Air conditioning production by a single effect absorption cooling machine directly coupled to a solar collector field. Application to Spanish climates.

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Abstract

Due to the increasing energy consumption of air conditioning in buildings and the need to decrease the fossil CO2 emissions to the environment, the interest of using renewable energy sources shows up stronger than ever.

The aim of the study presented is to evaluate the upper bound in the potential of solar cooling, using as example the very diverse climates of Spain. In the paper it has been assumed a direct solar coupling between the solar collector field and a single effect absorption cooling machine, without any intermediate solar storage tank. An equation is obtained that shows the dependence of the generator/solar-collectors equilibrium temperature on basic design parameters. The paper analyzes the effect of these on the total amount of cooling produced along the year and the peak cooling power; for instance, the relative size of the solar collectors field with respect to the size of the machine or the evaporation temperature. The paper also includes a discussion on optimal design of endorreversible machines. This study is used to clarify the influence of design parameters of the absorption cycle.

Finally tables are included for the 12 climates of Spain that can be used in sizing at the design step of such direct coupled system.

Keywords: Solar energy, Energy efficiency, Absorption cooling, Low Carbon,

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1. Introduction

The absorption cooling machines are also named heat conversion machines, since they mainly take and release heat at different temperature levels. Therefore they have a great potential for the use of renewable energy sources. However the technology, in our opinion, is not mature at domestic level. Nevertheless this article tries to show the potential of a solar driven single effect absorption machine for cooling purposes.

The solar collectors are supposed to be directly coupled with the machine with very little accumulation in-between. Therefore whenever the heating power and the delivery temperature of the solar field are able to drive the machine, this is turned on and produces a cooling effect.

In the paper the equation that modelizes the coupling is deduced and the total cooling capability during the year, its temporal distribution and the peak power are studied. The delivered cooling effect is analyzed at two temperatures; 5 [ºC] and 14[ºC]. The first corresponds to the use of conventional fan-coil elements and the second to the more recent cooling floor or ceiling as terminal units.

The model presented is a steady state model. No transients are studied. For the solar collectors the very well know algebraic equation for its performance is used and for the absorption cooling machine we have used the algebraic characteristic equation model presented by Ziegler [10]. We have tried to used the simplest models to show the main outcomes from a direct coupling, nevertheless more complex models might be used.

The Ziegler's model [10], is an abridged formulation based on a local energy balance at each element of the machine; evaporator, condenser, absorber and generator and on a linear map approximation of the equilibrium LiBr-H2O. The model must be adjusted with experimental results. The adjustment serves to guess the approximate values of the design parameters like the UA in each element, in a way that resembles a reverse engineering procedure.

It would be interesting to get insight into the meaning of the reasonable range of the values for these design parameters and to find their relationship with the whole absorption/solar collectors system. In order to get this insight, the paper shows the optimal design [6] based on the endorreversible model proposed by Tozer [9]. In this model only the irreversibilities due to the external heat exchanges are taken into account. The model is quite useful since the optimal design equations are solved in close form.

At this point, this background will allow to deduce the coupled behavior of the solar collector field and the absorption machine.

Finally, as an example and with the aim of giving useful values for designing systems like these, a set of tables are included with the total cooling capacity for each zone of Spain as a function of some design parameters of the system. The weather data has been taken from hourly data (8760 values) of a mean typical year for each climatic zone. The weather data are the same than those used by the Spanish building code and the certification software (LIDER and CALENER), [2].

In another paper the application of this model to an energy simulation software will be presented. The software is intended to be used to energy certification of dwellings.

Nomenclatura

α	distribution UA parameter
β	ratio of temperatures
$\Delta\Delta t_{min}$	minimun temperature difference [K]
$\Delta\Delta t$	double difference of temperatures [K]
$\dot{m}$	mass flow rate [kg s^{-1}]
η	efficiency
ψ	ratio of area of solar collectors to machine size [m^2 K kW^{-1}]
A	enthalpy ratio
a	absorber
A_{col}	solar collector area [m^2]
B	isostere slope in Dühring chart
C	enthalpy ratio
c	condensser
COP	Coefficient of perfomance
e	evaporator
G	enthalpy ratio
g	generator
h	specific enthalpy [KJ kg^{-1}]
I	solar global radiation intensity [W m^{-2}]
Q	heat flux [kW]
s	total themal conductance of the installed UA in the absorption cooling machine [kW K^{-1}]
SE	single effect absorption machine
T	mean internal temperature [ºC]
t	mean external temperature [K]
t_g^{eq}	equilibrium temperature of generator [K]
UA	thermal conductivity [kW K^{-1}]

2. Mathematical model

2.1. Pose

The power of a solar collector field, of a certain type (vacuum, plane-selective coating, plane-standard, etc..), depends basically on the total intensity of the solar radiation, the ambient dry temperature (t_{amb}) and of the mean temperature of the produced hot water. This mean value will be equated to the mean temperature of the hot water which drives the absorption cooling machine (t_g). If all the values are constant but t_g, the greater is this value the less is the heating power coming from the solar field at this temperature. However, for an absorption cooling machine, when all the mean temperatures of the external streams (at evaporator, condenser and absorber) are fixed, the greater the mean temperature of the hot water driving the generator the greater the cooling capacity and therefore the power demanded from the solar collector field. Putting both effects together, this means that for a certain area of the solar collector field and for a certain size of the machine, there must be a balance or equilibrium temperature t_g^{eq} that equates the rate at which the energy is collected from the solar field to the heating power needed at the generator of the machine. Besides, it must be pointed out that the machines need a minimum generator temperature t_g^{min} to start cooling. It is then possible, that if the ratio (area solar field/size of machine) were not enough, the equilibrium t_g^{eq} were below this minimum and therefore the direct coupling were not possible.

In what follows the equations of the solar collectors and the absorption machine used to deduce t_g^{eq} are shown.

2.2. Model of the solar collector field

The efficiency of the solar collector [η] is defined by the well known equation:

$$\eta(t_g, t_{amb}, I) = FR_{\tau\alpha} - FRU_L \cdot \left(\frac{t_g - t_{amb}}{I}\right) \tag{1}$$

The terms of (1), represent: on one hand the thermal losses FRU_L as a function of the ambient temperature t_{amb} and the mean hot water temperature t_g and on the other the optical factor $FR_{\tau\alpha}$. In this paper these parameters are kept fixed to: $FR_{\tau\alpha} = 0.825$, $FRU_L = 1.1\,WK^{-1}m^{-2}$, and correspond to a vacuum solar collector.

Therefore for a certain area of collectors $A_{col}(m^2)$ the total power of the collected energy is given by:

$$\dot{Q}_{col} = A_{Col} \cdot I \cdot \eta(t_g, t_{amb}, I) \tag{2}$$

where, I is the total solar radiation incident over the plane of the collectors.

2.3. Characteristic equation of the absorption cooling machine

The deduction of this equation is due to Ziegler and can be found elsewhere (see [4] and [1]). However for completeness and easy reading of the paper, a brief summary is included. The following set of equations is used to model the external heat exchange in each part of the machine. The UA values depend on the technology employed and include implicitly the mass transfer capacity in absorber and generator.

$$Q_e = UA_e \cdot z_e \cdot (t_e - T_e) \tag{3}$$
$$Q_c = UA_c \cdot z_c \cdot (T_c - t_c) \tag{4}$$
$$Q_a = UA_a \cdot z_a \cdot (T_a - t_a) \tag{5}$$
$$Q_g = UA_g \cdot z_g \cdot (t_g - T_g) \tag{6}$$

where

t mean internal temperature.

T mean external temperature.

z corrector factor .

The Dühring's law for the mixture LiBr-H2O is modelled by linear equations where B_o and B are coefficients that depend on the salt concentration (x). The isostere equations written for the concentration of the diluted line (also named weak, that goes from absorber to generator) and of the concentrated line (also named strong, that goes from generator to absorber) are:

$$T_a = B_o(x_a) + B(x_a)\,T_e \tag{7}$$
$$T_g = B_o(x_g) + B(x_g)\,T_c \tag{8}$$

By subtracting both :

$$T_g - T_a = B \cdot (T_c - T_e) \tag{9}$$

(B is taken in the literature as 1.15).

Substituting equations (3-6) into (9), the following relationship for the external mean temperatures is reached:

$$\frac{Q_g}{UA_g \cdot z_g} + \frac{Q_a}{UA_a \cdot z_a} + B \cdot \left(\frac{Q_c}{UA_c \cdot z_c} + \frac{Q_e}{UA_e \cdot z_e}\right) = \Delta\Delta t \tag{10}$$

where:

$$\Delta\Delta t = t_g - t_a - B \cdot (t_c - t_e) \tag{11}$$

This equation (11) is also known as the double delta t. The physical idea behind would be that of a "*thermodynamical driving force*" that allows the machine to produce the cooling effect.

The energy balance equation for each element of the absorption cycle, taking care of joining the absorber and the generator with one side of the solution heat exchanger, are:

$$\dot{Q}_e = \dot{m}_{ref} \cdot \left(h_{vapor(e)} - h_{liquid(c)}\right) \tag{12}$$
$$\dot{Q}_c = \dot{m}_{ref} \cdot \left(h_{vapor(g)} - h_{liquid(c)}\right) \tag{13}$$
$$\dot{Q}_a = \dot{m}_{ref} \cdot h_{vapor(e)} + \dot{m}_{strong} \cdot h_{strong(g)} - \dot{m}_{weak} \cdot h_{weak(a)} - Q_{hex} \tag{14}$$
$$\dot{Q}_c = \dot{m}_{ref} \cdot h_{vapor(g)} + \dot{m}_{strong} \cdot h_{strong(g)} - \dot{m}_{weak} \cdot h_{weak(a)} - Q_{hex} \tag{15}$$

where, Q_{hex} represents the heat exchanged between the strong and weak solution. Clearing from the previous equations the heat released during condensation, evaporation and generation:

$$\begin{aligned}
\dot{Q}_c &= \frac{h_{vapor(g)} - h_{liquid(c)}}{h_{vapor(e)} - h_{liquid(c)}} \cdot \dot{Q}_e = C \cdot \dot{Q}_e && (16)\\
\dot{Q}_e &= \frac{h_{vapor(e)} - h_{liquid(g)}}{h_{vapor(e)} - h_{liquid(c)}} \cdot \dot{Q}_e + \dot{m}_{weak} \cdot \left(h_{strong(g)} - h_{weak(a)}\right) - Q_{hex} && (17)\\
&= A \cdot \dot{Q}_e + Q_{loss} \\
\dot{Q}_g &= \frac{h_{vapor(g)} - h_{liquid(g)}}{h_{vapor(e)} - h_{liquid(c)}} \cdot \dot{Q}_e + \dot{m}_{weak} \cdot \left(h_{strong(g)} - h_{weak(a)}\right) - Q_{hex} \\
&= G \cdot \dot{Q}_e + Q_{loss} && (18)
\end{aligned}$$

where, Q_{loss} represents an efficiency loss of the cycle, more concretely that part of the heat delivered to the solution at the generator that goes or bypasses (due to a non-ideal solution heat exchanger and non-ideal cycle) to the absorber, like a heat flux short circuit. The G,C,A letters are quotients of enthalpies. Their values depends on the operating conditions, nevertheless its variation is small and in the model they are assumed constant.

