AIR-CONDITIONING REFRIGERATION PLANTS WITH ICE STORAGE IN HOTELS IN SPAIN AND PORTUGAL. DESIGN AND ECONOMIC ANALYSIS.

Celestino R. Ruivo[1], José M. Pinazo [2] and Angel Barragán [2]

[1] Área Departamental de Engenharia Mecânica, Escola Superior de Tecnologia da Universidade do Algarve - Campus da Penha, 8000 Faro, Portugal
[2] Departamento de Termodinamica Aplicada, Universidad Politécnica de Valencia , España

ABSTRACT

The cooling needs in hotels, like in a lot of buildings, are usually high during the day when the price of electricity is high.
The integration of a thermal storage system in a HVAC refrigeration plant can be advantageous, because the storage system can be charged at night, with the chiller running at maximum capacity with low electric energy costs. The other main advantage is due to the possible reduction of electric power that one needs in the refrigeration plant.
This change in the electricity consumption, the reduction of the consumption during times expensive electricity price and the increase in usage during low cost periods of electricity, is also advantageous for to the electricity production and distribution companies because the consumption is more stable.
When designing a refrigeration plant with an ice bank for air conditioning purposes, for example in a hotel, it is first necessary to estimate the cooling needs considering an adequate scenario for design purposes and a realistic scenario for energy consumption. The histogram showing cooling needs on an hourly basis over several days, for example a typical day each month, must be considered in order to size the refrigeration plant, chiller and the ice bank.
This work presents some results of a case study of two similar hotels situated in South Spain and South Portugal respectively. The cooling loads in several zones of the building are calculated with the program "DPClima". Then, using the program SIPfrio, first the refrigeration loads of the central plant are estimated, and then the preliminary sizing of the refrigeration plant imposing an adequate storage strategy is estimated and next the preliminary sizing by dynamic simulation processes is confirmed or rectified. At the end, the economic analysis is done considering a realistic scenario for energy cooling needs.

KEYWORDS

Keywords: Refrigeration Plants, Ice Banks, Storage Strategies, Dynamic Simulation, Economic Analysis

1-INTRODUCTION

Nowadays, the main reason to consider the use of ice banks in the central refrigeration plants of buildings, such us in hotels, is to obtain a significant reduction in the electricity bill. This can be achieved by shifting the electric energy consumption to low-cost periods and also by reducing the electric power, installed and registered. In this paper the economic impact of the use of ice banks systems in the central refrigeration hotels situated in the South of Portugal and South of Spain is shown, and is compared to conventional refrigeration plants. The initial investment costs associated with the acquisition of the chillers and ice storage systems are considered and also the actual electricity energy and power prices in Portugal and Spain.
This case study was done with the following main steps:

Definition of building characterization and utilization of the hotel

Calculation of the hotel cooling loads to size the central the refrigeration plant (Program "Dpclima")

Definition of the HVAC system

Calculation of the central plant refrigeration loads (Program "Sipfrio_Pot_IPF")

Preliminary sizing of the chiller and the ice bank (Program "Sipfrio_Pre_Dim")

Confirmation/Rectification of the sizing by dynamic simulation (Program "Sipfrio_Sim_Din")

Celestino Ruivo, cruivo@ualg.pt

Calculation of the hotel cooling loads imposing a scenario less severe and more adequate to evaluate the energy consumption in the central plant in the several months (Program Dpclima)

Dynamic simulation of the central refrigeration plant considering the scenario imposed in the last step. (Programa "Sipfrio_Sim_Din")

Economical analysis of the investment

2-CASE STUDY

The present case study concerns two hotels (A) and (B), situated respectively in South Portugal and South Spain, respectively. They have the same cooling needs because it was assumed that the use, typology and weather conditions are the same.

Both building hotels have 84 bedrooms, 1 restaurant (500 m^2), 1 cafeteria (100 m^2), 1 lobby (100 m^2), 6 offices (15 m^2), 1 meeting room (40 m^2), 1 ball rooms (200 m^2). These spaces are distributed through out the 6 floors of the hotel.

The air conditioning plant is composed basically for a central refrigeration plant producing chilled water that is distributed to the terminal units in the several zones. Each terminal unit has a coil that cools the air supply (exterior and return air mixture) to only control the indoor temperature of the zone. This controlling is done via the alteration of the surface temperature of the coil, changing the inlet temperature or the fluid flow rate. In those situations, we can have or not air dehumidification caused by the condensation over the surface of the coil and the relative humidity will change instantaneously. The relative humidity values can be higher or lower than 50%, usual value considered in the cooling thermal load calculation.

Figure 1 shows the layout of the air conditioning plant. In the primary circuit, the central refrigeration production, the storage unit and the chiller are connected in a parallel layout. It is possible to leave out the heat exchanger that is represented between the central production and the distribution. This HVAC installation does not have heat recovery exchanger between the exhaust air and the renovation air.

