account the required latent cooling/dehumidification ratio of 1.18 and the six floors.

The solar plant performance will be assessed not in an hourly annual simulation but in an hourly monthly average simulation as presented in Figure IV-6, showing and average yearly profile of the cooling load of the selected building. Its 12 peaks represent an average weekday for each of the 12 months of the year.

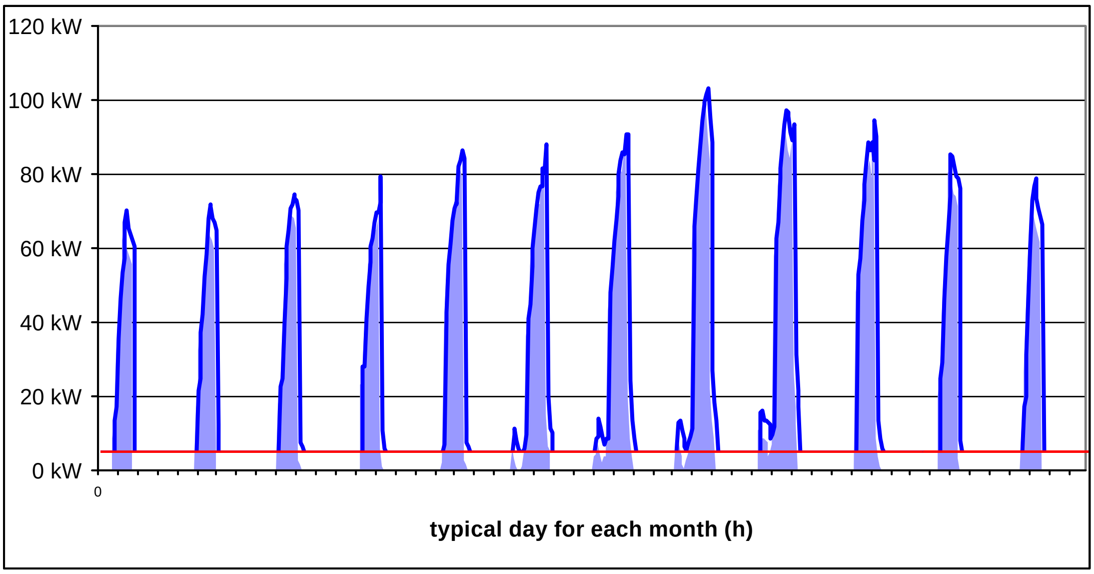


Figure IV-6 building's cooling demand profile

Hence the third column of Table IV-5 indicates the monthly averaged cooling peak.

The decision on working on monthly averages was taken after considering the different approaches when sizing equipment with annual energy and with power. In typical HVAC industry, equipments are sized based on the required power, since the equipment must meet all the cooling peaks and provide thermal comfort during all the times. The “energy” approach is very different since the equipments must be sized for its economics and energy output – if the “power” approach was taken, an expensive solar air conditioning plant would have to be sized for 150kW while in average only 100kW are required and representative of the typical loads the plant will be subjected to. A top-up electrical chiller would meet the remaining load.

V THE PLANT MODELLING TOOL

This chapter describes the tool created for modelling the plant.

Originally, this project was intended to use TRNSYS for the building simulation and for the plant simulation.

A TRNSYS model of the plant was started, but problems with the simulation of storage and control loops lead the development of the model to a stop. As a contingency plan, it was decided to develop a spreadsheet-based tool in line with the "GenCalc", developed at the Strathclyde University for Part B of the Masters in Energy Systems and the Environment by the Energy Efficient Estates Group [17].

The new tool inherits the philosophy behind "GenCalc", as it is designed to match supply and demand at an hourly basis, by establishing energy balance on plant equipments, reflecting changes in the demand profile, available solar hot water, etc.

It tries to meet an annual hourly demand profile[4] with the available solar energy generation resources, and up to the maximum capacity of the equipments. The remaining is matched by conventional backup technologies.

For a better comprehension on the tool and how it was designed, this chapter starts with a small description of a generic AC plant with a heat driven chiller. It then moves for the description of the modelling tool itself.

V.1 heat driven AC plant

The AC plant consists in a heat production subgroup and a chilled water production subgroup. The schematic of the plant is presented in Figure V-1.

[4] Due to limitations in the processing power of the computer running the spreadsheet in Excel with an 8760 hours year, it was decided to use monthly averages. Chapter IV includes a brief description on the practical effects of this decision

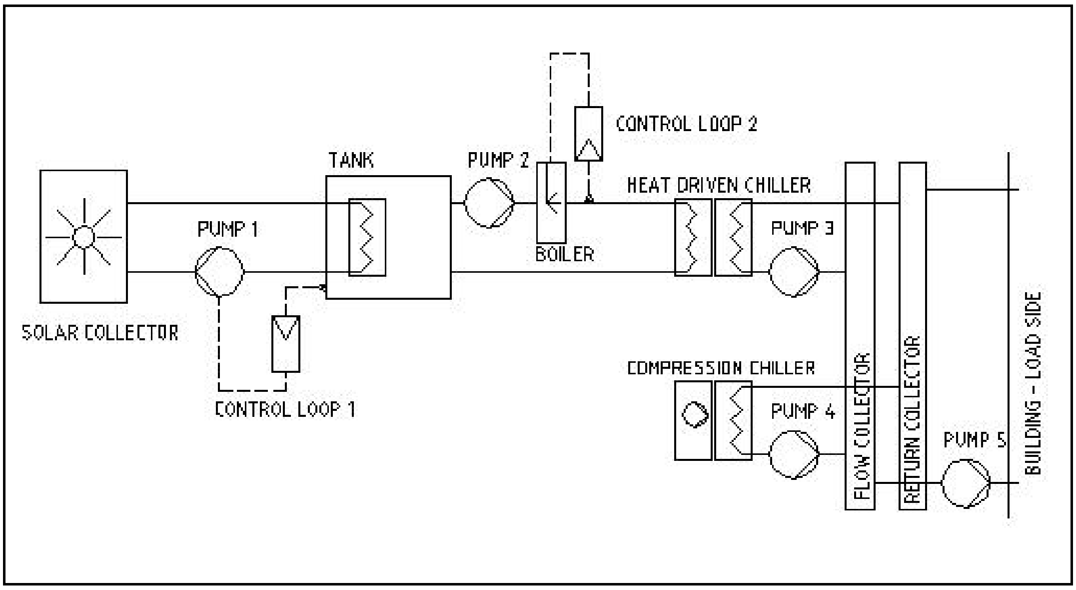


Figure V-1 plant schematic

The chilled water production subgroup consists in a heat driven chiller in parallel with a compression chiller. They are both connected to flow and return collectors that make the boundary between the plant and the building/ load side.

When the building/ load side asks for chilled water, the heat driven chiller will have the priority in producing the cooling. If it is not capable, then the compression chiller will produce the cooling or will top up the heat driven chiller. The reasons why the heat driven chiller may not meet the cooling demand are:

Reason	**Action**
The building load is superior to the heat driven chiller capacity	The heat driven chiller will produce its maximum load and the rest is produced by the compression chiller
There is not enough solar hot water (1)	The boiler is activated and meets the required temperature
There is not enough solar hot water (2)	The compression chiller is activated. The solar hot water is circulated between tank and solar collectors at variable speed until part of it reaches the desired temperature.

Table V-1 chilled water production control matrix

If there is not enough solar hot water, two actions can be taken. Either the boiler is activated and its heat drives the chiller, or the compression chiller produces alone the chilled water instead.

In this project it was decided the last option since it is more rational in terms of CO_2 emissions – burning gas specifically for a heat driven chiller (single stage) releases more CO_2 than using electricity in a compression chiller due to the superior COPs of the latter (see Chapter II).