Combining equations (3,4,5,6 and 16,17,18) an expression for the size parameter s is reached. It accounts, roughly, for the total amount of UA spent in building the machine (but for the solution heat exchanger), and gives and idea of the size of the machine:

$$s = \frac{1}{\frac{G}{UA_g z_g} + \frac{A}{UA_a z_a} + B\left(\frac{C}{UA_c z_c} + \frac{1}{UA_e z_e}\right)} \tag{19}$$

Another parameter appears α which indicates how was done the distribution of the heat transfer capacity among the evaporator, absorber, generator and condenser. Notice that generator and absorber sizes are compared with the total size s.

$$\alpha = \left(\frac{1}{UA_g z_g} + \frac{1}{UA_a z_a}\right) \cdot s \tag{20}$$

Both parameters appear in the expression for the cooling capacity as a function of the double delta t. This is one of the characteristic equations:

$$\dot{Q}_e = s \cdot \Delta\Delta t - \alpha \cdot \dot{Q}_{loss} = s \cdot (\Delta\Delta t - \Delta\Delta t_{min}) \tag{21}$$

the Q_{loss} has been modelled as $s \cdot \Delta\Delta t_{min}$, this minimum double delta t means a minimum thermodynamical driving force or a “friction loss”. Its value may change with the thermal load of the machine (or $\Delta\Delta t$), see [1].

From (21), it is clear the relationship between $\Delta\Delta t$ and the cooling capacity. The stronger the force, the higher the cooling power, (see figure 1).

The efficiency or COP of a single effect machine is written according to the model as:

$$COP = \frac{\dot{Q}_e}{\dot{Q}_g} = \frac{\dot{Q}_e}{G \cdot \dot{Q}_e + Q_{loss}} = \frac{\Delta\Delta t - \Delta\Delta t_{min}}{G \cdot \Delta\Delta t + \left(\frac{1}{\alpha} - G\right) \Delta\Delta t_{min}} \tag{22}$$

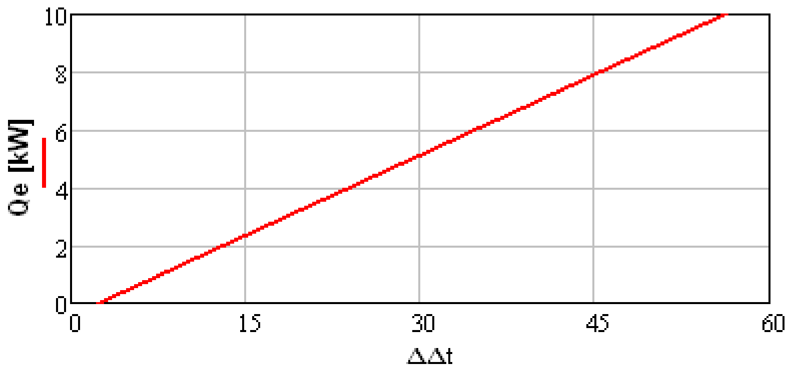

Figure 1: Cooling capacity as a function of $\Delta\Delta t$

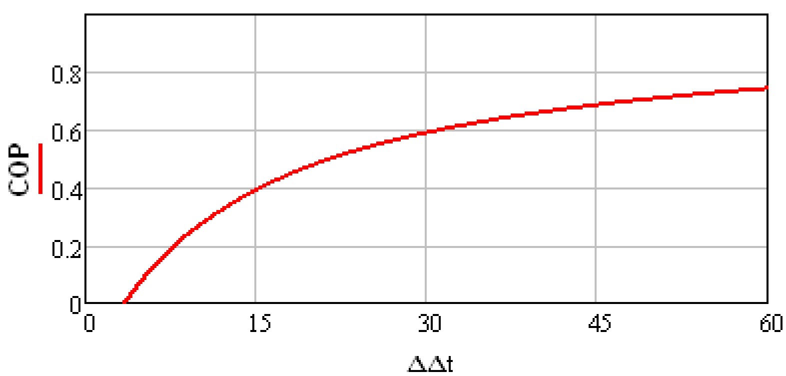

Figure 2: COP of a single effect machine as function of $\Delta\Delta t$

Once more, the efficiency is also related to the $\Delta\Delta t$. The greater the "thermodynamic force", the higher the efficiency (although the COP shows a plateau) (see figure 2).

These two equations represent the characteristic equations of a single effect machine. If a real machine is taken with data from two known working conditions then four equations can be posed; {COP, Qe} at each working point. Out of these four equations, the next set of four parameters can be discovered; $\{\alpha , s , \Delta\Delta t_{min,1}, \Delta\Delta t_{min,2}\}$. If data from a catalog are used, it can be shown that α and both $\Delta\Delta t_{min}$, are the same for all the catalog. They only change with the scale, that is, only s grows with the nominal cooling capacity of the machine the rest are the same and include technological aspects.

2.4. Optimal design of an endorreversible absorption cooling machine

This section has been included to gain insight into the reasonable values of the parameters of the characteristic equation as well as its relationship with the optimal design of absorption cooling machines. Mainly it is sought the answer to the following question: *What is the best UA distribution of the external heat exchangers (evaporator, absorber, condenser and generator) for an endorreversible machine?*. The losses due to the external heat exchanges are the main ones, therefore it is interesting to find out what occurs for cycle which works ideally inside.

Trozer [9] formulates the hypothesis of an ideal endorreversible absorption cycle. He starts from a "real" absorption cycle in a T-S diagram and looks for the minimum number of hypothesis that make the cycle internally reversible (i.e. no entropy production). These are:

- Specific heat capacity of the saturated liquid is negligible.
- The evaporation heat is constant and independent of the temperature.
- The dissolution heat changed only with salt concentration.
- The expansion of the refrigerant is isoentropic.
- The specific heat capacity of the solution is negligible.
- The area of the solution heat exchanger is infinite.
- The mass flow rate of solution between the absorber and the generator, is infinite. This in turn means that the concentration of the strong and weak solution are equal.
- The entropy of mixing is negligible with respect to the evaporation enthalpy of the refrigerant.
- The refrigerant vapor is an ideal gas.
- The specific heat capacity of the reheated vapor coming from the generator is negligible.
- The vapor pressure of the absorbent (the salt) is nil.
- The circulation of the solution mass flow rate consumes no power.

Since the cycle does not produce entropy internally (i.e. entropy increase rate at heat inputs= entropy decrease rate at heat outputs) , the internal entropy balance led Tozer to a simple thermodynamic relationship of temperatures (written in [K]):

$$T_e T_g = T_c T_a \tag{23}$$

This relationship is fulfilled for an endorreversible machine and that means that the levels of temperature cannot be chosen freely since there is a relationship among them. Assuming an infinite UA in the external heat exchangers, or in other words, that the mean external and internal temperatures are the same, the COP for this cycle was shown by Tozer to be:

$$COP = \frac{T_e}{T_c} = \frac{T_a}{T_g} \tag{24}$$

What occurs with the COP if finite UAs are used instead, while keeping the internal cycle ideal according to Tozer hypothesis?.

In general the entropy flux going out the cycle equals the entropy generated inside the machine plus that generated at the external heat exchangers:

$$\frac{Q_a}{t_1} + \frac{Q_c}{t_1} - \frac{Q_g}{t_2} - \frac{Q_e}{t_0} = \Delta S_{total} \tag{25}$$

However, the entropy generated internally is:

$$\frac{Q_a}{T_a} + \frac{Q_c}{T_c} - \frac{Q_g}{T_g} - \frac{Q_e}{T_e} = \Delta S_i \tag{26}$$

This last equation (26) in our case equals cero, since we assume the ideal internal cycle. In general the difference between both (25) and (26) represents the entropy generated at the external heat exchanges. In this case this is the total entropy generated by the machine. The irreversibility I may be then expressed as:

$$\begin{aligned} I &= \Delta S_{ext} \cdot t_1 = (\Delta S_{total} - \Delta S_i) \cdot t_1 = \\ &= Q_g \cdot \left[\left(1 - \frac{t_1}{t_2}\right) - \left(1 - \frac{t_1}{T_g}\right)\right] + Q_c \cdot \left(1 - \frac{t_1}{T_c}\right) + Q_a \cdot \left(1 - \frac{t_1}{T_a}\right) + Q_e \cdot \left[-\left(1 - \frac{t_1}{T_e}\right) + \left(1 - \frac{t_1}{t_0}\right)\right] \end{aligned} \tag{27}$$

The best COP efficiency of the machine is reached when I is at a minimum. However there are minimization constraints. For instance the total amount of UA installed should be fixed in advance. Following Tozer [8] the total heat exchange capacity, is given by:

$$\begin{aligned} K = A_g U_g + A_c U_c + A_e U_e + A_a U_a = \tfrac{Q_g}{\Delta T_g} + \tfrac{Q_c}{\Delta T_c} + \tfrac{Q_e}{\Delta T_e} + \tfrac{Q_a}{\Delta T_a} \\ K = \tfrac{Q_g}{(t_g - T_g)} + \tfrac{Q_c}{(T_c - t_c)} + \tfrac{Q_e}{(t_e - T_e)} + \tfrac{Q_a}{(T_a - t_a)} \end{aligned} \tag{28}$$

Other constraints are: the thermodynamic relationship of temperatures (23), the desired cooling capacity and the mean external temperatures at the design operating conditions chosen by the designer.

$$Q_c = Q_e = 1\,kW \tag{29}$$

$$\begin{aligned} t_0 &= t_0{}^{design} \\ t_1 &= t_1{}^{design} \\ t_2 &= t_2{}^{design} \end{aligned} \tag{30}$$

and using the COP (24) we get the other heat fluxes:

$$Q_a = Q_g = \frac{1\,kW}{COP} = \frac{T_g}{T_a} \tag{31}$$

Subtituting the heat fluxes into (27) and into the total UA constraint (28) the minimization problem can be posed as :

$$I(\beta, T_g, T_a) = \beta^{-1} \cdot t_1 \cdot \left(\frac{1}{T_g} - \frac{1}{t_2^{design}}\right) + \left(1 - \frac{t_1^{design}}{\beta \cdot T_g}\right) + \beta^{-1} \cdot \left(1 - \frac{t_1^{design}}{T_a}\right) + t_1 \cdot \left(\frac{1}{\beta \cdot T_a} - \frac{1}{t_0^{design}}\right) \tag{32}$$

$$f_k(\beta, T_g, T_a) = \frac{\beta^{-1}}{\left(t_2^{design} - T_g\right)} + \frac{1}{\left(\beta \cdot T_g - t_1^{design}\right)} + \frac{\beta^{-1}}{\left(T_a - t_1^{design}\right)} + \frac{1}{\left(t_0^{design} - \beta \cdot T_a\right)} - K = 0 \tag{33}$$

where:

$$\beta = \frac{T_c}{T_g} \tag{34}$$

$$\beta^{-1} = \frac{T_a}{T_e} = Q_g = Q_a \tag{35}$$

After simplifying the above expression, it is show that (32) only depends on β, or in other words on how the internal temperatures adjust themselves according to the UA capacities. Finally the minimization problem is reduced to the following;

$$\begin{array}{c} min_{\beta,T_a,T_g}\; I(\beta) = \left(1 + \frac{1}{\beta}\right) - t_1^{design} \cdot \left(\frac{1}{\beta \cdot t_2^{design}} + \frac{1}{t_0^{design}}\right) \\ f_k(\beta, T_g, T_a) = \frac{\beta^{-1}}{\left(t_2^{design} - T_g\right)} + \frac{1}{\left(\beta \cdot T_g - t_1^{design}\right)} + \frac{\beta^{-1}}{\left(T_a - t_1^{design}\right)} + \frac{1}{\left(t_0^{design} - \beta \cdot T_a\right)} - K = 0 \end{array} \tag{36}$$