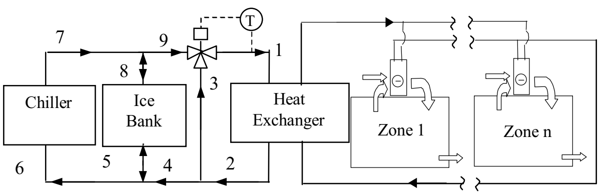

Fig. 1- General schema of HVAC plant

The Heat Transfer Fluid (HTF) used is an ethylene glycol and water solution and the chiller is usually of the compression type, powered by an electric engine. A three-ways valve controls the outlet HTF temperature for the distribution.

The HTF flow through the ice bank is variable and changes circulation direction when a transition from charge mode to discharge mode occurs or vice-versa. The HTF flow is constant through the heat exchanger.

The chiller usually has several working stages and the HTF flow through the evaporator assumes a different constant value, when even happens a change in working according the alteration.

The ice bank is a special heat exchanger like a reservoir filled with plastic capsules. The content of the plastic capsules is the Phase Change Substance PCS. Another possibility is a tank filled with tubes that are surrounded by PCS that accumulates the energy as long as the HTF circulates inside of the tubes. Those two configurations are shown in figure 2.

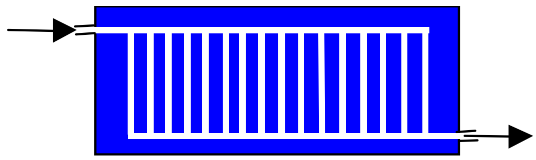

Figure 2. Two configurations of latent storage unit

Time dependence characterisation of the heat and mass transfer processes that occur in the different parts of the installation shown in figure 1 is done by numerical simulation imposing an adequate heat transfer coefficient between the HTF and the PCS, and knowing some properties of both medium.

2.1-THERMAL LOADS CALCULATION

The hotel was divided in several zones to simplify the calculation of the cooling loads. This was done using the program DPClima [6]. It was necessary to select several types of environments (walls, roofs, windows, and shadow devices), to characterize internal gains and also ventilation needs in each zone according to the occupancy or type of zone. Indoor temperature and relative humidity were fixed respectively at 25º and 50% respectively. The outdoor conditions, (incident radiation throughout building, air temperature and relative humidity) were generated in an hourly basis automatically by the program DPClima. In this case we are talking about a town in South Spain near Portugal.

The occupation loads were calculated by the program DPClima considering the occupation schedule in each zone. The associated ventilation rates in each zone were also calculated according to the occupation schedule and type of space (Restaurant, Bedrooms, etc).

The first cooling load calculation was done considering high occupation, high ventilation rates and severe weather conditions for each month. The results obtained with these first conditions were useful in the design step of the HVAC plant, in particular for sizing the chiller and the energy storage system.

Figures 3 and 4 illustrate some graphical results obtained by running the program DPClima. Figure 3 shows several components of cooling loads for a particular zone, e.g. attic bedrooms, and for the design day on July. Figure 4 shows the global results of the cooling loads calculations for the hotel (Building and ventilation load) for the design day on each month (from April to October).

2.2-DETERMINATION OF THE HOTEL COOLING NEEDS

Most of the time, the instantaneous cooling needs in the hotel are not equal to the cooling loads calculated by the program DPClima. This is because the HVAC system, represented above in figure 1, does not control the relative humidity; it only controls the indoor temperature. However, using a simplified algorithm, it is possible to determinate the real cooling needs of each zone based on the cooling loads calculated by DPClima.
In reality, the instantaneous cooling needs of each zone can be related to the latent and sensible components of the cooling loads calculated by DPClima for each zone (space and ventilation loads). The algorithm used to do this correction procedure also needs the working parameters of the HVAC installation, e.g. the contact factor and surface temperature of each cooling coil.
In this case study, it was supposed that the supply airflow is constant, and the supply temperature when working at maximum capacity is 14ºC.
At times with only partial loads, the supply temperature is obviously higher than 14ºC and depending on the surface temperature of the cooling coil, in comparison to the dew point of the inlet airflow, the coil can be totally wet, partially wet or completely dry. If the dew point of the inlet airflow is lower than the surface temperature of the coil, the cooling coil load is exactly equal to the sensible component of the cooling load zone (Space and ventilation load) calculated by DPClima. In other situations, when the surface is partially or completely wet, the cooling coil load can be lower or higher than the cooling load zone calculated by DPClima. If it is higher, the air relative humidity will be lower than 50%, and if it is lower, the relative humidity is obviously higher than 50%. The important issue is to guaranty that the humidity does not assume critically low and high values.