The heat production subgroup consists in an array with solar collectors backed up with a boiler in series. The solar collectors are connected to a storage tank to store hot water at the best temperature the collectors can supply. The water in the primary circuit is circulated by a pump, which is activated by the differential temperature of a hot water tank (located at the bottom of the tank, near the return water inlet from the secondary circuit – where the water will be colder). The sensor compares this temperature with the outlet of the solar collectors' array. This is a standard procedure for solar installations since it avoids heat losses by transmission and radiation caused by circulating hot water on the primary circuit.

If the boiler was activated, the return of the hot water from the chiller to the tank would come at a fairly high temperature. If it was superior to the tank temperature, it would inhibit the primary pump or low its speed, resulting in a lower flow speeds in the collectors, more time of permanence of the water in the collectors and therefore a higher temperature on the outlet of the collectors.

The hot water tank is sized to store energy but is essential to act as a buffer minimising hourly demand supply mismatches. It is also essential to deliver some thermal inertia to the system, since clouds, or even changes in the AC load could unbalance the system. From a hydraulic perspective it is also required to separate the primary and secondary circuits since they have different flow speeds.

A temperature sensor before the chiller is set to ensure the temperature is high enough to drive the absorption/adsorption process. It controls the boiler, case it is prioritised over the backup chiller.

V.2 description of the plant tool

The tool is organized in different sheets. The main ones are the "SCALC1", "SCALC2" and "PCALC". Figure V-2 gives an overview of "SCALC1".

K20 = =(I20+J20)/SCOOL!C8

	A	B	C	D	E	F	G	H	I	J	K
1	SUPPLY:		Cooling								
2			thermal cooling demand (kWh)	chill generated by heat driven chiller (kWh)	COP	Heat required to drive chiller	Solar Heat available to drive chiller	Backup Heat required to drive chiller	Alternative coolingl backup (1) (kWh thermal)	Peak cooling top up (2)(kWh thermal)	supplementary electrical cooling (1+2)
3											
4	January		2.811	2.811	402	5.111	1.658	3.454	1.900	0	760
5	February		3.026	3.026	402	5.501	2.686	2.815	1.548	0	619
6	March		3.273	3.273	402	5.951	4.443	1.508	829	0	332
7	April		3.213	3.213	402	5.842	5.641	201	111	0	44
8	May		3.922	3.922	402	7.132	7.132	0	0	0	0
9	June		4.089	4.089	402	7.435	7.435	0	0	0	0
10	July		4.695	4.695	402	8.537	8.517	19	11	0	4
11	August		5.460	5.460	402	9.927	9.168	759	418	0	167
12	September		5.595	5.595	402	10.172	8.431	1.741	957	0	383
13	October		4.554	4.554	402	8.280	6.997	1.283	706	0	282
14	November		3.700	3.700	402	6.727	5.800	926	510	0	204
15	December		3.172	3.172	402	5.767	4.435	1.332	733	0	293
16											
17	ANNUAL TOTALS		47.510	47.510	4.818	86.382	72.343	14.038	7.721	0	3.088
18											
19	January wkday	0	0	0	0,55	0	0	0	0	0	0
20	January wkday	1	0	0	0,55	0	0	0	0	0	0
21	January wkday	2	0	0	0,55	0	0	0	0	0	0
22	January wkday	3	0	0	0,55	0	0	0	0	0	0
23	January wkday	4	0	0	0,55	0	0	0	0	0	0
24	January wkday	5	0	0	0,55	0	0	0	0	0	0
25	January wkday	6	0	0	0,55	0	0	0	0	0	0
26	January wkday	7	0	0	0,55	0	0	0	0	0	0
27	January wkday	8	0	0	0,55	0	0	0	0	0	0
28	January wkday	9	1	1	0,55	3	2	0	0	0	0
29	January wkday	10	2	2	0,55	4	4	0	0	0	0
30	January wkday	11	5	5	0,55	9	9	0	0	0	0
31	January wkday	12	7	7	0,55	13	13	0	0	0	0
32	January wkday	13	8	8	0,55	15	15	0	0	0	0
33	January wkday	14	9	9	0,55	16	9	7	4	0	2
34	January wkday	15	10	10	0,55	19	1	18	10	0	4
35	January wkday	16	11	11	0,55	20	0	20	11	0	4
36	January wkday	17	10	10	0,55	18	0	18	10	0	4
37	January wkday	18	10	10	0,55	18	0	18	10	0	4
38	January wkday	19	10	10	0,55	17	0	17	10	0	4
39	January wkday	20	9	9	0,55	17	0	17	9	0	4
40	January wkday	21	0	0	0,55	0	0	0	0	0	0
41	January wkday	22	0	0	0,55	0	0	0	0	0	0
42	January wkday	23	0	0	0,55	0	0	0	0	0	0
43											
44	January wkend	0	0	0	0,55	0	0	0	0	0	0
45	January wkend	1	0	0	0,55	0	0	0	0	0	0
46	January wkend	2	0	0	0,55	0	0	0	0	0	0

Figure V-2 snapshot of generic spreadsheet

The structure of all of the main sheets is always the same, and it is divided in 3 areas.

In the bottom and forming the core of the calculations are 24 sets of 24 hours representing typical weekdays and typical weekend days for all the 12 months. On Figure V-2, this starts in January weekdays, 0 hours (row 19) and it goes down to December, weekend, 23h, although on the snapshot is only visible down to January, Weekend, 2h (row 46). This is the area where the calculations are made. Its cells generally establish horizontal relations with other cells, since they occur at the same timestep.

The other two areas work as a summary of the calculations. The top area shows the monthly figures for the quantity that is being calculated in that column. It sums the typical days' relevant quantities from the columns in the third area and multiplies it by 261/12 for weekdays and 104/12 for weekends (261+104=365). This is found in Figure V-2 between rows 4 and 15.

The middle area (in Figure V-2 row 17) gives the annual totals, simply the sum of all the months.

The same sheet SCALC1 is represented again in Figure V-3 with figures from Option 2 (only Area 3, January weekday figures presented). SCALC1 is more related with the refrigeration group of the plant while SCALC2 relates with the Solar Water Heating group.

A	B	C	D	E	F	G	H	I	J	K
SUPPLY:		Cooling								
	hour	thermal cooling demand (kWh)	chill generated by heat driven chiller (kWh)	COP heat driven chiller	Heat required to drive chiller	Solar Heat available to drive chiller	Backup Heat required to drive chiller	Alternative coolingl backup (1) (kWh thermal)	Peak cooling top up (2)(kWh thermal)	Electricity consumption (kWh) (1+2)
January wkday	0	0	0	0,46	0	0	0	0	0	0
January wkday	1	0	0	0,46	0	0	0	0	0	0
January wkday	2	0	0	0,46	0	0	0	0	0	0
January wkday	3	0	0	0,46	0	0	0	0	0	0
January wkday	4	0	0	0,46	0	0	0	0	0	0
January wkday	5	0	0	0,46	0	0	0	0	0	0
January wkday	6	0	0	0,46	0	0	0	0	0	0
January wkday	7	0	0	0,46	0	0	0	0	0	0
January wkday	8	0	0	0,46	0	0	0	0	0	0
January wkday	9	9	9	0,46	19	14	5	2	0	1
January wkday	10	12	12	0,46	27	27	0	0	0	0
January wkday	11	31	31	0,46	66	52	15	7	0	2
January wkday	12	42	42	0,46	90	60	30	14	0	3
January wkday	13	48	48	0,46	105	51	54	25	0	6
January wkday	14	53	53	0,46	114	35	79	36	0	9
January wkday	15	62	62	0,46	135	8	128	59	0	14
January wkday	16	65	65	0,46	142	0	142	65	0	16
January wkday	17	60	60	0,46	131	0	131	60	0	14
January wkday	18	59	59	0,46	128	0	128	59	0	14
January wkday	19	58	58	0,46	125	0	125	58	0	14
January wkday	20	56	56	0,46	122	0	122	56	0	13
January wkday	21	0	0	0,46	0	0	0	0	0	0
January wkday	22	0	0	0,46	0	0	0	0	0	0
January wkday	23	0	0	0,46	0	0	0	0	0	0

Figure V-3 "SCALC1" snapshot

- Column A identifies the month and type of day

- Column B identifies the hour, from 0 to 23h

- Column C imports the final cooling demand profile as described in Chapter IV. No cooling demand is available on the weekend.