Solving the minimization problem by using the Lagrange multipliers, we get the following system of equations:

$$\begin{array}{c} \frac{\partial I}{\partial \beta} - \lambda \frac{\partial f_k}{\partial \beta} = 0 \\ \frac{\partial I}{\partial T_g} - \lambda \frac{\partial f_k}{\partial T_g} = 0 \\ \frac{\partial I}{\partial T_a} - \lambda \frac{\partial f_k}{\partial T_a} = 0 \\ f_k(\beta, T_g, T_a) = 0 \end{array} \tag{37}$$

It is solved for {β, Tg, Ta, λ}. It can be shown that the solution for λ forces, jointly with the temperature equation (23), the relationships $UA_e = UA_a$ and $UA_c = UA_g$. This is a general result -valid for any endorreversible cycle- and it can be stated: “For a closed endorreversible cycle the sum of all the UA at the heat inlets must equal the sum of all the UA at the heat outlets at the minimum generating entropy working point” (this agrees with [5]) . The expressions for the internal temperatures at the optimum point are:

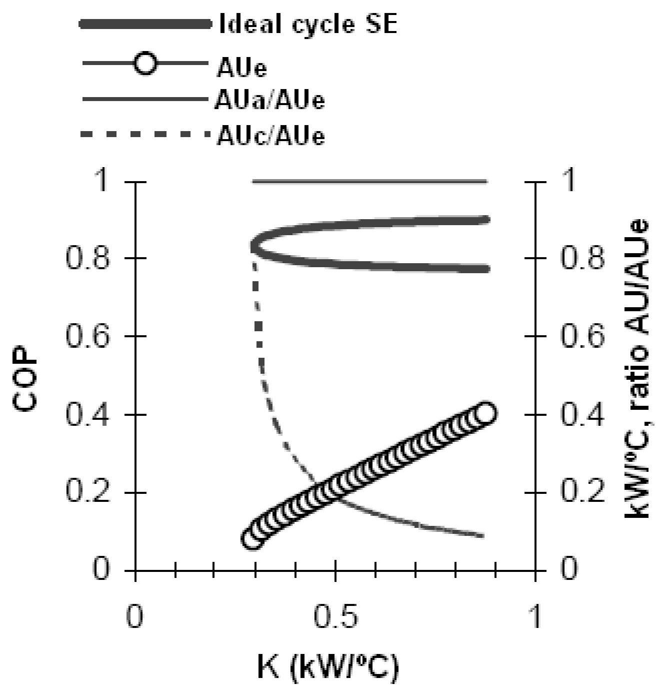

Figure 3: Optimal design of an endorreversible cycle. $t_0^{design} = 7^oC, t_1^{design} = 32^oC, t_2^{design} = 127^oC$

$$T_g = \frac{\beta \cdot t_2^{design} + t_1^{design}}{2 \cdot \beta} \tag{38}$$

$$T_a = \frac{\beta \cdot t_1^{design} + t_0^{design}}{2 \cdot \beta} \tag{39}$$

Finally if these values are substituted into (33) the following second degree equation in β is obtained with K as a parameter:

$$t_1^{design} \cdot t_2^{design} \cdot \beta^2 + \left[\frac{4}{K}\left(t_2^{desin} - t_1^{design}\right) - \left(t_0^{design} \cdot t_2^{design} + (t_1^{design})^2\right)\right]\beta + \left[\frac{4}{K}\left(t_0^{design} - t_1^{design}\right) + t_0^{design} \cdot t_1^{design}\right] = 0 \tag{40}$$

After solving for each root β of (40) and using the equations (3-4-5 and 6) all the UA of the components are obtained, (see figure 3). It can be observed that the greater the K the smaller is the ratio $UAg/UAa = UAc/UAe$, that is, relatively more exchange capacity should be allocated at the bottom elements of the cycle (absorber and evaporator) than in the top ones (condenser and geneator) as K increases.

Notice that there are two values of β that represent the best and the worst design which make the COP curve in figure (3) to have the shape of a horse shoe. Other studies that used a non-endorreversible model (see Summerer [7]) reach the same conclusion. Summerer got a minimum

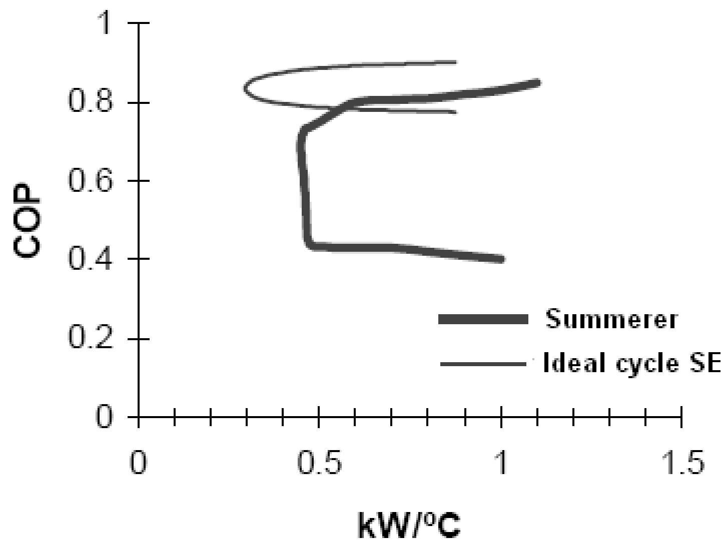

Figure 4: Comparison COP, endorreversible and non-endorreversible single effect absorption cycles $t_0^{design} = 7^{\underline{o}}C, t_1^{design} = 32^{\underline{o}}C, tde_2^{sign} = 127^{\underline{o}}C$

COP branch, which means that it is possible to do the worst UA distribution for a total K. The Summerer curves have also a horse shoe shape, but are located below our curve (see figure 4) since our cycle is internally ideal, with no internal losses like the cycles employed by Summerer.

The horse shoe shape also indicates that there is a minimum K ($K > K_{min}$) in order to get a real solution of β. The physical meaning is that to achieve the cooling rate fixed beforehand at the chosen design working point, it is needed a minimum amount of total heat exchange capacity Kmin.

2.5. Characteristic equations for an endorreversible cycle.

Once we have gained some insight into the optimal design of an endorreversible cycle, it would be interesting to work out what would be the value of the model parameters in the characteristic equations. As it has been shown the design parameters of the characteristic equations are: on one side $\{s, \alpha\}$ and on the other, the internal losses $\Delta\Delta t_{min}$. Taking into account that for an internally ideal cycle $\Delta\Delta t_{min} = 0$, the optimal design obtained previously leads to the following simplifications.

Putting equation (19) into (20), and taking into account that $UA_g = UA_c$ and $UA_e = UA_a$ while assuming that the factors z in (20) are close to 1, the α parameter of the distribution of UA, transforms into:

$$\begin{aligned} \alpha &= \frac{1 + rc}{G + A \cdot rc + B \cdot C + B \cdot rc} \\ &= \frac{1 + rc}{(G + B \cdot C) + (A + B) \cdot rc} \end{aligned} \tag{41}$$

where, $rc = UA_g/UA_a$

Here, an approximated value for the constants {A,B,C,G} is needed. If it is assumed a mean external temperature of 90ºC at the generator, 30ºC at the saturated liquid at the condenser outlet and 5ºC the saturated vapor at the evaporator outlet, the value of the constants would be: $G = 1.02$, $A = 0.963$, $C = 1.06$ and $B = 1.15$.

As it has been shown in the previous section, the optimal rc ratio decreases with the total K installed, however in a big range of rc values, the α in equation (41) varies slightly between 0.44 and 0.47 as K increases (or in other words when a more expensive machine is built).

In a real machine (non-endorreversible one) α would be a design parameter used, in practice, to try to compensate for the internal losses, represented by the evolution of $\Delta\Delta T_{min}$. Consequently, in the case of $\Delta\Delta T_{min} \backsimeq 0$ (minor internal losses) , it could be assumed that α is around 0.5.

Notice that from equation (24), if α decreases, for a certain $\Delta\Delta T_{min}$, then the COP gets worse.

Therefore, the values of α chosen in this paper are in the range between $\alpha = 0.2$ and $\alpha = 0.5$. They correspond to a bad and a good machine design, assuming a fix term for the losses $\Delta\Delta T_{min}$. In Coronas et al. [1], it may be found a study about the empirical values of $\Delta\Delta T_{min}$ for real machines. Coronas et al. found $\Delta\Delta t_{min}$ values in the range between 1.59 and $5.22\,K$ and also found $\alpha = 0.61$ which , according to our previous discussion, indicates a quite good design. So, in what follows, it has been chosen a $\Delta\Delta T_{min}$ fixed at $\Delta\Delta T_{min} = 3.5\,K$ which is realistic quite low value of the internal losses.

2.6. Coupling the solar field and the machine

The characteristic equations (19) and (21), are used to find the power needed at the generator as:

$$\dot{Q}_g = \frac{\dot{Q}_e}{COP} = \frac{s\cdot(\Delta\Delta t - \Delta\Delta t_{min})}{\frac{\Delta\Delta t - \Delta\Delta t_{min}}{G\cdot\Delta\Delta t + \left(\frac{1}{\alpha} - G\right)\Delta\Delta t_{min}}} = s\left(G\cdot\Delta\Delta t + \left(\frac{1}{\alpha} - G\right)\Delta\Delta t_{min}\right) \tag{42}$$

This power (42) must equal the power coming from the solar field (2) to allow them both to work coupled. The balance equation is written as:

$$A_{col} I\left(FR_{z\alpha n} - \frac{FRU_L}{1000}\cdot\left(\frac{t_g^{eq} - t_{amb}}{I}\right)\right) = s\left(G\cdot\Delta\Delta t + \left(\frac{1}{\alpha} - G\right)\Delta\Delta t_{min}\right) \tag{43}$$

We define ψ, as the ratio of the area of the solar collector field A_{col} to the size of the absorption machine s.

$$\psi = \frac{A_{col}}{s} \tag{44}$$

Substituting into (43):

$$FR_{z\alpha n} - \frac{FRU_L}{1000}\cdot\left(\frac{t_g^{eq} - t_{amb}}{I}\right) = \frac{1}{\psi\cdot I}\cdot\left(G\cdot\Delta\Delta t + \left(\frac{1}{\alpha} - G\right)\Delta\Delta t_{min}\right) \tag{45}$$

Using (45) and (11) the expression for $\Delta\Delta t$ we get:

$$t_g^{eq} = t_{amb} + \frac{I\cdot 1000}{FRU_L}\cdot\left[FR_{z\alpha n} - \frac{1}{\psi\cdot I}\cdot\left(G\cdot\left(t_g^{eq} - t_a - B\cdot(t_c - t_e)\right) + \left(\frac{1}{\alpha} - G\right)\Delta\Delta t_{min}\right)\right] \tag{46}$$

Finally clearing for t_g^{eq} the coupled system is modelled by:

$$t_g^{eq} = \frac{FRU_L \cdot \psi}{FRU_L \cdot \psi + 1000 \cdot G} \cdot \left[t_{amb} + \frac{1000}{FRU_L} \cdot \left(I \cdot FR_{z\alpha n} + \frac{G}{\psi} t_a + \frac{G \cdot B}{\psi} (t_c + t_e) - \left(\frac{1}{\alpha} - G \right) \frac{\Delta\Delta t_{min}}{\psi} \right) \right] \quad (47)$$

The temperature t_g^{eq} is the balance or equilibrium temperature which allows all the heat power from the solar field to be equal to the generator needs while both are assumed to be working in a steady state.