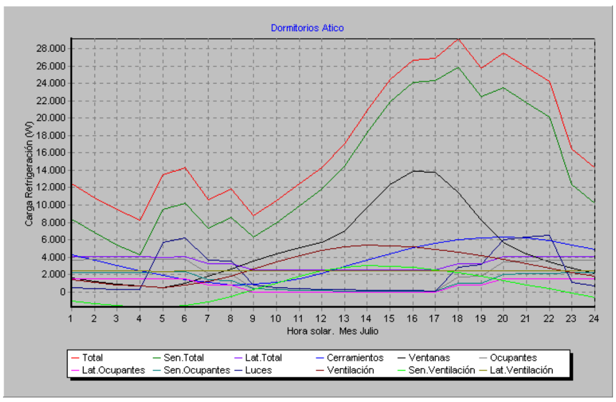

Fig 3- Attic bedrooms cooling loads on July (Program DPClima)

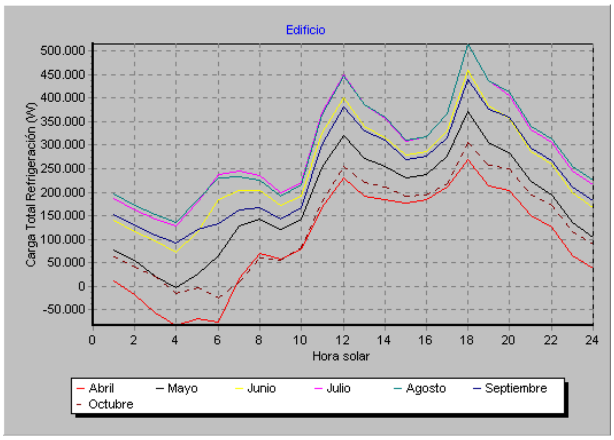

Fig 4- Hotel cooling loads on July (Program DPClima)

The algorithm developed was implemented into the sub-program “Sipfrio_Pot_IPF”. Basically the algorithm works with manipulation of mathematical relations used in psycrometry, and simple energy and mass balances to characterized cooling process in the each zone and associated terminal air conditioning.
The ventilation rates in the environments, e.g. bedrooms and management offices, were considered constant during all 24 hours, but in the restaurants and ball rooms two ventilation rates were considered: a maximum rate when occupied and a minimum rate when empty.
The cooling coil in each terminal unit, the fan-coil for example in the bedrooms, has a constant contact factor value of 80%. The temperature of the coil´s surface is controlled in order to assure that the interior zone temperature is 25ºC.
The results presented in figures 5 and 6 illustrate, for the design day on July, are the values of cooling coil load versus the cooling load zone. Figure 5 shows the attic bedroom zone and figure 6 shows the global results of the hotel. For the other months and other zones similar results were obtained.

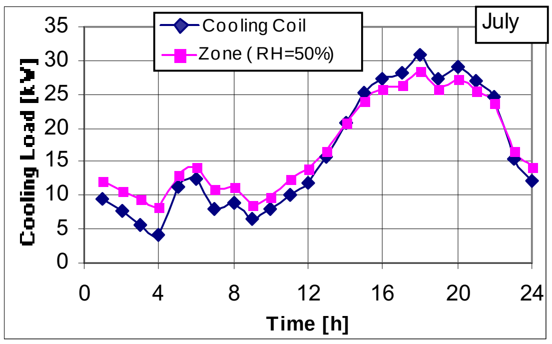

Fig 5- Attic bedrooms cooling loads for the design day on July- Cooling coil and zone (RH=50%)

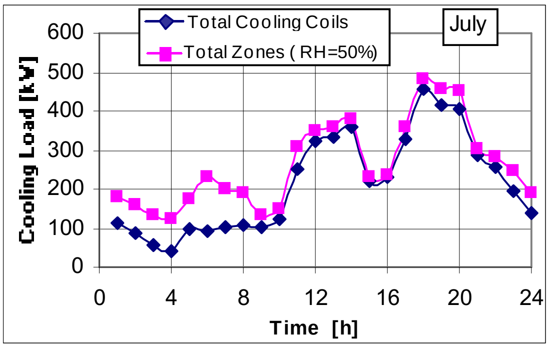

Fig 6- Hotel cooling loads for the design day on July- Total cooling coil and total zones (RH=50%)

Figure 7 represents the total cooling coil loads for the design day on each month. These cooling loads are useful to size with more accuracy a central refrigeration plant with a storage energy system.
However, if we are interested in evaluating the energy consumption in the HVAC plant, full load scenario, in which we have extreme design conditions, is not adequate because the energy consumption will be very high and not realistic.
So, to know the energy consumption of a HVAC plant, we must consider a second scenario having a medium day on each month characterized by an occupancy lower than the maximum limit and medium weather data (temperature and radiation values). Figure 8 represents the final cooling coil loads for the medium day on each month. The calculation procedure followed was equal to the procedure used in the first scenario.

2.3-PRELIMINARY SIZING OF THE CHILLER AND ICE BANK

In this case study was selected a chiller capacity lower than the maximum total cooling coil load to get some reduction in electrical power costs in comparison to a conventional plant. To simplify the actual analysis it was assumed that the coefficient of performance of the chiller is a constant medium value. This assumption has in fact a certain realism if the chiller has an exterior air condenser because during the charge mode at night the exterior air is at lower temperatures in spite of the fact that the HTF fluid trough the evaporator is at negative temperatures. However, the influence of the HTF fluid temperature in the refrigeration load and electrical input power of the chiller was considered. During the ice bank charge mode the maximum refrigeration load of the chiller is reduced to about 2/3 of the value that was reach during the discharging mode. In both modes the chiller has four working stages.
The ice bank capacity will depend on the operational strategy selected, like total storage, partial storage, demand-limited storage or a mode matching some of these three.