- Column D dictates the capacity of cooling that can potentially be provided by the heat driven chiller – it is a conditional statement that compares with the defined installed capacity of the heat driven chiller with the demand, e.g. if the cooling demand is 120kWth (for one hour time steps, it means 120kW of power), then only 98kWth would be generated by the chiller (assuming a chiller capacity of 98kW).

- Column E defines the COP of the chiller. It was assumed constant for a conservative approach, since it is dependant on the rejected heat temperature (it can be assumed constant because it is done by the heat dissipators, external to our model – see chapter IV and VI) and on process hot water temperature. This temperature is always above a minimum, since is controlled by the boiler backup (see again chapter IV and VI), when the temperature is higher than the nominal, this causes a better performance on the chiller.

- Column F calculates the heat required to drive the chiller. It is calculated by the ratio between Column D and Column E:

$$Heat\ \mathrm{Re}\,quired = \frac{CoolingDem\,and}{COPheatdrivenChiller}$$

- Columns G and H breakdown the origin of the heat into solar heat and backup heat. These are dependent on the availability of the solar resource and are calculated in sheet "SCALC2", detailed ahead.

- Column I is optional but it was adopted on this project. It allows modelling a more environmental and rational solution, since it quantifies the thermal cooling energy that was provided by the backup boiler (natural gas) to drive the chiller. This is obtained by multiplying Column E by Column H:

$$BackupCoolingbyCompressionChiller = \frac{HeaGeneratedbyBackupBoiler}{COPheatdrivenChiller}$$

- Column J calculates the thermal cooling energy met by the compression chiller for top ups when the cooling demand is superior to the thermal driven chiller's capacity. This is obtained by subtracting Column D minus Column C, as can be seen in the following equation:

$$TopUpbyCompressionChiller = CoolingDemand - CoolingbheatDrivenChiller$$

- Column K sums the thermal cooling energy met by the compression chiller (Columns I and J) and divides by the average COP of the compression chiller. A discussion on the assumption of using a constant COP can again be seen in Chapter IV and VI. The expression is:

$$ElectricalConsumption = \frac{BackupbyCompressionChiller + TopUpbyCompressionChiller}{COPCompressionChiller}$$

The sheet "SCALC2" is more focused on modelling the heat production subgroup. Has a reminder, this sheet originates the breakdown on solar and "boiler" heat used on "SCALC1". A snapshot is presented in Figure V-4, for the same conditions as before.

- Column C imports the heat demand from the "SCALC1" sheet, Column F. It is assumed as a Solar Hot Water demand.

- Columns D and E contain the solar radiation in kWh/m2 and the external temperature respectively. These figures were obtained directly from TRNSYS, using the module "radiation processor" to calculate the hourly radiation with an optimum panel orientation for Summer applications in Portugal (South facing, azimuth 180º (TRNSYS assumes an azimuth South at 0º), inclination approximately as the latitude 38º, albedo 0.4 just as a default value) [18]

- Column F and G calculate the collectors' efficiency. The efficiency is again defined by

$$h = k(\Theta) \cdot c_0 - \frac{(T_{av} - T_{amb})}{G_p} \cdot c_1 - \frac{(T_{av} - T_{amb})^2}{G_p} \cdot c_2$$

- Column H calculates the actual SHW produced by multiplying the radiation (Column D) with the efficiency (Column G)
- Columns I, J and K are iterative: Column I defines surplus heat as the stored solar heat (at previous hour) plus the SHW produced during the actual hour minus the actual hour's heat demand

- Column J defines the backup heat required as the difference between demand and the SHW produced and the anterior (previous hour) stored solar heat

- Column K defines the stored heat by summing to the previous hour stored heat, the difference between heat demand and SHW produced.

A	B	C	D	E	F	G	H	I	J	K
SUPPLY:										
	hour	SHW demand (kWh)	Radiation (kWh/m2)	External dry temperature	x	collector efficiency	SHW produced (kWh)	surplus heat (kWh)	backup heat required (kWh)	stored solar heat (kWh)
January wkday	0	0	0,00	9	0,00	0,63	0,00	0	0	0
January wkday	1	0	0,00	9	0,00	0,63	0,00	0	0	0
January wkday	2	0	0,00	9	0,00	0,63	0,00	0	0	0
January wkday	3	0	0,00	9	0,00	0,63	0,00	0	0	0
January wkday	4	0	0,00	9	0,00	0,63	0,00	0	0	0
January wkday	5	0	0,00	9	0,00	0,63	0,00	0	0	0
January wkday	6	0	0,00	8	0,00	0,63	0,00	0	0	0
January wkday	7	0	0,00	8	0,00	0,63	0,00	0	0	0
January wkday	8	0	0,00	8	0,00	0,63	0,00	0	0	0
January wkday	9	19	0,30	8	0,18	0,15	13,64	0	5	0
January wkday	10	27	0,39	8	0,14	0,26	30,37	0	0	3
January wkday	11	66	0,48	9	0,11	0,33	48,46	0	15	0
January wkday	12	90	0,54	10	0,10	0,37	60,46	0	30	0
January wkday	13	105	0,49	10	0,11	0,35	51,39	0	54	0
January wkday	14	114	0,40	11	0,13	0,29	34,89	0	79	0
January wkday	15	135	0,25	12	0,20	0,10	7,64	0	128	0
January wkday	16	142	0,12	12	0,43	-0,48	0,00	0	142	0
January wkday	17	131	0,02	12	2,47	-5,75	0,00	0	131	0
January wkday	18	128	0,00	12	0,00	0,63	0,00	0	128	0
January wkday	19	125	0,00	11	0,00	0,63	0,00	0	125	0
January wkday	20	122	0,00	11	0,00	0,63	0,00	0	122	0
January wkday	21	0	0,00	10	0,00	0,63	0,00	0	0	0
January wkday	22	0	0,00	10	0,00	0,63	0,00	0	0	0
January wkday	23	0	0,00	10	0,00	0,63	0,00	0	0	0

Figure V-4 snapshot of "SCALC2"

The “PENERGY” sheet balances the used electricity, breaking it up into day and night time tariffs. Figure V-5 is a snapshot of this sheet.

SUPPLY:		PRIMARY ENERGY				
		Total Imported Electricity	Imported other uses Gas (kWh)	Imported GRID electricity ON peak (kWh)	Imported GRID electricity OFF peak (kWh)	CO2 produced (kg CO2)
January		2.284	0	2.284	0	891
February		2.203	0	2.203	0	859
March		2.061	0	2.061	0	804
April		1.739	0	1.739	0	678
May		1.883	0	1.874	9	734
June		1.895	0	1.785	110	739
July		2.096	0	1.917	179	817
August		2.693	0	2.488	205	1.050
September		3.084	0	2.715	369	1.203
October		2.443	0	2.438	4	953
November		2.245	0	2.245	0	876
December		2.147	0	2.147	0	837
ANNUAL TOTALS		26.771	0	25.895	876	10.441
January wkday	0	0		0	0	0
January wkday	1	0		0	0	0
January wkday	2	0		0	0	0
January wkday	3	0		0	0	0
January wkday	4	0		0	0	0
January wkday	5	0		0	0	0
January wkday	6	0		0	0	0
January wkday	7	0		0	0	0
January wkday	8	0		0	0	0
January wkday	9	1		1	0	0
January wkday	10	0		0	0	0
January wkday	11	2		2	0	1
January wkday	12	3		3	0	1
January wkday	13	6		6	0	2
January wkday	14	9		9	0	3
January wkday	15	14		14	0	5

January wkday	16	16		16	0	6
January wkday	17	14		14	0	6
January wkday	18	14		14	0	5
January wkday	19	14		14	0	5
January wkday	20	13		13	0	5
January wkday	21	0		0	0	0
January wkday	22	0		0	0	0
January wkday	23	0		0	0	0
January wkend	0	0		0	0	0
January wkend	1	0		0	0	0

Figure V-5 snapshot of "PENERGY"

- Column C imports the electricity used from Column K, "SCALC1" (total electricity used by compression chiller), and Column E does the same to the Gas imported (from "backup heat needed to drive chiller". The heat to gas conversion is done by dividing by a seasonal average efficiency of the boiler – typically 85%.