We must add to (47), the following restrictions: the t_g^{eq} will not be higher than 110ºC and will not be lower than 75ºC since the machine cannot start.

In short:

if $t_g^{eq} \geq 110^{\text{o}}C$

$$\dot{Q}_e = \frac{1}{\psi} \cdot A_{col} \left[(110 - t_a) - B \cdot (t_c - t_e) - \Delta\Delta t_{min} \right]$$

if $75^{\text{o}}C \leq t_g^{eq} \leq 110^{\text{o}}C$

$$\dot{Q}_e = \frac{1}{\psi} \cdot A_{col} \left[\left(t_g^{eq} - t_a \right) - B \cdot (t_c - t_e) - \Delta\Delta t_{min} \right]$$

if $t_g^{eq} \leq 75^{\text{o}}C$

$$\dot{Q}_e = 0$$

3. Application and results

At this point an answer to the following question is sought: *What is the effect of the evaporation temperature, the ratio ψ or solar covering of a certain size of a machine and its design parameter α, on the peak cooling power and on the whole annual cooling effect and its distribution along the year?*. The answer to this question comes first and then, in a second part, as an outcome, a brief sizing method is given.

The model has been applied assuming:

- s is constant. Just a machine of one concrete size. In order to keep s fixed, the ψ varies according to A_{col}, (see 44). It has been chosen $s = 0.1\,kWK^{-1}$. Such a machine for $\alpha = 0.5$, $t_e = 5^{\text{o}}C$, $t_c = 28^{\text{o}}C$, $t_a = 30.5^{\text{o}}C$, $t_g = 90^{\text{o}}C$, $\Delta\Delta t_{min} = 3.5$ has a cooling capacity of $2.92\,kW$ so it is a very small one.

Here is a resume of the values of the parameters employed in the models:

- Location
  - A mean Julian year has been used. Not the coldest neither the warmest, but an expected mean year. The data comes from the Industry and Trade Ministry of Spain [2]. The files contain hourly data (8760 values) on: dry ambient temperature (t_{amb}), beam and diffuse radiation over horizontal surface, wet bulb temperature (t_a).
  - Geographical coordinates (latitude, longitude).
- Solar collectors

- The solar field was oriented to optimize the collection during summer. They have a tilt $15^{\underline{o}}$ less than the latitude of the site. ($FR_{\tau\alpha} = 0.825$, $FRU_L = 1.1\,WK^{-1}m^{-2}$, and correspond to a vacuum solar collector).
- The azimuth is to the south.
- The area (A_{col}), was changed from $5\,m^2$ to $70\,m^2$ in order to keep the size of the machine fixed.
- The solar coverage ψ changed from 50 to $700\,m^2K\,kW^{-1}$. The lower value is the minimum solar coverage that allows the direct coupling of the collectors and the machine. It varies from place to place. In the case of Valencia is 50. The warmer the site the smaller needs to be this value. Smaller values than this minimum, lead to t_g^{eq} temperatures that never go above the minimum $75^{\underline{o}}C$ and no direct coupling is possible.

• The absorption machine

- $\Delta\Delta t_{min}$ and α, as seen previously, $\Delta\Delta t_{min} = 3.5[K]$ and $\alpha = 0.2$, $\alpha = 0.5$.
- The condensation temperature (t_c) will be equal to the instantaneous wet bulb temperature plus $2.5^{\underline{o}}C$.
- The evaporator mean external temperature (t_e), was fixed to $5^{\underline{o}}C$ in case of fan-coils, and to $14^{\underline{o}}C$ in case of cooling ceilings or floors .
- The generator mean external temperature (t_g) equals the equilibrium temperature t_g^{eq} (47).
- The absorber mean external temperature (t_a) equals the instantaneous wet bulb temperature.

The picture (5) shows the hourly simulation done for Valencia with ($\psi = 80\,m^2K\,kW^{-1}$, $A_{col} = 8\,m^2$, $\alpha = 0.5$, $t_e = 5^{\underline{o}}C$) of the behavior of the equilibrium temperature (t_g^{eq}). It is obvious that the middle of the graph corresponds to the summer, where this temperature is higher.

Only when t_g^{eq} is above $75^{\underline{o}}C$, the machine produces cooling.

The figure(6) shows, using the computed t_g^{eq}, (see 5) the cooling power at each hour. The maximum takes place at 12 hrs as expected and has a peak value of $5,07\,kW$. The machine is able to produce cooling during the months of: May, June, July , August and September to a total amount of $1667.7kW\,h$ ($6003.6\,MJ$).

If in the previous case the evaporation temperature t_e is raised from 5 to $14^{\underline{o}}C$, the results for the hourly cooling capacity are shown in (8). The peak cooling power rises a bit until 5,16 kW , however the total production of cooling decreases to $668{,}8\,kWh$ ($2407.5MJ$). This trend has been observed for all the Spanish climates. The reason could be summarized as follows: when the machine is producing water at $5^{\underline{o}}C$ the mean temperature of equilibrium of the generator with the solar field is higher than when it is produced at 14ºC. This is noticed when comparing (5) with (7). The cause of this seemingly contradiction, is that at higher evaporating temperatures the absorption machine needs a higher heat input at the generator at a certain temperature, however, due to the solar coupling, the collector field is not able to collect more heat at that temperature and as a result the t_g^{eq} decreases searching for a new energy balance. The final result is a reduction in the number of annual hours when the machine is able to produce cooling. On the other side, as a general rule the highest $\Delta\Delta t$ obtained when evaporating at 14 ºC, is higher than when doing it at 5ºC and that is the reason for the higher peak power.

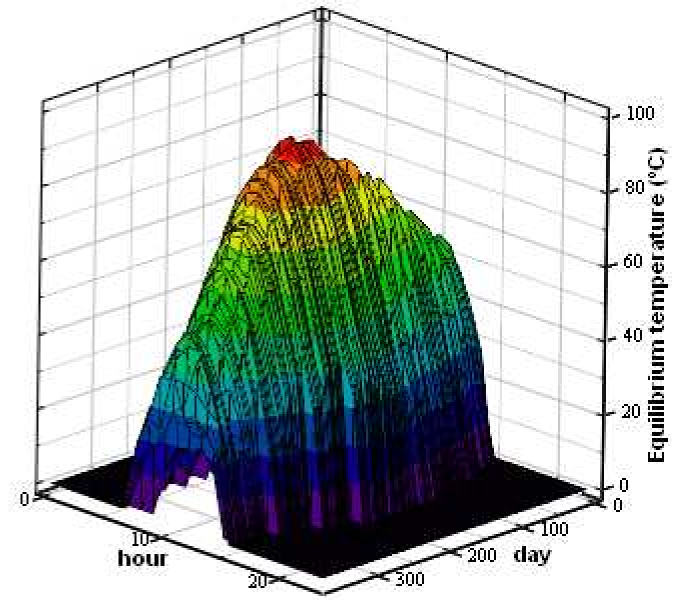

Figure 5: Expected annual evolution of the equilibrium temperature (t_g^{eq}),for Valencia with $\alpha = 0.5$, $\psi = 80\,m^2K\,kW^{-1}$, $A_{col} = 8\,m^2$, $t_e = 5^oC$ as a function of the julian day and solar hour.

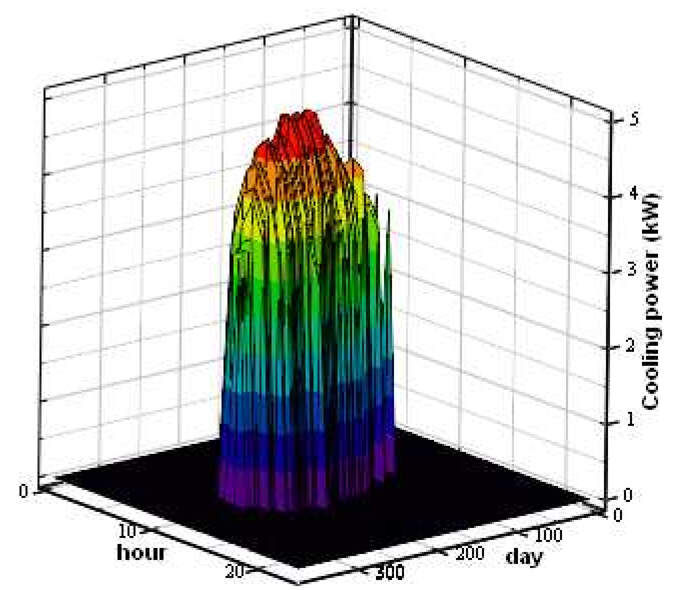

Figure 6: Hourly cooling power [kW], for Valencia $\alpha = 0.5$, $\psi = 80\,m^2K\,kW^{-1}$, $A_{col} = 8\,m^2$, $t_e = 5^oC$ as a function of the Julian day and solar hour.

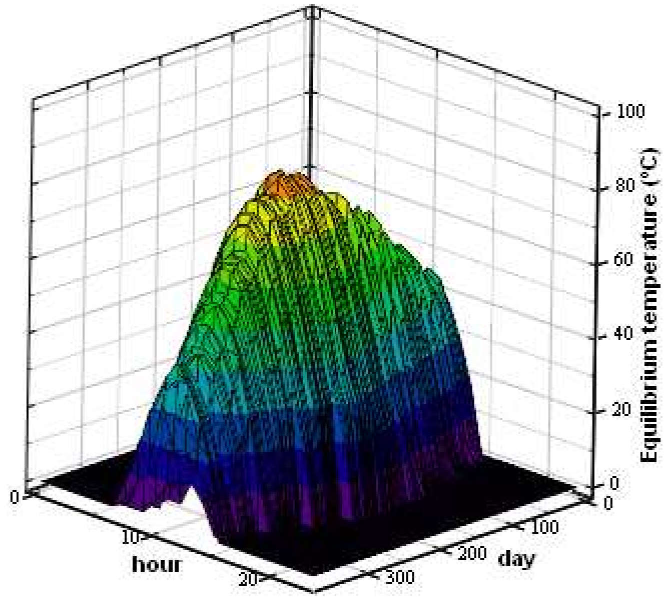

Figure 7: Equilibrium temperature t_g^{eq}, for Valencia with $\alpha = 0.5$, $\psi = 80\,m^2K\,kW^{-1}$, $A_{col} = 8\,m^2$, $t_e = 14^{\circ}C$ as a function of the Julian day of the year and the solar hour.