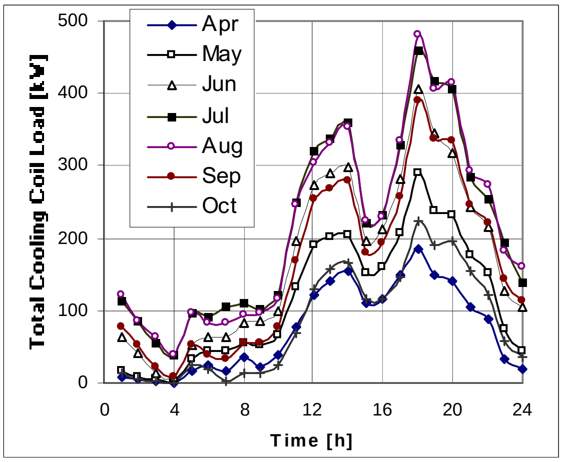

Fig 7- Total cooling coil loads for the design day on each month

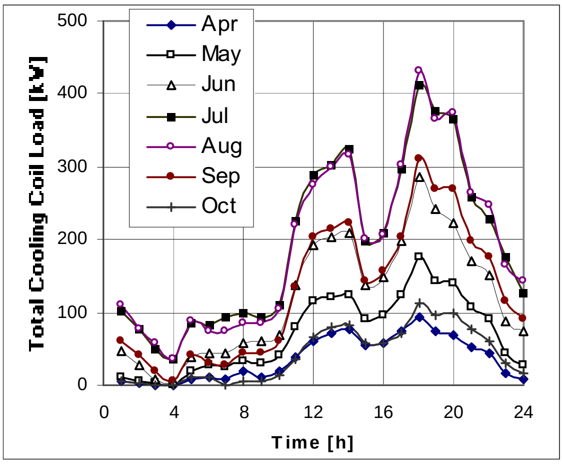

Fig 8- Total cooling coil loads for the medium day on each month

With the histograms of cooling loads and the chiller refrigeration load, during the charge and discharge modes of the ice bank, and also considering the actual triple electrical tariff schedule, in South Spain and Portugal, the best strategy on each month was chosen according to the following priorities:

- 1st Priority: Chiller is working during off peak energy electrical price hours
- 2nd Priority: Chiller is working during off peak and mid energy electrical price hours
- 3th Priority: Chiller is working off peak, mid and on peak energy electrical price hours

The main objective of this plan is to shift the electrical energy consumption in the central refrigeration to periods that have the cheaper price per kWh. Then the histogram loads in the ice bank, for the design day on each month, were obtained to calculate the quantities of energy stored and discharged on each cycle day. Using the following expression, it is possible to estimate the ice bank volume on each month:

$$V = Q_{stored} \Big/ \Big[\rho_s \cdot (1-\varepsilon_f) \cdot \Big(c_{p_{s,liq}}\left(T_{max} - T_{phase}\right) + q_L + c_{p_{s,sol}}\left(T_{phase} - T_{min}\right)\Big) + \rho_f \cdot \varepsilon_f \cdot c_{p_f}\left(T_{max} - T_{min}\right)\Big] \quad (1)$$

This calculation procedure permits us to obtain the volume of the storage system. The volume that must be considered to verify the energy criterium is the maximum of the values calculated, to be sure that we have capacity on each day to store and discharge the desired quantities of energy.
However, the ice bank needs to have, at each instant in charge or discharged modes, capacity to change the energy transfer rate, charge or discharge load, between the HTF fluid and the PCS substance. This is a load criterium that must also be satisfied.
Considering the ice bank like a classical heat exchanger having a finite heat transfer area and knowing the global heat transfer coefficients and the mean logarithm temperature difference for both operation modes, we can use the following equation to obtain the volume of the ice bank according this second criterium:

$$V_i = \frac{\dot{Q}_i}{U'_i \cdot \Delta T_{\ln i}} \quad , \qquad i = disch, ch \qquad (2)$$

The final step that must be considered results from the verification of these two criteria.
In the present case study the following main data are used:

-Maximum total cooling coil load: 480 kW.
-Maximum chiller capacity: 300 kW at positive temperature and 200 kW at negative temperatures.
-Number of chiller working stages: 4 stages (positive temperatures: 75, 150, 225 and 300 kW, negative temperatures: 50, 100, 150 and 200 kW).
-Coefficient of performance of the chiller: COP=2.6.
-Heat transfer coefficients between the HTF and PCS: $U'_{ch} = U'_{disch}$ =2.0 kW/m^3°C).

With this formulation it is possible to achieve the storage strategy that minimizes the volume of the ice bank and also choose the best strategy for volumes that are larger than the critical volume. In order to do this preliminary sizing of the refrigeration plant containing an ice bank, a simple algorithm that is used by the sub-program "Sipfrio_Pre_Dim" was developed.