Please note that in this project it was assumed the compression chiller would provide the backup cooling, therefore on the actual simulations on the next chapter this value is zero

- Columns F and G breakdown the electricity use into the day and night time tariffs (on this project, day time tariff assumed between 07:00 and 23:00).

- Column L calculates the CO_2 emissions by multiplying the electricity and gas energy consumption by its CO_2 factors (for electricity, the Portuguese figure of 0.39 $kgCO_2/kWh$ [19] and for natural gas, 0.19 $kgCO_2/kWh$, the UK figure)

VI THE SUPPLY SIDE

For determining the applicability of solar air conditioning for the selected office, it was decided to model the dynamic behaviour of two heat driven chiller plants: the first plant with an absorption chiller and the second plant with an adsorption chiller. A third option with conventional chiller was selected to reference the other two. Desiccant cooling was excluded since the cooling agent is air, and is not used in offices with heavy cooling loads.

VI.1 the solar plant

The calculations were made for the 6 floors, with a 98 kW cooling peak (monthly average). It was assumed that the comfort cooling requirements (the extra 51kW difference from the hourly peak and the hourly average peak) was met in all three options with conventional chillers, and therefore is not included in these calculations.

Because different compression, absorption and especially adsorption chillers are found in the market with different capacities, required figures such as capital costs were estimated not from real market products but by parameters such as €/kW. This was required to establish a fair comparison.

In all the three options, and also for the sake of simplification, the rejected heat dissipation was not modelled. However it is safe to assume that the electrical load consumed in the cooling towers would be the same since it is only a function of the cooling demand and the ambient temperature, which is constant to all options.

The three options selected are:

- Option 0 (base case with conventional compression chillers)
- Option 1 (chilled water produced by an adsorption chiller)
- Option 2 (chilled water produced by an absorption chiller)

The model performed an annual simulation with the tool described in Chapter V. Each option was run with hourly time steps to reflect changes in demand profiles, available solar power, external temperature etc.

VI.2 sizing of equipments

The sizing of the plant's main equipments started by selecting fixed working conditions for the chillers, i.e. defining minimum hot water temperatures and the respective COP.

Compound Parabolic Collectors were also selected for the heating production sub-group for their low cost in Portugal and their competitive performance at high temperatures [9]. Tube vacuum collectors would increase the performance of the system, but their cost is more than thrice.

The total cooling load for the building is around 150kW. However the monthly hourly average is around 100kW. The heat driven chillers were sized to meet the monthly annual average, and therefore avoid unwanted capacity only used for top-up peaks.

For costing, it was verified that the designed equipment couldn't be found in the market with the exact capacities. Therefore, and to keep the comparison neutral, all the costs were estimated as a pro-rata (€/kW) of equipment with a capacity of around 100kW.

The sizing procedures for all the main equipments were done in the following sequence:

1. Define heat driven chiller capacity – (100kW for Options 1 and 2)
2. Define chillers hot water temperature and COP (from technical catalogues and other sources of information)
3. Assume CPC collectors working at the chillers required temperature
4. Determine area of collectors based on cost effectiveness - assume no heat storage. This is an iterative process with the plant-modelling tool. A selling price for chilled water should be defined, usually the ratio between the electricity price (assumed 12p/kWh) and the conventional chiller seasonal COP (assumed 4.2 [16]).
5. Fix the area of collectors
6. Determine the "real" heat storage[5], again and like in the solar collectors, based on its cost effectiveness.

[5] The heat storage model carries a small inaccuracy – since it runs monthly averages, there is a discontinuity at the end of the months. In a normal 8760h simulation, the stored heat at the end of the month would pass for the first day of the next month. In this simulation, the loop is closed and passes for the 1st day of the same month. This means that it is not possible on the model to simulate seasonal storage. This inaccuracy does not reflect a large overall error since in practical terms seasonal storage would be unrealistic due to costs, space occupied and thermal losses of the tank. Chapter V describes the model, and its reading is suggested for a better understanding of the limitations of the model.

The following sub-chapters describe each option and demonstrate how the sizing procedure was done.

VI.3 option 0 – compression chiller

Option 0 is the base case. It consists in a conventional compression chiller with an annual Coefficient of Performance (COP) of 4.2 [16]. This COP is for the chiller only and is relatively high since the electrical load concerning the heat dissipation (cooling tower) is not being taken into account. It was obtained from the catalogue of a chiller manufacturer [16] for 100kW Water – Water Chiller.

Figure VI-1 represents a schematic of the plant. Flow and Return collectors were included to accommodate the constant flow in the primary circuit (chiller) and the variable flow in the secondary circuit (building – load side).

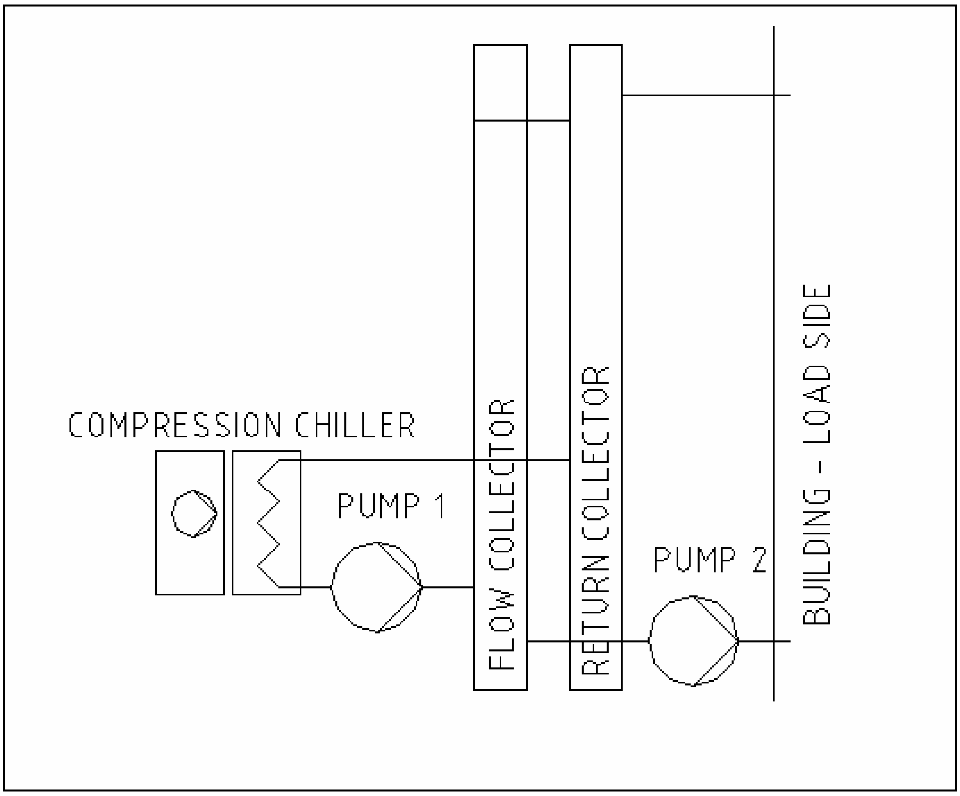


Figure VI-1 option 1 schematic

The sizing of the equipments is straightforward, since a single chiller is required. The comparative capital costs [20] are available in Table VI-1.

