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ICE STORAGE SYSTEM (ISS): SIMULATION OF A TYPICAL HVAC PRIMARY PLANT EQUIPPED WITH AN ICE STORAGE UNIT.

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ABSTRACT

Within the framework of energy management in a tertiary building, it is necessary to evaluate the possibilities of cold energy storage. This study includes the consideration of ice storage for the air-conditioning of the buildings.

To compare objectively the operating strategies, the manager needs to reproduce the ISS performances for the same conditions (building, cooling and electrical demands, climate conditions,...). He needs also informations to chose the appropriate plant configuration or operating strategy to meet the global cooling demand taking into account the characteristics of the tank, of the demand curve,... at each moment. A simulation model of an entire HVAC primary plant should provide this information.

This paper focuses on the adaptation of a latent storage model to an example of ISS. In order to validate the model the results of the simulation are compared to the performances of a small-size.

INTRODUCTION

More and more, ice storage tanks are installed in HVAC plant in the non residential buildings. These systems require less space (1m³/100kWhf 4°C to -4°C) than chilled water systems. Nevertheless, the technology is a little more complex. There is an other hydraulic circuit of glycol water which implies a water/glycol water exchanger, a safety to avoid to freeze water at the secondary loop. In the ice storage tank, there is phase change material (PCM) (water)/glycol water exchanger. Finally, it needs to produce cold energy under 0°C. So, it isn't cost affective to work with an absorption machine.

The main advantage of the ISS is the reduction of volume compare to the chilled water system. The ratio is 6. 6 times less cumbersome, 6 times less heavy. The ISS should be installed on a roof of a building, in the technical room or be buried.

Two types of ISS are widespread in the non residential buildings. The first technology is the internal melting system. The Phase Change Material (PCM) is contained in little elements named nodules made of polyolefin's. All the nodules are floating in the glycol water which carries or brings the cold energy. The problem of the expansion of the ice when changing phase is solved. The second technology is the external melting system. Generally, a coil is immersed in the PCM in the tank. These systems need less glycol water than internal melting system. The capacity is higher because there is more water volume for the same tank. But, the freezing zone is limited to avoid mechanical stresses on the wall of the tank.

The ISS are used for different purposes:

- Reduction of the electrical peak power consumption.
- Displacement of the electrical consumption from the on-peak hours to the off-peak hours.
- Sometimes, the ISS is used in parallel with the unit which produces the cold energy to increase the refrigerating plant power.
- An ISS is designed to store cold energy. However it could also be used to store heat.
- Occasionally, ISS are used as emergency units.

MODELLING OF THE ICE/WATER CHANGE PHASE ON COILS

TRNSYS model

A TRNSYS model was developed during years '80 within the framework of a European program. The objective was to design a model of latent energy storage compatible with a simulator of solar equipment largely used. This model made it possible to simulate the dynamic behaviour of a storage of PCMs (paraffin's, …) belonging to a solar installation made up of heat pumps, buried storages, internal or external melting,… It is easy to recognize the analogy between a traditional storage of ice on coils and the second configuration simulated by the model: Same geometrical disposition, the water/ice is the PCM, the fusion temperature is equal to 0°C. However, we will have to pay attention to certain differences which will be likely to involve modifications of the code:

- The transition interval between the solid phase and the liquid phase is null[1] :

$$T_{m1} = T_{fusion} = T_{m2}$$

- The cold energy to store has a mathematical negative value.

Geometrical description

The model of energy storage is able to simulate the dynamic behaviour of two types of storage. In the first configuration, the PCM is contained in cylinders and the coolant runs out around these cylinders (Figure 1).