Additionally it is observed that in the first case (evaporating at 5ºC) it is possible to start the machine early in the morning and to stop it latter in the afternoon.

Table (3) shows the total or annual cooling capacity and the peak cooling capacity for the same conditions as before for different climatic zones of Spain.

The figures 9 and 10 show the annual distribution of the cooling capacity for Valencia when ψ rises. A high solar size with respect to the machine size, increases the number of hours per year that the machine can work directly coupled, even in winter. Moreover, depending on the site the solar field production gets over-sized at a certain ψ and then the production of cooling during summer becomes flat (corresponding to the limiting value of $t_g^{eq} = 110\,^{\circ}C$). After this point the machine behavior and the solar field becomes decoupled in summer, that is, an increase in the evaporator temperature produces annually more cooling since the generator temperature is not altered. The trend then is changed; annual cooling and peak cooling power are: $11186\,kW\,h$ ($402629.6\,MJ$) and $6.7\,kW$ for the case $t_e = 5\,^{\circ}C$ and $11331.6\,kW\,h$ ($4079376\,MJ$) and $7.8\,kW$ for the case $t_e = 14\,^{\circ}C$.

In figure (11) the annual cooling is shown as a function of ψ, using two different evaporator temperatures and the two values of the αdesign parameter. There are two outstanding points; on one side it is possible to quantify the minimum amount of area of the solar field needed for a certain machine size, i.e. ψ, to make possible the direct coupling, on the other, the annual cooling is quite lower for poor designs (low α) while ψ is fixed but this decrement is reduced for $\psi_{limit} < \psi$ where ψ_{limit} depends on the climate. Let's put two extreme examples: the difference in the minimum amount of solar area between a low evaporator temperature and a good machine design (i.e. high α) and a high evaporator temperature and a bad design (i.e. low α) for Valencia is $\psi = 50\,m^2K\,kW^{-1}$ and $\psi = 70\,m^2K\,kW^{-1}$ (40% more area in the second case) (see table 5). At the same location, regardless of the evaporator temperature, if $\alpha = 0.2$ then the annual cooling is 35 to 40 % lower

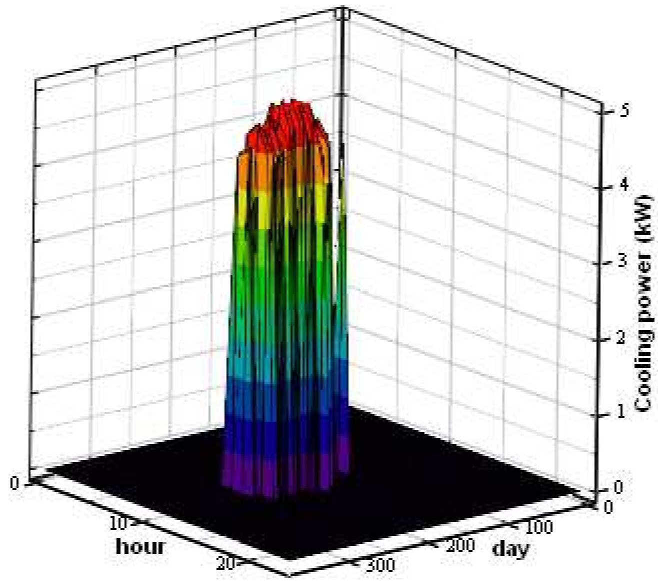

Figure 8: Hourly cooling power kW, for Valencia with $\alpha = 0.5$, $\psi = 80\, m^2 K\, kW^{-1}$, $A_{col} = 8\, m^2$, $t_e = 14^{\underline{o}}C$ as a function of the Julian day of the year and the solar hour.

Table 1: Peak hourly cooling capacity kW and annual cooling capacity $kW\,h$ (MJ) for different climatic zones of Spain, with $\psi = 80\, m^2 K\, kW^{-1}$, $A_{col} = 8\, m^2$,

	$t_e = 5^{\underline{o}}C$		$t_e = 14^{\underline{o}}C$	
	$\alpha = 0.5$		$\alpha = 0.5$	
	Annual cooling $kW\,h$ (MJ)	Peak power kW	Annual cooling $kW\,h$ (MJ)	Peak power kW
Almería	2245.9 (8805.1)	5.11	1175.2 (4230.6)	5.20
Barcelona	1269.4 (4569.9)	5.05	470 (1692.1)	5.14
Bilbao	301.6 (1085.9)	4.48	52.8 (190.1)	4.57
Burgos	677.2 (2438)	4.38	57 (205.2)	4.47
Cádiz	2450 (8819.8)	5.08	1064 (3765.7)	5.16
Granada	1318 (4744.8)	4.61	181.4 (653.1)	4.70
Madrid	805.1 (2898.4)	4.38	65.5 (235.7)	4.47
Sevilla	2173.5 (7824.5)	5.05	748.9 (2696.1)	5.14
Toledo	1473.3 (5304)	4.83	397.9 (1432.5)	4.91
Valencia	1667.7 (6003.6)	5.07	668.8 (2407.5)	5.16
Vitoria	476.2 (1714.2)	4.51	39.5 (142.2)	4.59
Zamora	775.2 (2790.6)	4.37	52.3 (188.2)	4.45

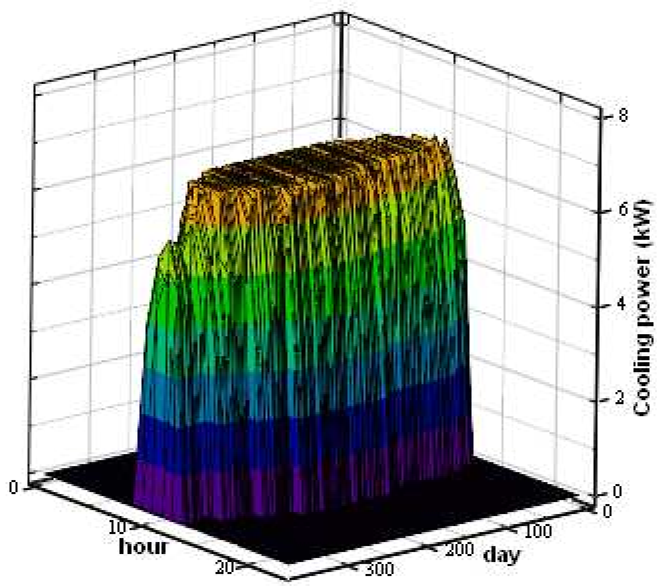

Figure 9: Hourly cooling power kW, for Valencia with $\alpha = 0.5$, $\psi = 200\,m^2K\,kW^{-1}$, $A_{col} = 20\,m^2$, $t_e = 5\ ^{\circ}C$ as a function of the Julian day and the solar hour.

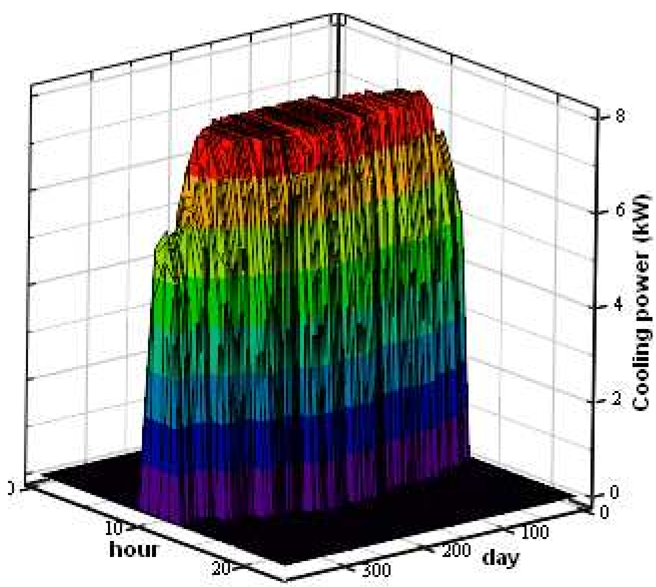

Figure 10: Hourly cooling power kW, for Valencia with $\alpha = 0.5$, $\psi = 200\,m^2K\,kW^{-1}$, $A_{col} = 20\,m^2$, $t_e = 14\ ^{\circ}C$ as a function of the Julian day and the solar hour.

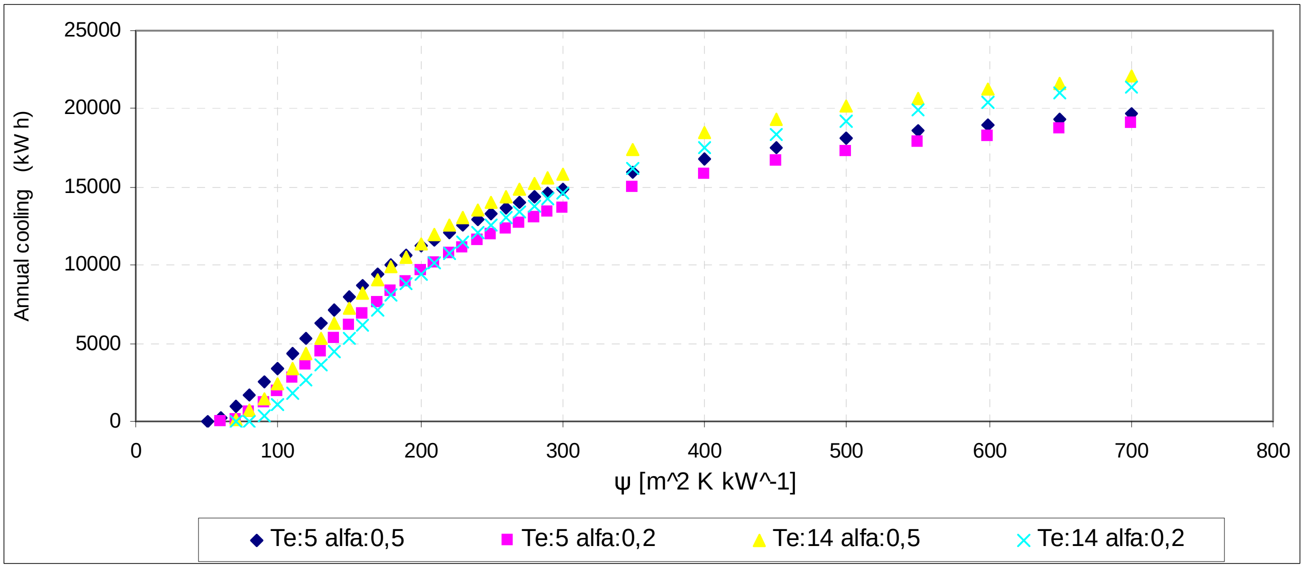

Figure 11: Annual cooling $kW\,h$ versus $\psi\,(m^2K\,kW^{-1})$, $t_e = 5^{\text{o}}C$ and $t_e = 14^{\text{o}}C$ and $\alpha = 0.2$and$\alpha = 0.5$ for Valencia

than at $\alpha = 0.5$ until the $\psi_{limit} = 200\,m^2K\,kW^{-1}$ is reached. Then on, the difference is just a 5% due to the big size of the solar area with respect to the machine size.