Using this simple program and according to the first scenario of cooling loads a critical volume of 13.1 m^3 was calculated. For other volumes (14, 16, 18 e 20 m^3) the corresponding best storage strategies were achieved. Figures 9 and 10 show the histogram loads (chiller, ice bank and distribution), concerning the particular case of the 16 m^3 volume ice bank, respectively, for the design days on April and on July. These figures show how the working strategy can be different from April to July.
On April, the cooling needs are not very high. However, we verify that the chiller needs to work during the day. In this case there are two reasons that justify this strategy. The first is the fact that at night the ice bank has no capacity to absorb the maximum charge load from the chiller (0 h to 5 h). The second is the fact that the ice bank does not have enough capacity to store all the energy that is distributed during the day (periods of high electrical prices).

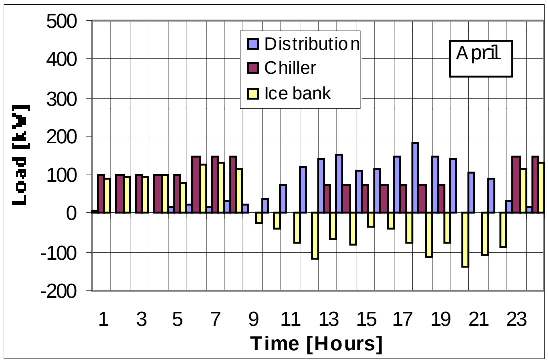

Fig. 9- Load histograms for the design day on April. Plant with 16 m^3 ice bank volume.

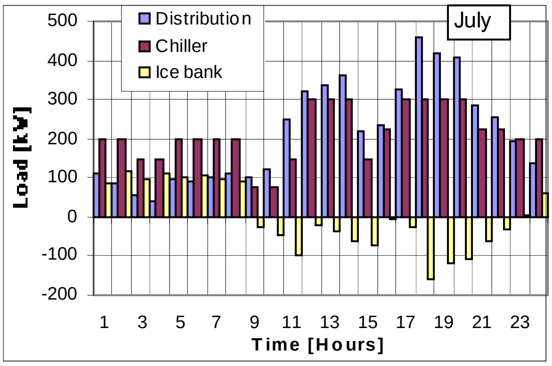

Fig. 10- Load histograms for the design day on July. Plant with 16 m^3 ice bank volume

On the design day on July, the cooling needs are greater. We can see that the chiller works most of the time during the charge period at maximum load (4 stages), only between 3 hours and 4 hours it works at partial load (3 stages) to satisfy the instantaneous criteria load. During the day, depending on the distribution load, the chiller always works with the maximum number of stages possible.
Another fact in this particular case is that the ice bank stores more energy on April than July.

2.4- DYNAMIC SIMULATION OF THE CENTRAL REFRIGERATION PLANT

The dynamic simulation of a particular central refrigeration plant with a storage system, e.g. an ice bank, supplies extremely important information to define the better operational strategy that must be followed, to refine the sizing of the chiller, storage system and the control system. This problem was formulated with some simplifications, although without loss of accuracy, considering a central refrigeration

plant having one storage unit and one chiller in a parallel layout as shown in figure 1.
Basically, the changes observed in the HTF temperature are associated with the particular processes that occur in the different subsystems of the plant. At the chiller, the HTF flow is cooled and at the air conditioning side, the HTF distribution flow is obviously heated. A mixing of the two flows in the three-ways valve controls the HTF temperature to the distribution. The analysis of these last three processes can be done in a simplified mode. The analysis of the heat storage unit is more complex because the phenomena associated with it. To simplify this task, numerical model to simulate the heat transfer phenomena occurring inside the storage unit was developed. The numerical spatial domain adopted is represented in figure 11.
This numerical domain has an equivalent geometry to the real configurations, like the examples represented in figure 2, since the specific transfer area and the volume fraction of each medium have the same values. Figure 12 shows the geometry of the numerical domain in more detail.
The characterisation of the time dependence of the process that occurs inside the bank is done by means of a heat transfer coefficient between the HTF and the PCS, and also by means of some properties of both mediums.
The thermal interaction between the HTF and the PCS is characterized by the following equation:

$$\rho_f \cdot v_f \cdot c_{P_f} \cdot \frac{\partial T_f(t,y)}{\partial y} = U \cdot \left(T_f(t,y) - T_s(t,y,0)\right) \quad (3)$$

The thermal behavior inside the PCS domain is expressed in a simplified way by the following equations:

$$\frac{\partial}{\partial z}\left(k_{s\_i} \frac{\partial T_{s\_i}(t,y,z)}{\partial z}\right) = \rho_s \cdot c_{p_s} \cdot \frac{\partial T_{s\_i}(t,y,z)}{\partial t}, \quad i = \text{sol}, \text{liq} \quad (4)$$

$$k_{s\_sol} \frac{\partial T_s(t,y,z)}{\partial z} - k_{s\_liq} \frac{\partial T_s(t,y,z)}{\partial z} = \rho_s \cdot q_L \frac{\partial s(t,y,z)}{\partial t}, \quad \text{at} \quad z = s(t,y) \quad (5)$$