Financial Summary					
Capital Costs					
	Unit	**Quantity**	**Unit Cost (Euros)**	**Amount (Euros)**	**Reference**
Compression Chiller (Water Cooled)	kW	98	152,00	14.956,95	*Spons M&E price book 2003*
Heat disspiation	kW	98	45,00	4.428,04	*Spons M&E price book 2004*
Other	u	0	0,00	0,00	
Total:				**19.384,99**	

Table VI-1 option 0 comparative capital costs

VI.4 option 1 – absorption chiller

Option 1 consists in a solar assisted air conditioning plant using an absorption chiller.

For the heat production an array of CPCs with 200m^2 was sized according with the method described in Chapter V. It was verified (Figure VI-2) that the model is not very sensitive to change with an area between 200 and 300m^2. This value was selected to lower the overall capital cost of the plant.

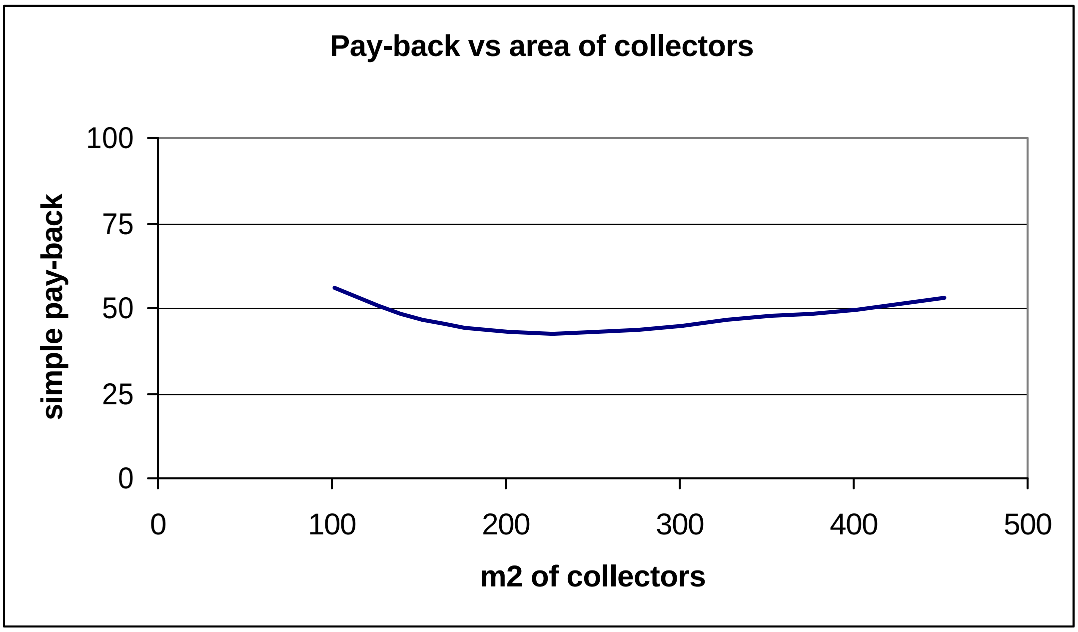


Figure VI-2 evolution of pay-back with number of collectors

The selection of the type of solar collectors was not performed, since CPC's have a good quality/price ratio for high temperature applications [9]. This is reinforced by the fact that CPC's are being produced nationally, being realistic that a project of this nature would be assigned to this technology. The formal procedures for selection of solar collectors are well detailed in Hans-Martin Henning's book, "Solar-Assisted Air-Conditioning in Buildings" [9].

The nearly 0.86m^3 hot water tank is sized to store around 5kWh. This will act as a buffer minimising hourly demand/ supply mismatches. It is also essential to deliver some thermal inertia to the system, since clouds, or even changes in the AC load could unbalance the system. From a hydraulic perspective it is also required to separate the primary and secondary circuits since they have different flows. Figure VI-3 shows the evolution of payback with the storage. It can be verified that it basically doesn't change in the range of values selected (the values on the graphic are rounded without decimal cases). Although there is an energetic benefit, the large requirements of storage are very expensive, and thus it was sized to a minimum only to act as buffer tank.

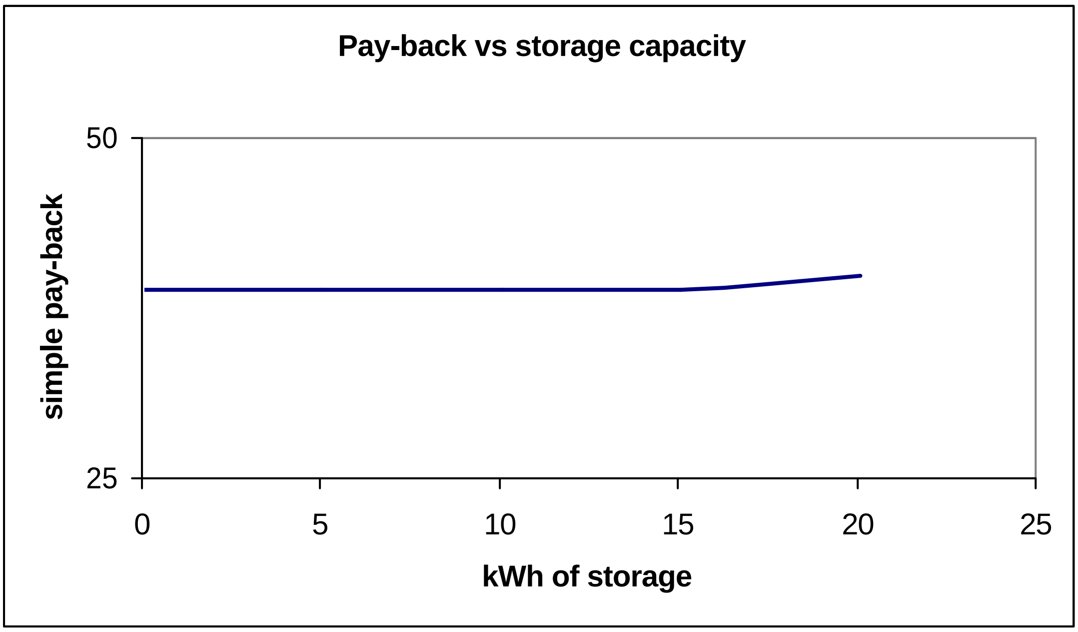


Figure VI-3 evolution of pay-back with size of storage

The 98kW absorption chiller has a seasonal COP 0f 0.68, for cooling water temperature of 31ºC, chilled water temperature of 7ºC, and process hot water temperature of 80ºC (inlet) and 75ºC (outlet) [9].

The backup chiller has a COP of 4.2, like in the base case [16].

It was decided to add the boiler to the diagram and to the costs' estimation because in a real plant, it would be better to have the flexibility to use it for backup instead of the backup chiller, in situations where the boiler tops up the temperature from the hot water leaving the collectors, and the return from the chiller is at a temperature low enough to allow some solar contribution.

Hence, a 137kW boiler was considered since it would be likely used in a real plant.