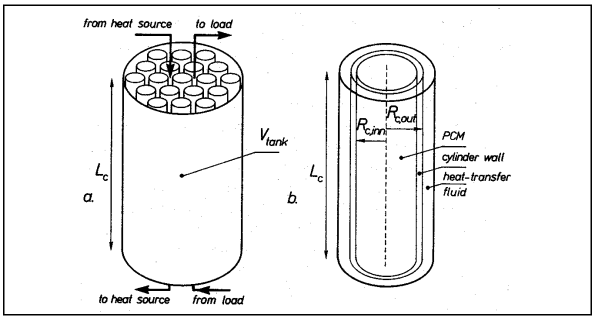

Figure 1 External melting

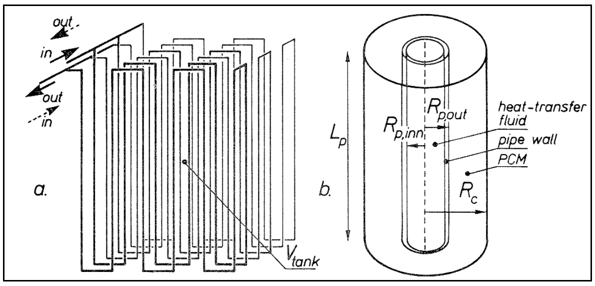

Figure 2 Internal melting

In the second, are pipes in which the coolant runs out are plunged in a tank containing the PCM (Figure 2). In this study, we will limit ourselves to the second form of storage: the storage of PCM (ice/water) on coils (internal melting case).

The enthalpy equation

The increase during the time of the quantity of energy contained in an arbitrary element of volume V (fixed in space) is equal to the assessment of the heat flows having crossed V through its surface A, if no external work applies to V and if no internal source of energy is considered. Moreover, while also considering that pressure is independent of time (absence of movement), we find the following equation:

$$\frac{d}{dt}\int_V \rho \cdot u(T) \cdot dV = \int_A \lambda \cdot grad(T) \cdot \bar{n} \cdot dA \qquad (1)$$

where n is the outgoing unit vector perpendicular to surface A. We will note that this relation does not take into account the interface of the phase change. To introduce this significant concept into our model, we make the assumption that the phase change temperature isn't exactly 0°C but is a restricted temperature range (around the melting point). We define the limits of this range:

- Lower fusion temperature T_{m1}
- Higher fusion temperature T_{m2}

The definition of energy (specific enthalpy) is given according to the temperature:

$$\begin{cases} T < T_{m1}: & u(T) = c_{p,s} \cdot T \\ T_{m1} < T < T_{m2}: & u(T) = c_{p,l} \cdot T + \dfrac{r_s (T - T_{m1})}{T_{m2} - T_{m1}} \\ T_{m2} < T: & u(T) = c_{p,l} \cdot T + r_s \end{cases} \qquad (2)$$

where $c_{p,l}$ and $c_{p,s}$ are respectively specific heats (at constant volume) of water in the liquid state and the solid state, rs the latent heat of fusion.

Finally, for the convenience of the presentation and the ease of mathematical handling, the concepts of enthalpy and temperature will be put in adimensional form:

$$u \rightarrow \quad \mathrm{H}(T) = \frac{u(T) - u(T_{m1})}{r_s} \qquad (3)$$

$$T \rightarrow \quad Q(T) = \begin{cases} \dfrac{c_{p,s}(T - T_{m1})}{r_s} & \text{if } T < T_{m1} \\ \dfrac{c_{p,l}(T - T_{m1})}{r_s} & \text{if } T > T_{m1} \end{cases} \qquad (4)$$

In these adimensional concepts, the real temperature (expressed in °C or K) is the parameter which will enable to follow the evolution of the phase change. To each value of the temperature corresponds a point in the graph of Figure 3.

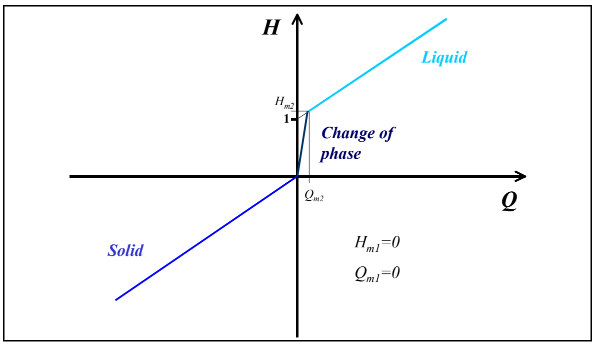

Figure 3 Relation between the enthalpy without dimension and the temperature without dimension.

1 The model developed in the years ' 80 simulated the phase shift of MCP such as paraffin's. Indeed, sometimes, this kind of materials presents a melting point which is not narrow but wide on a "zone of temperature". For example, the phase change (liquid-solid) of the medicinal paraffin is carried out between 40°C and 44°C.