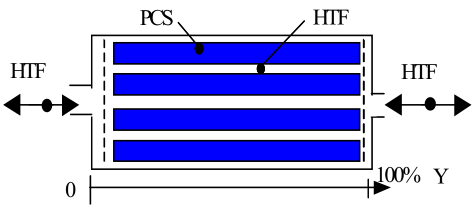

Fig. 11. Numerical domain of the storage unit

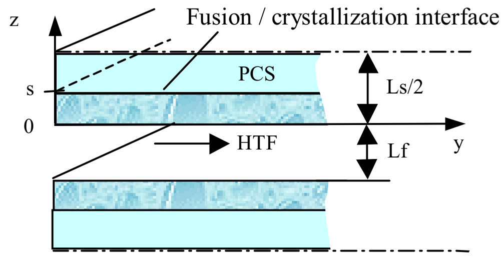

Fig 12- Detail of the ice bank numerical domain

An algorithm to make a time advance calculation process was developed, which allow ones to find out the time evolution of the HTF and PCS temperatures as well as the mass fractions, liquid and solid, of PCS. This algorithm is the basis of the sub-program “Sipfrio_Sim_Din”, which can be used to dynamically simulate the storage unity, but also to simulate the other sub-processes in the central refrigeration plant, e.g. the HTF cooling in the chiller and the mixing in the three-way valve.
The confirmation or rectification of the preliminary sizing is basically done by specifying the volume of the ice bank, the histogram loads on each design day, and some temperature values. In some cases, the strategy adopted in the preliminary sizing step can not be implemented if we detect that the second criteria, by means of the dynamic simulation, is not satisfied, for example by a sequence of several hours at maximum charge load or at maximum discharge load. When this happens in the charge mode, a suddenly reduction in the temperatures of the HTF crossing the ice bank and the chiller occurs that can strongly penalize the chillers performance or can in more extreme temperatures disable the chiller. In the opposite mode, a suddenly rising in the HTF temperature at the outlet of the ice bank can occur that compromises the temperature control done by the tree-ways valve.
Figures 13, 14, 15, 16, 17 and 18 illustrate some results obtained by dynamic simulation using the sub-program “Sipfrio_Sim_Din” concerning the particular design days and an ice bank measuring 16 m^3. The figure 13 represents the evolution of stored energy in the ice bank. The other graphical results are the evolutions of HTF mass flow rates and HTF temperatures at specific points in the central refrigeration plant (inlet and outlet of the chiller, inlet and outlet of the ice bank, inlets and outlet of the tree-way valve).
This simulation process was extended to all design and medium days to confirm or rectify the sizing storage strategy in both scenarios (sizing scenario and energy consumption scenario).

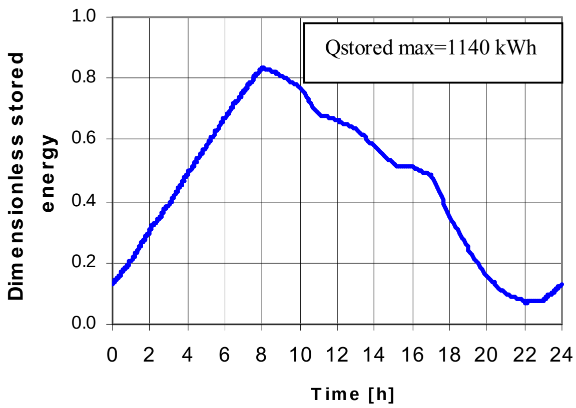

Fig 13- Dimensionless stored energy in the bank

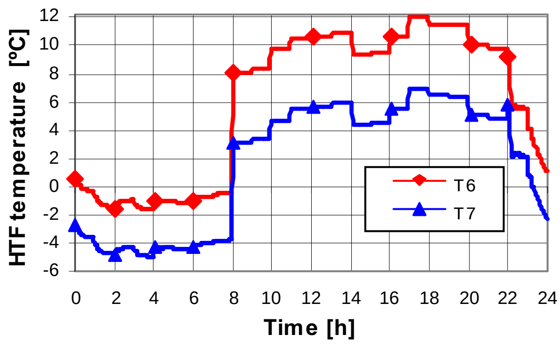

Fig 14- HTF temperatures (Points 6 and 7)

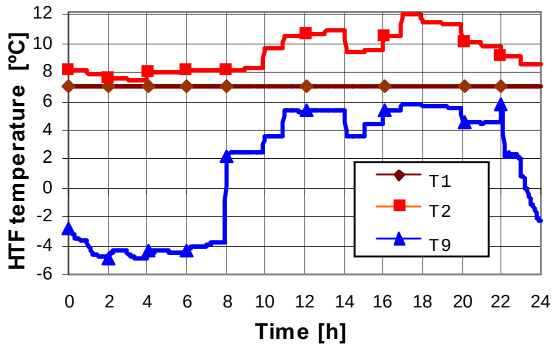

Fig 15- HTF temperatures (Points 1, 2 and 9)

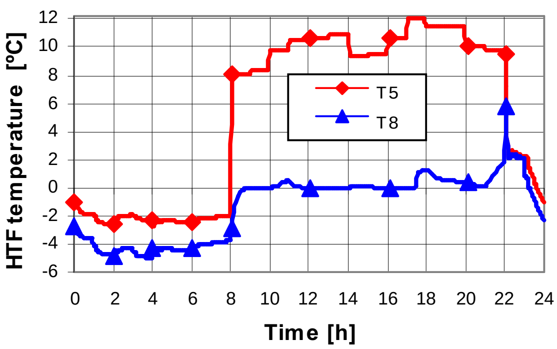

Fig 16- HTF temperatures (Points 5 and 8)