Another way to evaluate the system would be to divide the annual cooling with respect to the annual solar radiation on the solar collector field, just defining a sort of efficiency η_s:

$$\eta_s = \frac{\sum_{year} Q_e}{\sum_{year} I \cdot A_{col}} \tag{48}$$

The figure 11 of the annual cooling turn into efficiency converts into figure 12. It shows clearly the differences in efficiency between a good and a bad machine as well as the effect of oversizing of the solar field at high ψ values.

Finally there is the possibility of generalize the results providing a manual method to size quickly a system. The idea is to introduce the concept of total driving thermodynamic potential DDGH applied to the "thermodynamical force"$\Delta\Delta t$. The following expression shows its definition;

$$\begin{aligned} DDGH &= \frac{\sum_{year} \dot{Q}_e}{s} = \sum_{year} (\Delta\Delta t - \Delta\Delta t_{min}) = \\ &= \sum_{year} \left\{ \left(\left(t_g^{eq} - t_{wet_{bulb}}\right) - B\left(t_{wet_{bulb}} + 2.5 - t_e\right) \right) - \Delta\Delta t_{min} \right\} \end{aligned} \tag{49}$$

The meaning is the sum of all the driving potential which allows to produce cooling for a typical mean year and for a certain machine of unitary size $s = 1\,kW\,K^{-1}$ directly coupled to a certain type of the solar collector field. The DDGH depends on the type and orientation of the solar

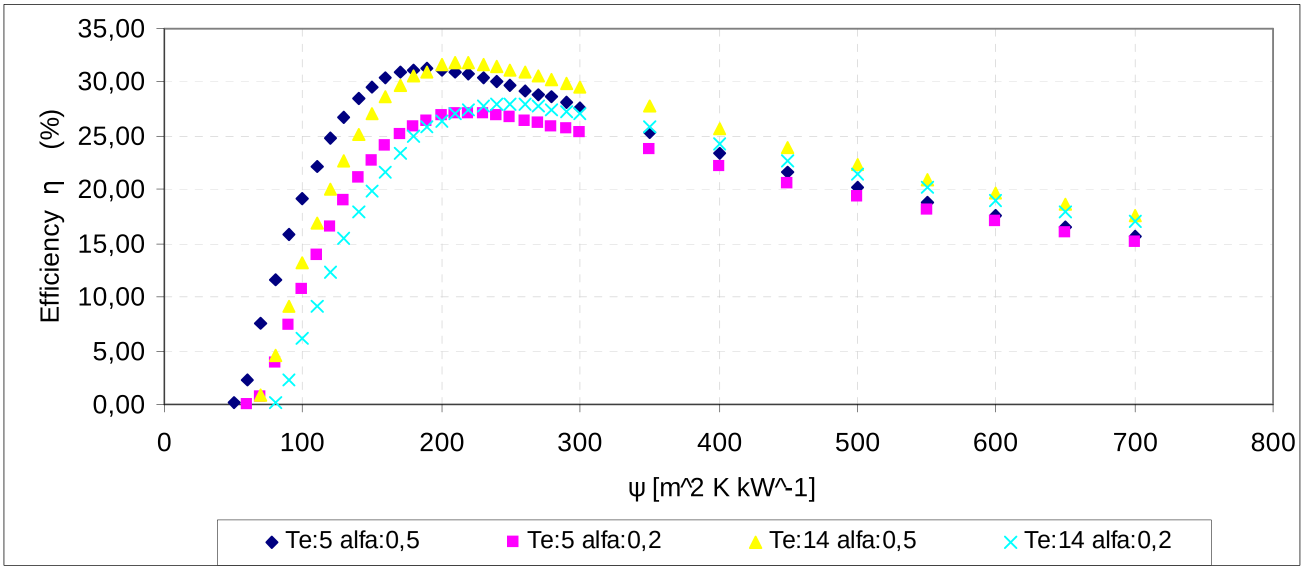

Figure 12: Efficiency of the direct coupled system as a function of $\psi\,(m^2K\,kW^{-1})$ for Valencia.

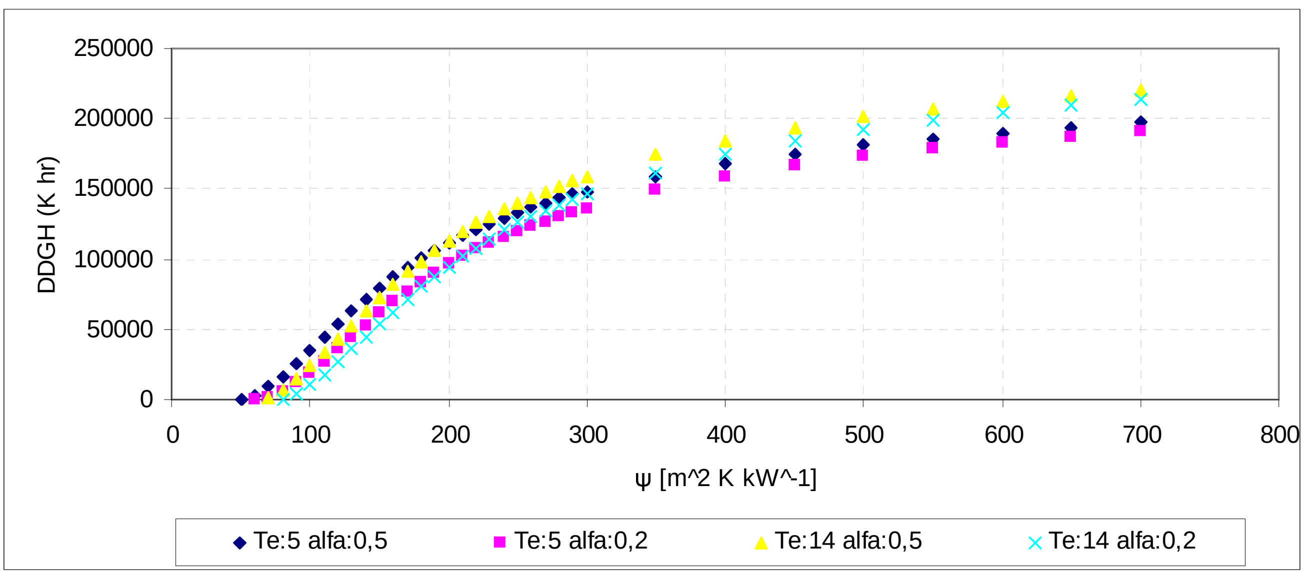

Figure 13: DDGH as a function of $\psi\,(m^2K\,kW^{-1})$, for Valencia.

Table 2: Climatic zones of Spain. Climatic severity.

4	Almería	Sevilla	Toledo		
3	Cádiz	Valencia	Granada	Madrid	
2			Barcelona	Zamora	
1			Bilbao	Vitoria	Burgos
	A	B	C	D	E

collectors, climatic zone, machine design and the coupling ratio ψ. (see 13 for a representation, result of dividing figure (11) by $s = 0.1\,kW\,K^{-1}$).

In the tables (3-14) the DDGH is show for the type of solar collector used in this paper, with a tilt 15º less than the latitude of the site and oriented to the south (azimuth=0). The Spanish normative uses a pair made up of one letter and one number to classify the climate of a site (see table2). The letter is used for winter and goes from A (less severe) to E (very cold), while the number is used for the summer and goes from 1 (mildest) to 4 (the hottest). The reference climate is that of Madrid.

Since ψ has a direct influence on the amount of annual cooling and its distribution through the year, the tables can be used to guess the size of a machine and that of the solar field to cover a given total cooling demand from a house. There are several combinations of machine size and solar field size which provide the same total annual cooling but with different distribution through the year. This will be covered in another article where the influence of the building will also be included.

4. Conclusions

A simple analytical model is obtained to study the direct coupling of a solar collector field with a single effect absorption cooling machine with the aim of obtaining the potential of solar cooling. Some insight has been gained on the meaning of some design parameters of the machine. The qualitative and quantitative trends have been obtained regarding the peak cooling power and the total annual cooling for a good and a bad design of the machine, as well as the ratio of solar area to the machine size (the Ψ coupling parameter) and also the effect of the evaporating temperatures. The concrete numerical values should be calculated in each case according to the methodology proposed.

The seemingly surprising negative effect of rising the evaporator temperature in the total annual cooling effect when there is a direct coupling has been pointed out.

Practical results are collected for the Spanish climatic zones. These are tables obtained with vacuum solar collectors, facing south and with a tilt 15 degrees less than the latitude of the site coupled to a bad and a good design of an absorption cooling machine. These tables can be used in sizing directly coupled solar cooling systems.

However it is needed an extra study about how to match the distribution of the cooling and perhaps other thermal energy demands (like heating) from a building along the year with the solar system. This article is thus the starting point for studying the coupling of active solar cooling systems into the buildings design and this will be the the subject of another article.

CLIMATIC ZONE "A3" (CADIZ)

DDGH (K h)

Ψ (m^2 K / kW)	Te: 5 (ºC)		Te: 14 (ºC)	
	α = 0.2	α= 0.5	α= 0.2	α = 0.5
50	-	-	-	-
60	-	4114,3	-	-
70	1296,3	13910,5	-	1649,8
80	8466,5	24499,5	253,9	10460,3
90	18227,2	36246,0	5718,0	22674,8
100	27707,2	48748,5	15790,9	33983,8
110	38859,4	59596,1	27328,6	47069,8
120	50210,4	71872,7	37445,4	60818,2
130	60345,1	84222,1	49958,7	72231,2
140	70850,2	94015,6	62261,3	84207,4
150	81920,3	103194,1	72815,6	97341,8
160	90639,1	111801,4	83506,1	107383,8
170	98608,7	119156,1	94749,5	116397,1
180	106850,9	125035,4	104059,1	125845,6
190	114183,7	130475,9	111847,9	134419,6
200	120253,7	135254,4	120642,3	141290,4
210	125745,9	139931,3	128617,8	147509,1
220	130445,4	144490,2	136448,7	152737,0
230	134707,7	148301,1	142262,4	157531,6
240	138520,7	152137,1	147708,1	161736,6
250	142363,4	155991,8	152796,8	166053,9
260	146026,9	159942,3	156772,9	170273,4
270	149113,7	162888,0	160539,6	173726,5
280	152425,4	165792,5	164397,2	177335,3
290	155637,1	167957,2	168208,1	181061,0
300	158559,8	170476,0	171568,1	184575,9
350	170385,7	180265,0	186582,7	198176,2
400	179362,7	188277,5	197989,8	208357,5
450	186453,9	194481,8	207383,2	216590,4
500	192040,4	199824,2	214896,7	223014,5
550	197196,7	204259,9	221020,4	228961,9
600	201546,2	207952,9	226328,8	233831,7
650	205043,8	210875,0	230863,2	237697,1
700	208331,3	213042,6	234965,5	241624,4

Table 3: DDGH for Cadiz

CLIMATIC ZONE "A4" (ALMERÍA)

DDGH (K h)