Equation (4) could be simplified. The change of state will be modelled primarily by the variation in temperature of the PCM. This mathematical formulation will be easier in order to describe the problem involved by the moving boundary.

TRNSYS implementation: Type166.for

Thanks to the modularity of the program TRNSYS [14], the model could be implemented as a sub-routine in language FORTRAN. Type 166 was developed for version TRNSYS 15. TRNSYS interface requires the definition of inputs, parameters and outputs.

SIMULATION AND EXPERIMENT

Validation of design performances

Before speaking about a model integrating the equipments characterizing a complete installation HVAC (chiller, pipes, building, …), it is preferable to evaluate, even to validate, the model which we have just developed. Therefore, we will try to reproduce the performances announced by a manufacturer of ice storage. We will reproduce the assumed experimental conditions of the tests. We will try to reproduce the discharge curves of the storage as given by the manufacturer. These are the tests for which we have a maximum of information.

With the aim of approaching the reality, we will consider an existing ice storage. We will be interested in the family of storages of ice "CALMAC" because those present a similar technology to that which we modelled. The system with accumulation of ice includes:

- An insulated tank full of water (in polyethylene)
- An internal heat exchanger supplied with glycol water (in polyethylene)

The exchanger is a series of horizontal layers. Each level consists of two polyethylene pipes spirally curved. These two pipes are laid out to create a counter-flow exchanger. Two collectors (supply and return) connect the different levels. They are vertical pipes of larger section.

When we explore the table of the design characteristics of the manufacturer, we notice that this one proposes a type of storage compatible with future application developed below. Indeed, type 1190 is characterized by:

- Total capacity of 670 kWh
- Maximum discharge power of about 200 kW

To realise an simulation of this storage unit, it is necessary to define a series of parameters (Table 1).

As regards the physical and chemical parameters of materials and fluids used, we referred to articles and publications to identify the required values. As regards the other parameters (geometrical, …), we referred to photographs, plans of tanks, data of manufacturer exchanged by mails with the manufacturer. We also made certain assumptions allowing us to determine certain parameters as the number of pipes and their length. The most significant assumptions are as follows:

- Conservation of the total heat-transferring surface between water and glycol water,
- Each pipe is connected at the top and at the bottom of storage (L_p = Height of the tank).

Table 1 Parameters and inputs of the model(validation of design performances)

Parameters - Inputs	TRNSYS Units	Parameters - Inputs	TRNSYS Units
λ_f	2.304 kJ/h.m.°C	λ_w	1.26 kJ/h.m.°C
η_f	2.35 kg/h.m	ρ_f	1036 kg/m³
T_{env}	21°C	$c_{p,f}$	3.87 kJ/kg.°C
ρ	917 kg/m³	T_{st}	-4 °C
$c_{p,s}$	2.04 kJ/kg.°C	U	8.002 kJ/h.m².°C
$c_{p,l}$	4.21 kJ/kg.°C	N_p	2056
λ_s	8.1 kJ/h.m.°C	L_p	2.222 m
λ_t	5.058 kJ/h.m.°C	V_{tank}	6.336 m³
λ_l	2.016 kJ/h.m.°C	A_{tank}	25.63 m²
T_{m1}	0°C	$R_{p,inn}$	0.00635 m
T_{m2}	0°C	$R_{p,out}$	0.007938 m
r_s	333.3 kJ/kg	M	50
ρ_w	920 kg/m³	N	10
$c_{p,w}$	2.3 kJ/kg.°C	Fr	-

To reproduce the results of the tests, certain data are missing. We are limited to the reproduction of the discharge curves provided by the manufacturer. Moreover, For the realization of the layouts of the curves, the manufacturer introduced two parameters: a correction factor and an overlap factor. However, we will try the exercise.

An example of results diagram is given on Figure 4. If we compare hour per hour the results of the "manufacturer" tests with our results, we notice a certain difference. Our curves seem more linear (at the beginning of discharge). The developed power is slightly decreasing with the time. We calculated the relative errors. We will notice simply that the relative error on the model varies between 0.09% and 73.25%. The absolute maximum error is at the beginning of each discharge. Taking into account the inaccuracies related to the ignorance of the conditions of the tests, we can affirm that with an average relative error of 18.29%, our model is not far from reality.