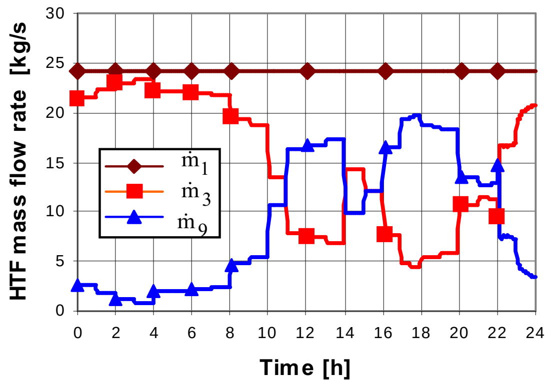

Fig 17- HTF mass flow rate (Points 1, 3 and 9)

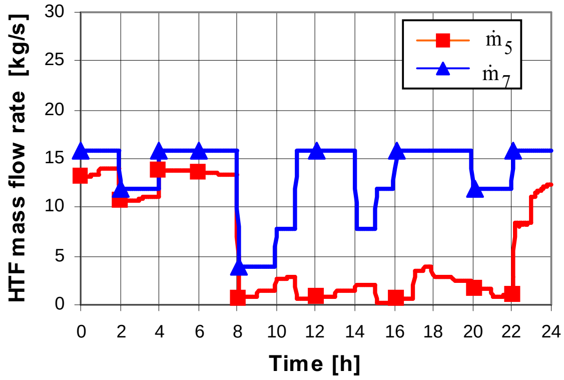

Fig 18- HTF mass flow rate (Points 5 and 7)

2.5- ECONOMIC ANALYSIS

The economic analysis consists basically in comparing two refrigeration plants solutions, one conventional without an ice bank, and the other with an ice storage system. The only exploitation costs considered were electrical energy and power costs. On the other hand, the initial costs were the acquisition costs of the ice storage system and the chillers. The sub-program "Sipfrio_Ana_Econ" does this task considering the actual electricity prices in Portugal and Spain. The costs of acquisition of the chillers (C Cost) and of the ice bank (IB Cost) were estimated by the following correlations, obtained from a particular commercial company:

$$\text{C Cost} = 95.6 \cdot \text{Max.Ref Load} + 4879 \qquad (6)$$

$$\text{IB Cost} = 1319 \cdot \text{IB Volume} + 7882 \qquad (7)$$

Figure 19 shows the difference between the operational costs of the central refrigeration plant, both with and without an ice storage system in both countries. Figure 20 shows the Payback values associated with investing in a refrigeration plant having an ice bank. Based only on the results of the present case study, we can say that the investment in Portugal is more attractive than the one in Spain, and the Payback values found are reasonable. The best economical solutions are for small volumes of the storage system. However, in those cases it is more difficult to control the discharge and charge of the ice bank.

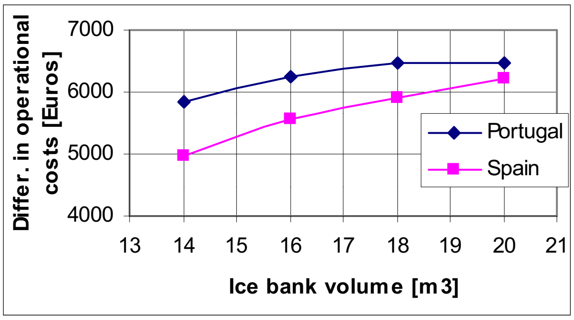

Fig 19- Difference between the operational costs of the plant with and without ice bank

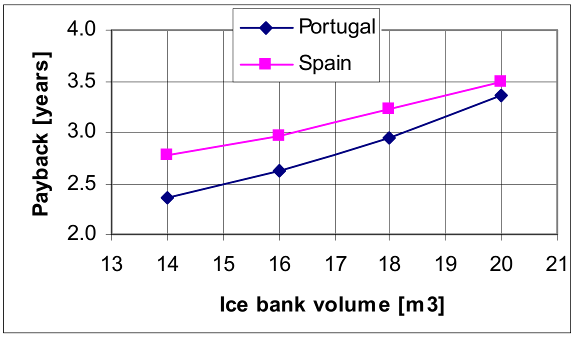

Fig 20- Payback of investment in a plant with an ice bank

3-CONCLUSIONS

A case study of an air-conditioning installation in two similar hotels, one situated in South Spain and the other in South Portugal was solved. The central refrigeration plant has an ice bank as an energy storage system.
The thermal cooling loads were supposed equal in both hotels and were calculated using the program "DPClima". The resulting cooling load values were used to size the refrigeration plant and also to get a premonition of the energy consumption in the central refrigeration plant.
Particular chiller working conditions was selected to obtain adequate relations of the preliminary sizing of the bank, the best storage strategy for each month and for several values of the ice bank. The simulation using the program "Sipfrio" was done to confirm or rectify the strategy imposed on each typical day.
Finally, an economic analysis of the investment in a central refrigeration plant with an ice bank was done considering the actual electrical triple tariff for both hotels.
The liberalization of the electrical sector that is occurring can significantly change the economical analysis. So, it is very important for a larger electrical consumer, like a grand hotel, to take some caution in this kind of investment. However, before the decision of the installation of a storage system, it is important to negotiate the electricity prices of energy and power with the electrical company, considering the two possible options, a plant with or without a storage system.
The application of this technology to air conditioning systems, particularly in hotels, integrating for example the heat recovery condenser for pre-heating sanitary water or to heat an interior swimming pool can give a good contribution to the rational use of the energy.