It was assumed that both chillers would be connected to rejected heat dissipators (e.g. cooling towers). The nature of these heat dissipators varies a lot and since they would dissipate the same amount of heat on the three options (the rejected heat is a function of the building's cooling demand and outside temperature), it was decided not to consider its electricity consumption. This translates is a slightly higher compression chiller Coefficient of Performance (COP). Figure VI-4 represents the schematic of this installation

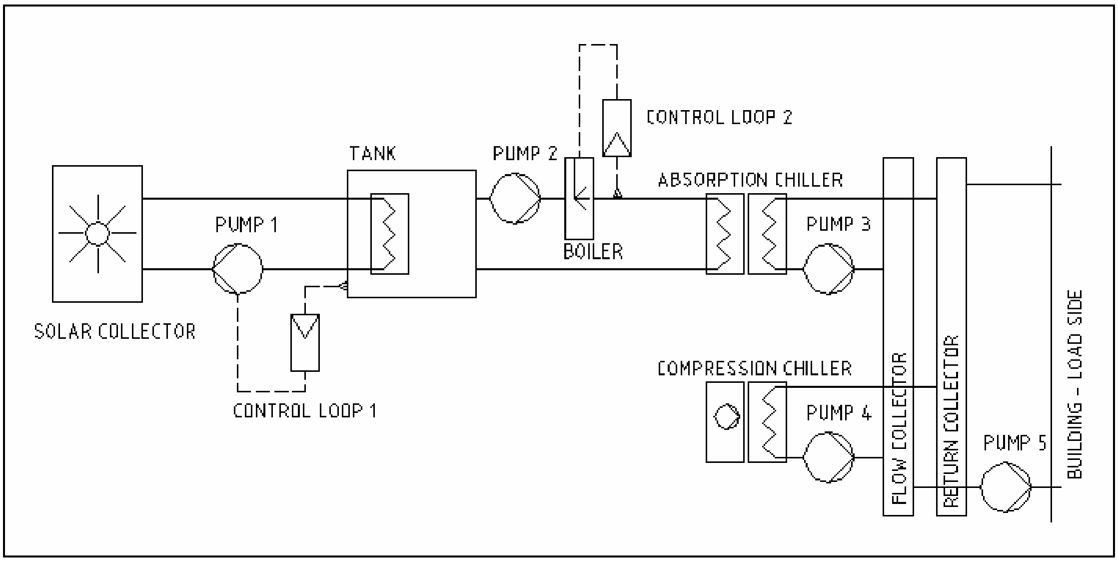


Figure VI-4 schematic of absorption plant

For summary, the main quantities and references are displayed in Table VI-2, and the estimated capital costs in Table VI-3 [9], [16], [20].

Name	Parameter	Unit	Notes
SCOOL1			
Heat Driven Chiller Capacity	98	kW	
Tinput Chiller	80	C	
Toutput Chiller	75	C	
Heat Driven Chiller seasonal COP	0,68		absorption chiller, cooling water inlet 31ºC, chilled water outlet 7ºC
backup/top up chiller COP	4,2		compression chiller, cooling water inlet 30ºC, chilled water outlet 7ºC
SCOOL2			
Area of solar colectors	200	m2	
Heat storage	5	kWh	
Heat storage	0,86	m3	
k(tau)	1		incident angle modifier
Tav	**77,5**	**C**	**average temperature of fluid**
c0	0,628		optical efficiency
c1	1,47		linear loss coefficient
c2	0,022		quadratic loss coefficient

Table VI-2 absorption plant main parameters

Financial Summary					
Capital Costs					
	Unit	**Quantity**	**Unit Cost (Euro)**	**Amount (Euro)**	**References**
Solar Collectors	m2	200,00	252,00	50.400,00	*Hennings, 2004*
Solar Hot Water storage	m3	0,86	1.200,00	1.032,00	*Spons M&E price book 2003*
Backup Boiler	kW	137	20,00	2.733,91	*Spons M&E price book 2003*
Heat Driven Chiller	kW	98	190,00	18.620,00	*Spons M&E price book 2002*
Compression Chiller	kW	93	152,00	14.189,82	*Spons M&E price book 2003*
Heat disspiation	kW	98	45,00	4.410,00	*Spons M&E price book 2004*
Other	u	0%		0,00	*other equipments, installation, risk*
Total:				**91.385,73**	

Table VI-3 absorption plant capital cost estimation

VI.5 option 2 – adsorption chiller

The installation in Option 2 is very to Option 1, but it is using an adsorption chiller instead. This translates in a lower temperature driving the chiller, and therefore an increased performance of the collectors. The drawback is that the cost of the adsorption chiller is higher. Figure VI-5 represents Option 2's simplified schematic.

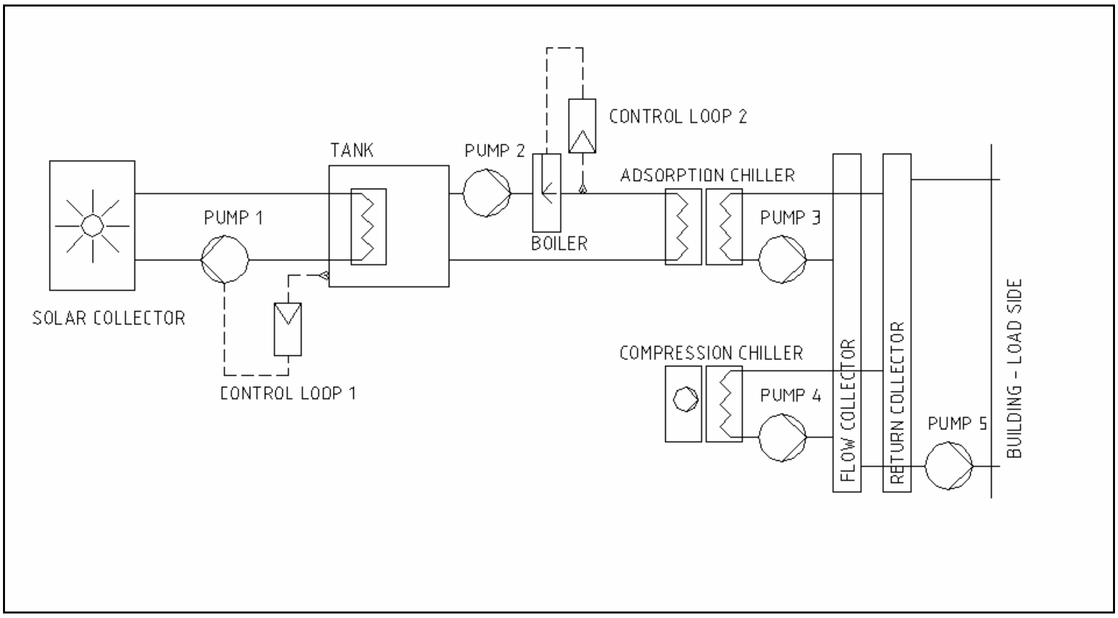


Figure VI-5 adsorption plant schematic

For the heat production, an array of CPCs with $300m^2$ was sized according with the method described in Chapter V. It was verified (Figure VI-6) that this parameter was more sensitive in adsorption than with absorption. However, it still permits a range between 300 and $350m^2$. Again $300m^2$ were selected to keep the overall installation capital cost low.

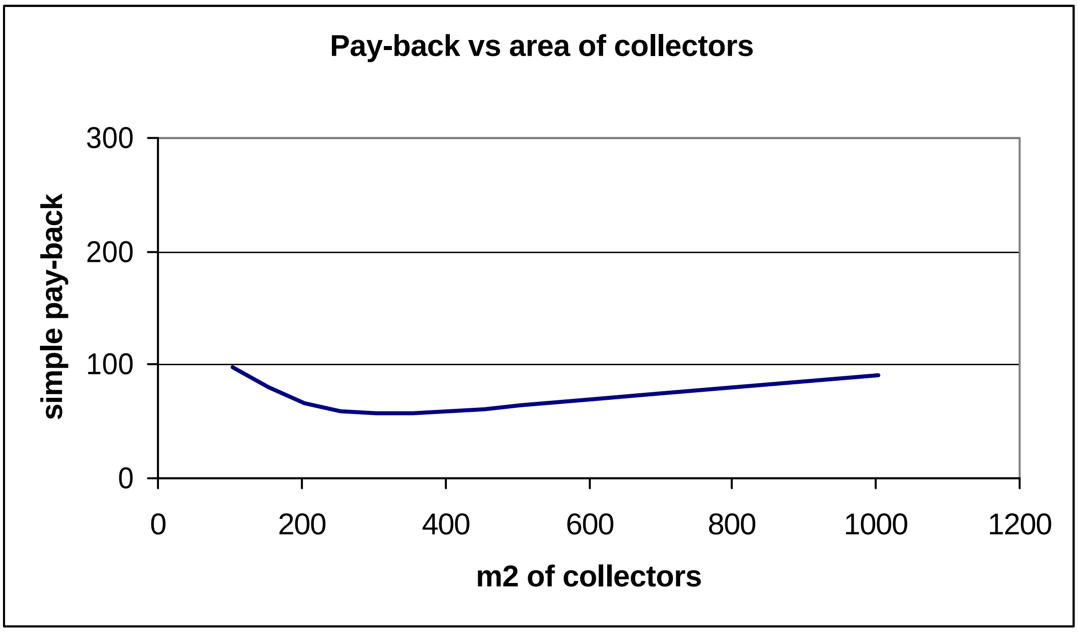


Figure VI-6 evolution of pay-back with number of collectors

Figure VI-7 shows the evolution of payback with the storage. Again it can be verified that it doesn't change much in the range of values selected (the values on the graphic are rounded without decimal cases). Although there is a benefit in energy, the large requirements of storage are very expensive, and thus it was again sized to a minimum hot water tank of nearly 0.86m^3, storing around 5kWh, only to act as buffer tank.