Ψ	Te: 5 (ºC)		Te: 14 (ºC)	
(m^2 K /kW)	α = 0.2	α = 0.5	α = 0.2	α = 0.5
50	-	119,9	-	-
60	248,5	5769,9	-	-
70	2326,0	14635,4	-	2906,4
80	9631,4	24458,5	592,0	11751,7
90	18667,5	36416,1	7654,5	23155,4
100	27647,5	46954,0	17097,0	33789,8
110	38593,4	58327,6	27073,2	46993,8
120	48731,5	69288,0	37883,6	58704,7
130	58820,9	80027,3	49247,2	70264,9
140	68275,8	90352,7	60283,2	81404,2
150	77456,7	98658,1	70628,4	91858,0
160	86991,8	106703,5	80188,3	102862,1
170	94948,3	113778,7	89898,9	112145,4
180	102165,8	119996,7	99184,3	120604,7
190	108809,5	126219,5	107664,6	128133,3
200	115094,2	131391,5	115964,7	135112,0
210	120669,8	136499,0	123686,0	141579,4
220	125913,6	141609,9	129877,3	147540,1
230	130374,7	145792,7	135915,9	152624,7
240	134735,5	149694,2	141779,1	157402,3
250	139033,5	153409,8	146979,2	162445,7
260	142959,7	157056,3	152024,9	166839,1
270	146878,5	160431,3	155808,5	171309,2
280	150011,3	163660,0	160273,7	174831,5
290	153289,2	166273,2	164114,6	178707,2
300	156114,0	169011,0	168053,4	181943,8
350	168992,6	179300,2	184129,1	196726,2
400	178338,7	187727,9	196756,8	207435,1
450	185956,7	193953,9	206750,5	216053,1
500	191912,2	199015,4	213882,6	222857,4
550	196656,6	203344,0	220585,2	228172,5
600	200969,6	206299,1	225641,3	233240,9
650	204246,2	209288,8	230294,6	236801,4
700	206761,9	212123,7	234469,4	239624,5

Table 4: DDGH, for Almeria

CLIMATIC ZONE " B3 " (VALENCIA)

DDGH (K h)

Ψ (m^2 K / kW)	Te: 5 (ºC)		Te: 14 (ºC)	
	α = 0.2	α = 0.5	α = 0.2	α = 0.5
50	-	89,75	-	-
60	-	2514,24	-	-
70	828,53	9433,77	-	1038,65
80	5554,68	16676,57	293,38	6687,54
90	11858,26	25649,53	3806,29	14678,83
100	19360,11	34400,59	10909,02	23693,10
110	27499,28	43741,26	18082,34	33379,92
120	35754,89	53443,16	26614,61	43226,85
130	44133,69	62540,90	35956,56	52956,60
140	52888,86	71430,58	44848,99	63055,52
150	61223,51	79729,93	53343,18	72877,58
160	69382,73	87187,94	62164,15	82281,77
170	76571,72	94301,89	71473,85	90795,20
180	83736,90	100761,37	80403,95	98697,13
190	89726,23	106723,04	87857,42	105604,01
200	96311,12	111860,38	94755,00	113316,08
210	101768,28	116522,22	101993,15	119700,97
220	107234,02	121350,08	108044,76	125835,68
230	111691,66	125592,90	114881,91	130883,28
240	115618,64	129399,14	120783,31	135350,30
250	119674,00	133136,62	125826,69	139874,45
260	123444,00	136534,77	130380,66	144116,97
270	126826,11	139902,29	134275,69	148093,02
280	130332,35	143652,54	137959,05	152139,65
290	133481,92	146084,83	142260,52	155747,60
300	136220,49	148508,05	145996,71	158811,90
350	149486,53	159252,62	161908,34	174239,02
400	158619,21	168177,96	174748,50	184633,13
450	166551,77	175302,74	183787,96	193605,05
500	173097,01	180975,84	192311,81	201155,62
550	178231,96	185395,85	198995,39	206888,23
600	182928,62	189738,33	204688,98	212396,14
650	186628,59	193585,68	209590,16	216603,54
700	190296,33	197421,62	213557,09	220630,43

Table 5: DDGH for Valencia

CLIMATIC ZONE "B4" (SEVILLA)

DDGH (K h)

Ψ (m^2 K / kW)	Te: 5 (ºC)		Te: 14 (ºC)	
	α = 0.2	α = 0.5	α = 0.2	α = 0.5
50	-	-	-	-
60	-	1991,67	-	-
70	663,95	10989,80	-	738,16
80	6133,15	21734,75	-	7489,22
90	16200,18	32581,65	3701,51	20064,29
100	25494,31	43465,65	13887,85	31443,10
110	35513,32	54171,96	25165,61	43349,11
120	45562,74	65704,34	35685,96	55010,20
130	55359,35	76277,66	46475,04	66058,81
140	65661,39	86133,42	57003,00	77912,34
150	74816,65	95330,36	67586,98	88619,82
160	83723,65	102826,73	77595,03	98839,74
170	91345,88	110820,17	87013,88	107778,10
180	98942,19	117550,48	96567,84	116614,89
190	105930,84	123062,15	104524,86	124583,34
200	112773,04	128320,51	112182,28	132395,39
210	118342,00	133030,56	119409,51	138984,10
220	123311,66	137230,63	127496,21	144533,76
230	128158,88	141080,66	134233,52	149911,43
240	132056,95	145457,58	139669,95	154327,94
250	135578,22	148979,03	144840,60	158446,72
260	139386,49	152884,09	149184,46	162531,36
270	142569,59	155967,15	153717,00	166265,81
280	145792,29	159254,53	157374,73	169829,70
290	149359,98	161986,73	160584,58	173828,70
300	152152,18	165141,05	164007,16	176982,51
350	165462,87	175897,08	179707,30	192598,32
400	175162,17	184501,60	192427,08	203634,73
450	182790,12	190769,56	203055,39	212549,04
500	188941,55	195708,31	211110,04	219514,80
550	193796,05	200278,89	217364,54	224925,07
600	197748,47	204062,42	222841,31	229312,91
650	201545,08	207439,04	227229,49	233677,73
700	204692,35	209662,11	237182,56	237182,56

Table 6: DDGH for Sevilla

CLIMATIC ZONE "C1" (BILBAO)

DDGH [K h]

Ψ (m^2 K / kW)	Te: 5 [ºC]		Te: 14 [ºC]	
	α = 0.2	α = 0.5	α = 0.2	α = 0.5
50	-	-	-	-
60	-	-	-	-
70	-	917,45	-	-
80	439,22	3016,28	-	528,06
90	1646,76	6657,36	-	2016,74
100	4213,07	10879,12	1147,60	5113,05
110	7815,25	15202,81	3385,38	9618,65
120	11312,56	19609,63	6438,89	13868,07
130	15238,52	23967,61	10479,22	18586,99
140	19184,07	27811,69	14394,41	23044,42
150	23164,65	32413,80	18645,90	27732,43
160	26777,57	36827,46	22950,44	31934,12
170	30405,07	41128,02	27062,10	36097,43
180	34394,76	44467,89	30908,65	40914,61
190	37983,69	48420,48	34815,58	45006,23
200	41665,42	51970,06	38474,00	49402,12
210	44811,72	55153,88	42480,74	52938,15
220	48110,31	58640,63	46150,03	56743,59
230	51165,23	62039,11	49883,95	60422,74
240	54321,69	65264,58	53325,87	63943,70
250	57056,31	67792,98	56490,79	67263,18
260	60121,16	70804,04	59841,87	70681,92
270	62819,55	73501,96	62878,43	73633,60
280	65607,40	76378,27	66095,65	77142,72
290	68281,41	79147,46	68993,05	80261,95
300	70490,67	81776,42	71904,95	82867,56
350	82256,47	92569,99	84851,93	96460,19
400	91911,53	102099,71	96840,93	107565,52
450	100028,84	109902,97	106691,78	116832,53
500	107344,28	116175,86	115397,96	125449,98
550	113258,11	122217,43	123122,77	132294,61
600	118564,10	126964,90	129674,94	138272,84
650	123573,08	130777,50	135017,86	144254,35
700	127590,17	134731,56	140665,94	148768,99

Table 7: DDGH for Bilbao

CLIMATIC ZONE "C2" (BARCELONA)

DDGH (K h)

Ψ (m^2 K/kW)	Te: 5 (ºC)		Te: 14 (ºC)	
	α = 0.2	α = 0.5	α = 0.2	α = 0.5
50	-	-	-	-
60	-	1595,08	-	-
70	265,96	6688,45	-	176,15
80	3700,86	12694,08	-	4700,34
90	8830,55	20029,64	2517,60	10977,21
100	14829,13	28538,85	7702,45	18217,12
110	21955,31	36816,11	13972,60	26769,35
120	29426,06	45636,48	20839,08	35362,63
130	37187,92	54116,76	28691,87	44707,08
140	44879,92	62170,38	36545,45	53714,64
150	52862,05	70096,03	45269,82	63022,06
160	60158,30	77160,39	53210,96	71407,39
170	67239,15	84613,24	61221,98	79867,44
180	73676,04	90410,28	69197,07	87269,34
190	80327,21	96919,51	76969,60	94577,30
200	86319,44	102611,98	84345,04	101810,79
210	91550,71	107969,12	90561,79	107932,27
220	96539,97	112597,00	97144,66	113460,27
230	101776,94	116861,98	103049,83	119439,73
240	106616,96	120890,37	108620,19	125057,44
250	111043,21	124631,14	113622,54	130085,53
260	114790,40	128145,43	118334,22	134359,54
270	118611,67	131573,75	123250,27	138862,32
280	122005,97	134739,52	128293,15	142600,97
290	125321,68	137783,57	132375,20	146567,05
300	128024,70	140461,17	136236,58	149598,39
350	141418,53	151224,54	152579,52	164649,37
400	150774,62	159299,47	165390,05	175493,00
450	157901,36	166556,07	174967,33	183367,84
500	164483,58	172024,06	182158,67	191122,41
550	169405,89	177354,61	188866,22	196633,75
600	174068,43	181798,76	194763,33	202066,09
650	178357,28	185652,45	199074,34	207080,57
700	182347,97	189002,61	203684,54	211744,79

Table 8: DDGH for Barcelona

CLIMATIC ZONE "C3 " (GRANADA)

DDGH (K h)