4-REFERENCES

1-ASHRAE Handbook (1989), Fundamentals Volume, American Society of Heating, Refrigerating and Air-conditioning, Inc., Atlanta, GA,

1-Patry, J. Stockage par Chaleur Latente. Pyc Édition. 1981

2-Ruivo, Celestino. Comportamento Térmico de uma Unidade Permutadora-Acumuladora de Calor Sensível. Tese de Mestrado. Coimbra. FCTUC. 1993.

3-Ramos, João; Carvalho, Mª Graça. Acumulação de Calor-Estratégias de Gestão; Revista Grau Celsius, pag 22. APIRAC. 1992

4-CRISTOPIA Energy Systems, Experiments Library "CRISLIB software". 1996.

5-Celestino Rodrigues Ruivo Modelação Numérica dos Sistemas de Produção de Frio com Acumulação de Energia em Bancos de Gelo, Congresso Ibérico de Ar Condicionado e Refrigeração, Lisboa 1999.

6- DPClima. Versión 1.3, Junio 2000. Universidad Politécnica de Valencia, Camino Vera s/n 46022 Valencia. España

7-Pinazo Ojer, José Manuel. Manual de Climatización. Tomo II: Cargas Térmicas. Servicio Publicaciones UPV. 1996.

NOMENCLATURE

$c_{p_{s,liq}}$	Constant-pressure specific heat of the liquid phase change substance, J/kg·ºC
$c_{p_{s,sol}}$	Constant-pressure specific heat of the solid phase change substance, J/kg·ºC
c_{p_f}	Constant-pressure specific heat of the heat transfer fluid, J/kg·ºC
C Cost	Chiller cost, Euros
IB Cost	Ice bank cost, Euros
IB	Volume Ice bank Volume, m^3
Max Ref Load	Maximum chiller refrig. load, kW
q_L	Fusion latent heat of the phase change substance, J/kg
Q_{stored}	Stored/discharged energy in the ice bank on each design day, J
$\dot{Q}_{ch}$, $\dot{Q}_{disch}$	-Maximum loads in the ice bank respectively in charge and discharge modes; W
T_{max}	Maximum temperature of the system at the end of the discharge mode, ºC

T_{min} Minimum temperature of the system at the end of the charge mode, ºC

U'_{ch}, U'_{disch} Heat transf. coef. between the HTF and PCS respectively in charge and discharge modes; W/m^{3}ºC

$\Delta T_{ln\,ch}$, $\Delta T_{ln\,disch}$ - Mean logarithm temperature difference respectively in charge and discharge modes; ºC

ε_f Volume fraction of the heat transfer fluid volume

ρ_s Density of the phase change substance PCS (water), kg/m^3

ρ_f Density of the heat transfer fluid HTF (ethylene glycol and water solution), kg/m^3

RÉSUMÉ

Les besoins de refroidissement des hôtels, comme en beaucoup d'autres cas, sont élevées quand le prix de l'électricité est aussi plus élevé.
L'intégration des systèmes de stockage de froid dans les installations de climatisation permets de profiter de quelques avantages associées à la exploitation des différentes tarifications de l'électricité et d'une puissance électrique installée plus basse. On peut profiter aussi des régimes de fonctionnement des groupes frigorifiques stables proches de la pleine charge, sans les démarrages e des arrêtes, permettant une meilleure efficacité énergétique.
Le déplacement de la consommation de électricité obtenue intéresse aussi aux entreprises de production et de distribution de l´énergie électrique.
Quand on fait le dimensionnement d'une installation de climatisation avec des banc à glace, par exemple dans un hôtel, il faut faire deux estimations de la charge thermique de refroidissement, une est valable pour dimensionner les équipements et l'autre pour faire les analyses de consommation d'énergie et économique. Il est suffisant de considérer des histogrammes avant une base horaire.
Dans ce travail on présent des résultats d'une étude considérant deux hôtels identiques, un au Portugal et l'autre en Espagne. Les charges thermiques de refroidissement aux différentes zones sont estimées par le logiciel "DPClima". On calcule ensuite les besoins frigorifiques réelles de l'installation de climatisation avec le logiciel SIPFRIO pour prè-dimensionner l'installation imposant une stratégie adéquat d'accumulation. La simulation dynamique permet alors de confirmer et d' optimiser le dimensionnement et la stratégie d'accumulation. L'analyse économique est enfin présentée..