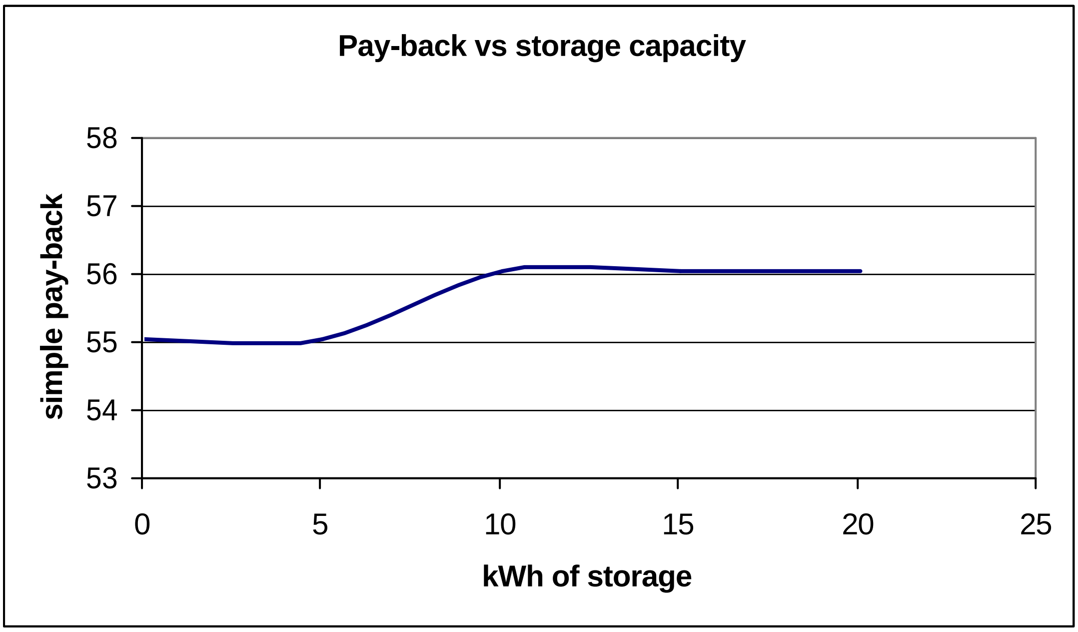


Figure VI-7 evolution of pay-back with size of storage

The 98kW adsorption chiller has a seasonal COP 0f 0.46, for cooling water temperature of 31ºC, chilled water temperature of 9ºC, and process hot water temperature of 65ºC (inlet) and 60ºC (outlet) [9].

The backup chiller has a COP of 4.2, like in the base case [4.2].

A 197kW boiler was also considered in the schematic and capital cost estimation since it would be used in a real plant.

For reference, the main quantities and references are displayed in Table VI-2, and the estimated capital costs in Table VI-3.

Solar Cooling			
Name	**Parameter**	**Unit**	**Notes**
SCOOL1			
Heat Driven Chiller Capacity	98	kWth	
Tinput Chiller	65	C	
Toutput Chiller	60	C	
Heat Driven Chiller seasonal COP	0,46		adsorption chiller, cooling water inlet 31ºC, chilled water outlet 9ºC
backup/top up chiller COP	4,2		compression chiller, cooling water inlet 30ºC, chilled water outlet 7ºC
SCOOL2			
Area of solar colectors	300	m2	
Heat storage	5	kWh	
Heat storage	0,86	m3	
k(tau)	1		incident angle modifier
Tav	**62,5**	**C**	**average temperature of fluid**
c0	0,628		optical efficiency
c1	1,47		linear loss coefficient
c2	0,022		quadratic loss coefficient

Table VI-4 adsorption plant main parameters

Financial Summary					
Capital Costs					
	Unit	**Quantity**	**Unit Cost (Euro)**	**Amount (Euro)**	**Reference**
Solar Collectors	m2	300,00	252,00	75.600,00	*Hennings, 2004*
Solar Hot Water storage	m3	0,86	1.200,00	1.032,00	*Spons M&E price book 2003*
Backup Boiler	u	197	20,00	3.932,99	*Spons M&E price book 2003*
Heat Driven Chiller	kW	98	500,00	49.000,00	*Hennings, 2004 (includes heat dissipator)*
Compression Chiller	kW	90	152,00	13.749,73	*Spons M&E price book 2003*
Heat disspiation	kW	98	45,00	4.410,00	*Spons M&E price book 2004*
Other	u	0	0,00	0,00	
Total:				**147.724,72**	

Table VI-5 adsorption plant capital cost estimation

VI.6 summary of plant parameters

The following table summarises the input parameters for each option and references them.

parameter	**Option 0**	**Option 1**	**Option 2**	**Reference**
Heat driven chiller capacity	0 kW	98 kW	98 kW	Monthly average "peak"
Heat driven seasonal COP	n.a.	0.68	0.46	[9]
Process hot water temperature	n.a.	80-75 °C	65-60 °C	[9]
Chilled water temperature	7 - 12 °C	7 - 12 °C	9 - 14 °C	[16], [9], [9]
Backup chiller capacity	98 kW	93 kW	90 kW	Maximum backup cooling load
Backup Chiller seasonal COP	4.2	4.2	4.2	[16]
Heat rejection water temperature	30 °C	31 °C	31 °C	[16], [9], [9]
Type of collectors	n.a.	CPC	CPC	Collector's parameters from Table II-1[9]
Area of collectors	0	200 m^2	300 m^2	Sized on best pay-back
Hot water storage volume	0	0.86 m^3	0.86 m^3	Sized on best pay-back
Backup boiler	0	137 kW	197 kW	Sized on maximum backup heat required[6]
Heat dissipation	98 kW	98 kW	98 kW	Electrical consumption equal for all options.

Table VI-6

[6] Electrical cooling was selected as backup. If boiler backup was selected, this would be the maximum required.

VI.7 results and validation

The main results obtained are summarised in table Table VI-7.

Columns 3, 4, 5 and 6 show the absolute values obtained, while columns 7, 8 and 9 show the relative values against Option 0, the base case.

		capital cost (k£)	Electricity consumption (kWh/year)	Running cost (€/year)	CO2 emissions (kg/year)	? electric. consmpt. (kWh/yr)	? running cost (€/year)	? CO2 emissions (kg/year)
0	Compression chiller	19385	48533	5770	18928	-	-	-
1	Absorption chiller	91385	29058	3434	11333	-19475	-2336	-7595
2	Adsorption chiller	147725	26771	3160	10441	-21762	-2610	-8487

Table VI-7 summary of plant simulation results

The solar fraction, as defined between the ratio of solar cooling and total cooling is 0.40 and 0.45 respectively for options 1 and 2.

Taking into account the displaced costs and energy savings (electricity), it is possible to calculate the payback, as shown in Figure VI-8.