Ψ (m^2 K /kW)	Te: 5 (ºC)		Te: 14 (ºC)	
	α = 0.2	α = 0.5	α = 0.2	α = 0.5
50	-	-	-	-
60	-	-	-	-
70	-	3998,48	-	-
80	1432,05	13179,92	-	1814,26
90	8541,57	22705,54	306,20	10653,56
100	17576,33	32142,61	6294,25	21834,76
110	26265,58	43124,94	15917,74	32235,93
120	34754,73	53512,76	26462,21	42058,87
130	45199,05	62679,33	35405,62	54234,58
140	54806,75	72154,99	44930,83	65476,40
150	63069,86	81298,45	56485,80	74947,38
160	70852,56	90121,17	66744,83	83641,31
170	78878,82	97742,69	75115,63	93147,70
180	86511,18	104730,08	82871,92	101954,60
190	94017,82	110727,58	91316,17	110767,95
200	100826,29	116265,11	98663,74	118664,34
210	106930,17	121528,20	106676,28	125625,22
220	112201,06	125785,85	113546,31	131703,30
230	116850,30	130132,81	121130,74	137035,98
240	121467,02	133854,76	127369,02	142171,23
250	125454,79	137469,22	132766,03	146863,93
260	129226,56	140921,76	137623,27	150990,45
270	132620,54	144136,29	141960,88	154891,84
280	135864,60	147711,08	146309,19	158498,08
290	139077,31	151143,82	149988,17	161883,28
300	141853,98	153884,92	153777,04	165242,02
350	155350,01	165652,43	168426,75	180791,98
400	165413,83	174496,51	181778,34	192228,77
450	173468,22	181768,47	192076,38	201627,55
500	180144,79	187324,44	200235,33	209282,62
550	185651,57	192034,24	207555,36	215538,05
600	189956,04	195766,17	213706,87	220358,99
650	193675,35	199347,88	218565,00	224668,58
700	196903,42	202212,84	222657,61	228306,94

Table 9: DDGH for Granada

Ψ (m^2 K / kW)	CLIMATIC ZONE "C4" (TOLEDO) DDGH (K h) Te: 5 (ºC)		Te: 14 (ºC)	
	α = 0,2	α = 0,5	α = 0,2	α = 0,5
50	-	-	-	-
60	-	234,24	-	-
70	-	5862,86	-	-
80	3176,49	14733,35	-	3979,06
90	10079,05	23285,01	1018,92	12374,41
100	18155,23	34098,18	8196,52	22358,66
110	26581,88	43816,68	17265,10	32190,46
120	36282,68	52953,69	26461,47	43823,47
130	45303,87	62189,23	35297,24	54495,19
140	53786,43	71184,45	45976,82	64253,64
150	61573,55	80305,50	56019,08	73181,53
160	69264,50	88737,65	65124,60	81534,51
170	77313,31	95705,71	73071,07	91205,35
180	85344,13	102073,14	80304,84	100593,22
190	92272,58	107646,84	87884,84	108797,82
200	98384,60	112340,83	96898,38	115745,72
210	103607,48	116895,45	104952,82	121618,24
220	108544,16	121343,34	111864,08	127288,19
230	112624,21	125396,86	117808,71	131923,66
240	116752,20	128976,68	123124,85	136369,98
250	120863,03	132514,57	127571,01	141385,78
260	124515,11	136134,70	132330,69	145467,94
270	127644,34	139582,95	136166,72	148911,82
280	130902,53	142505,08	140636,85	152554,67
290	133424,02	145480,48	144579,45	155325,05
300	136304,50	147956,53	147769,95	158455,88
350	148943,47	159181,68	161673,40	173314,62
400	158604,11	167766,66	174237,97	184483,35
450	166619,43	174499,91	184014,17	193788,31
500	173002,03	179708,77	192784,33	201085,09
550	177832,02	184717,28	199436,33	206512,14
600	182354,25	188888,40	204966,83	211631,77
650	186328,39	192253,60	209515,18	216022,34
700	189875,17	195122,46	213990,94	220124,63

Table 10: DDGH for Toledo

CLIMATIC ZONE "C2" (VITORIA)

DDGH (K h)

Ψ (m^2 K / kW)			Te: 14 (ºC)	
	α = 0.2	α = 0.5	α = 0.2	α = 0.5
50	-	-	-	-
60	-	-	-	-
70	-	1269,02	-	-
80	303,97	4761,74	-	395,03
90	2496,91	9272,57	43,56	3152,56
100	6536,78	14530,85	1769,12	8139,64
110	10843,67	20522,03	5340,00	13313,06
120	16138,15	26715,62	10029,58	19669,72
130	21155,23	33014,52	15035,13	25729,57
140	27076,67	38734,91	20915,97	32454,92
150	32738,70	44311,38	26516,28	39103,57
160	37694,42	49629,63	33016,92	44962,41
170	42409,59	54806,96	38804,01	50287,04
180	47415,07	59742,38	44221,47	56160,30
190	51968,25	64314,68	49015,64	61427,14
200	56422,62	68955,16	53712,86	66627,09
210	60750,17	72686,20	58896,64	71733,54
220	65061,20	76745,97	63783,33	76725,73
230	68835,15	80584,37	68154,61	80878,91
240	72280,79	83612,12	72476,26	85013,74
250	75775,64	86733,47	76563,83	88927,60
260	79203,92	89848,29	80487,52	93033,52
270	82583,51	92920,70	84511,33	96745,59
280	85341,71	95708,24	88127,88	100123,68
290	88159,21	98453,93	91561,93	103132,57
300	90346,25	101470,40	94996,47	105766,31
350	102678,24	113173,12	108882,21	119914,27
400	112858,88	122075,19	121172,74	131738,32
450	120907,94	129818,41	131535,78	140967,25
500	128016,08	136109,67	140294,44	149228,00
550	134029,30	141607,71	147160,12	155974,19
600	139126,62	146245,08	153996,34	161885,94
650	143646,49	150626,78	159302,92	167160,34
700	147754,19	153926,69	163903,71	171646,26

Table 11: DDGH for Vitoria

CLIMATIC ZONE "D2" (ZAMORA)

DDGH (K h)

Ψ (m^2 K / kw)	Te: 5 (ºC)		Te: 14 (ºC)	
	α = 0,2	α = 0,5	α = 0,2	α = 0,5
50	-	-	-	-
60	-	-	-	-
70	-	1475,34	-	-
80	401,36	7751,58	-	522,77
90	4786,85	15618,92	-	6057,53
100	11334,03	24009,00	3116,29	13968,36
110	19211,74	32367,47	10220,74	23620,84
120	26502,33	41754,17	18594,53	32219,00
130	34242,27	49492,42	27233,11	41407,21
140	43203,42	57725,09	34719,91	51771,95
150	49997,56	65137,04	43334,63	59453,45
160	57035,90	71629,20	52905,01	67589,78
170	64024,21	78152,93	60042,71	75690,51
180	69562,78	84850,22	67293,63	82079,66
190	75583,12	90602,36	74306,58	88898,53
200	81134,12	95590,02	80567,52	95211,71
210	87013,12	100374,25	86205,47	102243,12
220	92060,81	105001,62	92286,15	108061,91
230	96456,27	108817,99	97998,50	113051,56
240	100490,26	112240,65	104383,04	117701,79
250	104738,82	115438,29	108979,42	122494,02
260	108378,87	118991,16	113621,33	126816,59
270	111610,19	122179,65	117680,38	130562,28
280	114270,61	125103,02	122046,12	133495,48
290	117388,83	128064,22	125962,49	136931,68
300	120360,52	130669,81	129457,39	140288,82
350	132331,73	143710,24	143898,06	153981,23
400	144011,40	152913,67	155557,76	167594,72
450	152060,30	160511,90	167608,29	177026,17
500	159382,65	166599,24	176034,63	185371,14
550	165162,47	172239,76	183659,10	192070,34
600	169787,76	176214,14	190070,62	197236,67
650	174283,46	179890,05	195324,65	202297,65
700	177618,69	182797,19	200099,32	205896,94

Table 12: DDGH for Zamora

CLIMATIC ZONE "D3" (MADRID)

DDGH (K h)

Ψ (m^2 K / kW)	Te: 5 (ºC)		Te: 14 (ºC)	
	α = 0,2	α = 0,5	α = 0,2	α = 0,5
50	-	-	-	-
60	-	-	-	-
70	-	2084,83	-	-
80	535,74	8051,07	-	654,83
90	4797,98	15518,52	-	6048,62
100	11726,58	23021,91	3850,55	14504,94
110	17992,93	31933,46	9780,40	22112,79
120	25423,11	39998,71	18172,26	31047,16
130	33579,37	47990,18	25296,17	40643,11
140	41049,21	55055,89	33329,66	49328,86
150	48028,52	62963,26	42248,08	57244,27
160	54728,42	70234,13	50144,27	64890,40
170	60875,77	76929,68	57537,74	71888,56
180	67278,49	82123,37	64402,95	79440,33
190	73739,20	87560,98	70380,31	86843,58
200	79626,04	92595,59	76857,42	93677,61
210	84192,59	97148,26	83389,53	98960,73
220	88817,42	101181,76	90173,06	104349,98
230	93127,87	104790,48	95996,46	109070,10
240	97347,66	108390,09	100808,54	114136,14
250	101057,11	111947,18	105428,65	118399,67
260	104308,54	115328,36	109640,43	122015,17
270	107369,59	118655,15	114079,47	125324,42
280	110437,27	121882,29	117781,73	128981,78
290	113544,97	125180,27	121437,12	132546,28
300	116291,30	128025,98	124546,60	135701,00
350	129562,92	140319,13	139146,66	150876,13
400	140386,50	149727,05	152249,34	163599,02
450	148779,24	157029,15	163495,48	173240,24
500	155752,46	163249,19	172009,20	181211,42
550	161334,16	168530,59	179576,65	187554,12
600	166335,92	172777,87	185565,71	193184,23
650	170789,48	176862,84	190892,72	198317,87
700	174160,44	180523,21	195738,67	202057,41

Table 13: DDGH for Madrid

CLIMATIC ZONE "E1" (BURGOS)
DDGH (K h)

Ψ	Te: 5 (ºC)		Te: 14 (ºC)	
(m^2 K/ kW)	α = 0.2	α = 0.5	α = 0.2	α = 0.5
50	-	-	-	-
60	-	-	-	-
70	-	1882,17	-	-
80	536,24	6772,19	-	570,00
90	4340,72	13190,19	-	5512,52
100	9487,78	19663,43	3128,23	11530,87
110	15774,02	26624,05	8436,72	19365,36
120	21550,50	33794,60	15173,97	26228,22
130	28137,76	40502,00	21687,05	33961,17
140	34672,70	46377,97	28062,98	41532,33
150	40713,44	52293,76	35377,77	48652,35
160	46076,99	58042,56	42195,73	54581,40
170	51028,90	64264,71	48997,54	60267,53
180	55810,45	69385,54	54420,63	65779,17
190	60980,79	74531,24	59357,89	71695,64
200	66290,28	78795,09	64164,97	78016,82
210	71091,54	83106,19	69209,45	83604,20
220	75375,32	86580,12	74648,56	88649,59
230	79335,39	89844,54	80106,69	93180,18
240	82962,53	93319,88	85194,78	97231,20
250	86387,37	96436,16	89628,84	101376,63
260	89341,03	99179,89	93318,34	104671,94
270	92062,03	101784,49	97073,37	107726,13
280	95080,81	104556,22	100785,19	111189,92
290	97875,36	107611,04	104019,50	114336,78
300	100098,72	110011,44	106815,17	116759,16
350	111745,62	121681,75	120117,12	130043,77
400	121861,30	131473,49	131521,28	141990,18
450	130385,81	139010,06	141886,67	151859,68
500	137585,33	145467,84	151213,88	160338,96
550	143569,35	151144,81	158837,33	167131,56
600	149114,67	156090,09	165223,99	173464,85
650	153506,33	160123,80	171139,21	178421,15
700	157669,82	163656,80	176261,55	183307,75

Table 14: DDGH for Burgos

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