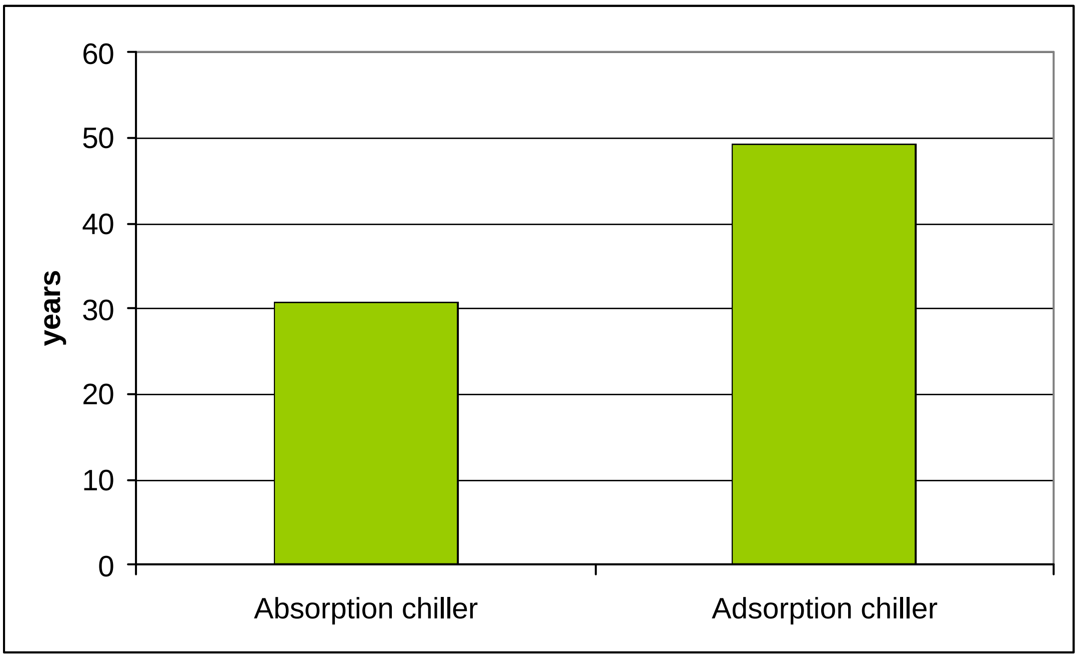


Figure VI-8 estimated pay-back

It is very difficult to validate the plant model tool presented in this project, since it involves so many parameters.

Henning [3] [9] uses frequently a parameter called *specific collector area* (m^2/kWchiller) to characterise an installation. The specific collector area for option 1 is 2.0 and for option 2 is 3.1. This is very similar to the example he provides for an office building in Madrid ([9] pp 114), where a CPC driving an absorption chiller with electrical backup (similar to Option 1) has a specific collector area of 1.49 m^2/kWchiller, and a CPC driving an adsorption chiller with electrical backup (similar to Option 2) has a specific collector area of 2.77m^2/kWchiller. In Henning's example it is not defined the criteria used for selection, and it is not presented the solar fraction obtained.

In a documented case study [9], an office building at Basse Terre, in Guadeloupe, uses a 30kW absorption chiller and a 55kW compression chiller for backup. The cooling load (68kWh/m2) is similar to the one calculated from TRNSYS (70kWh/m2– not counting with Zone Central which is not air-conditioned). The solar fraction obtained is nearly 0.48 against the solar fractions of 0.40 for Option 1 and 0.45 for Option 2. The specific collector area for kW of absorption chiller is 2.04.

Teocharis [21] estimates the pay-back for solar absorption and adsorption air conditioning in Athens in respectively 60 years and 90 years. The figures calculated in this report are quite below the ones in the reference, but it must be taken into account that different assumptions were made concerning energy and capital costs.

It however important to note that the ratio between the paybacks of adsorption and absorption is very similar in both studies (1.5 for Teocharis and 1.7 in present calculations.

VI.8 future work

One of the main contributions of this thesis is the plant tool. The plant tool presents logical results and within what is reasonably expectable. However, a more consistent approach must be taken to validate the tool.

The validation could be made against the simulation of a real existing plant, and checked against actually measured data.

Another area with potential for development is more accurate information from suppliers regarding their equipment. The capital cost of the chillers and their performance under different conditions is still surrounded by uncertainty, and different publications come with slightly different values.

Finally, a method for a better selection of the most cost effective solar collectors and optimum size for the heat driven chiller could be developed to enrich the selection methodology.

VII CONCLUSIONS

Solar air conditioning is technically viable, and despite its relative complexity there is enough expertise and examples of success to make it happen more often. Due to the conventional backup requirements, it brings good opportunities of implementation in hybrid cooling plants (compression + solar).

It is a technology that is not yet economically competitive, and even though some improvements in equipment's technology may be expected, the final barrier to a more mainstream application will be turned down as the capital costs get lower and particularly when energy prices get higher.

The results from the modelling exercise are consistent with other examples and cases studies, yet it is difficult to make a consistent validation of the method due to the quantity of parameters present.

The solar fraction of the adsorption plant (0.45) is superior to the one for the absorption plant (0.40), but the average payback, as can be seen in Figure VII-1, is worse for adsorption due to the high capital cost not only of a more expensive chiller, but also more $100m^2$ of solar collectors, i.e. the lower COP of the adsorption chiller is more determinant than the loss of efficiency of the collectors in absorption.

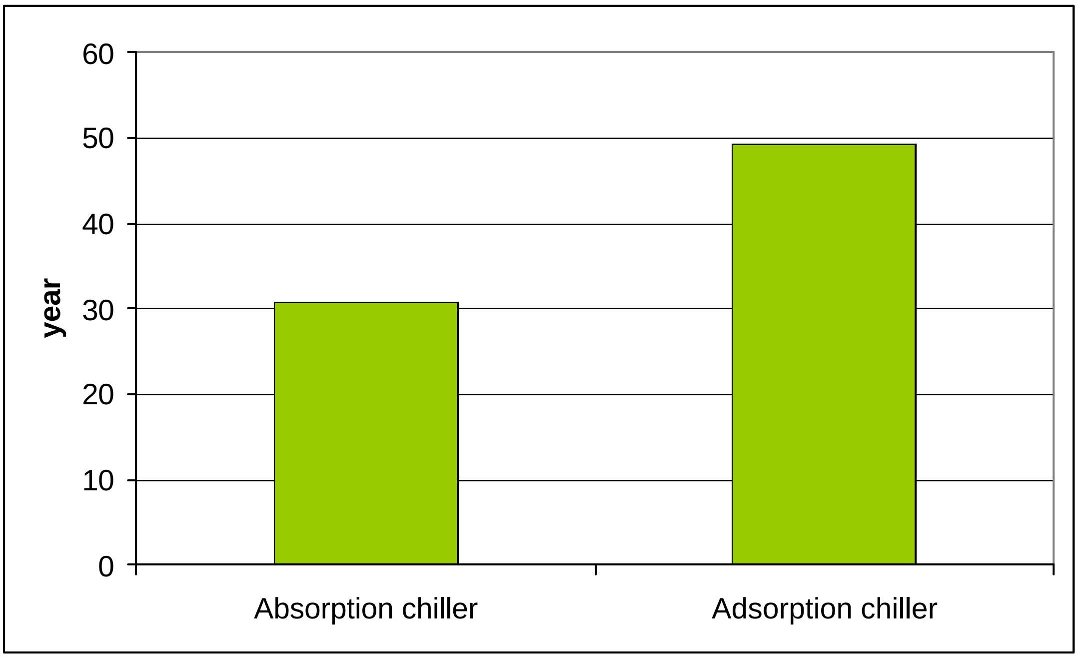


Figure VII-1 estimated pay-back

Presently, absorption seems a more competitive technology for Lisbon because:

- It is economically more attractive
- It is an established technology
- There is an established and competitive market of equipments
- There is local expertise for installation and maintenance
- The difference in the process hot water's temperature is only about 15ºC.
- Adsorption chiller can work at lower temperatures but the COP also drops rapidly

Solar air-conditioning is a renewable energy technology with an enormous marketing potential. It makes the Sun generate chilled water. A demonstrational project in Portugal could fill many of the existent knowledge gaps and perhaps become a precursor on an industry revolution.

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