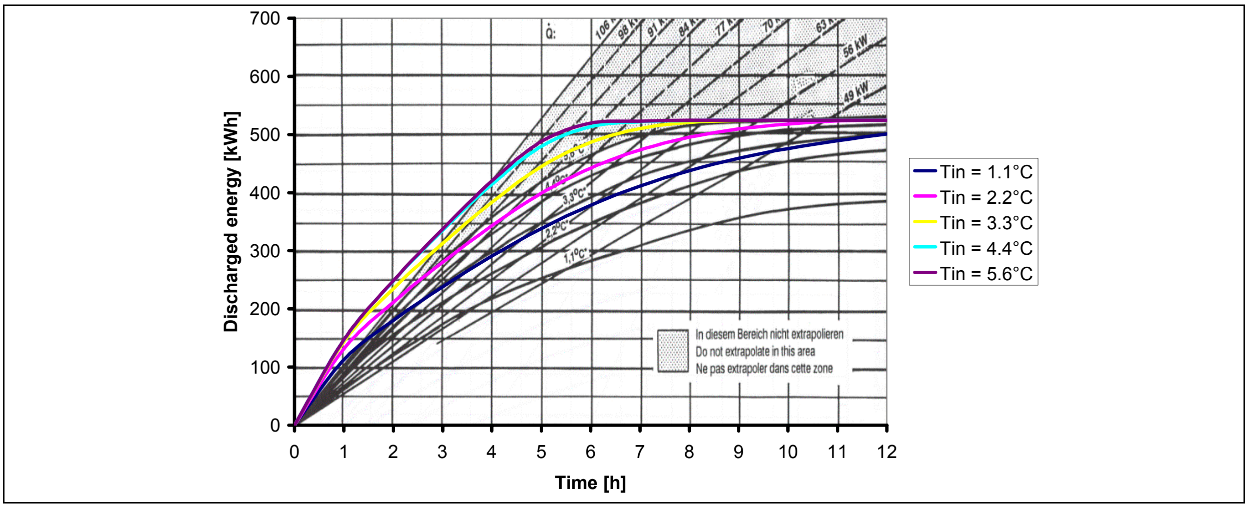

Figure 4 Simulated and manufacture data discharging curve

Experimental validation

After having carried out a validation on the basis of data manufacturer, it’s important to validate the model with experimental results. A small-size test bench has been built to reproduce the experimental conditions. It has been added to an existing HVAC plant of an experimental building devoted to fog production located on FUL campus. The test bench is composed of:

- A reversible heat-pump (1 Maneurop MTZ28 compressor - R404a)
- An external static air exchanger (Evaporator or condenser)
- A chilled water tank of 500 litres (cooling by the reversible heat-pump or by the ISS)
- A hot water tank of 1000 litres (heating by the heat-pump or by the solar collectors)
- A small-size ice storage tank of 10 kWh (internal melting)
- 5 distribution circuits to climate the building (offices, buffer, test chamber floor, test chamber ceiling and AHU coils)

On the other side of this system, there is a 300 litres water tank devoted to fog production. It is heated by the solar collectors or by a electrical resistance.

The acquisition system is composed of:

- 2 immerged temperature sensors (supply and return glycol water temperature)
- 1 volumetric flow rate (return flow rate)

To validate the model, we will try to reproduce the behaviour of the ice storage tank in charging mode. The inputs of the simulation are the flow rate and the supply temperature of glycol water. The output is the return temperature. The stored energy is not measured. The new parameters of the model are not the same as the design validation. The characteristics of tank and the internal heat exchanger are different. The new parameters are shown on Table 2.

Figure 6 shows the results of the experimental validation. On the top of the figure, the temperatures and the energy are represented. During the liquid phase, the simulated curve and the measured curve are the same. During the first part of the transition phase, a deviation of the simulated return temperature is observed. The relative error between the simulated and the measured values is about 170%, this represents only 1.3°C. Next, when a greater part of ice is done, the model is good. The two temperature curves are the same. The simulated energy curve is compatible with the capacity of the ice storage tank. From 17°C to 0°C, the tank has stored about 9 kWh of cooling energy.

Table 2 Parameters and inputs of the model(validation of experimental performances)

Parameters - Inputs	TRNSYS Units	Parameters - Inputs	TRNSYS Units
λ_f	2.304 kJ/h.m.°C	λ_w	1.152 kJ/h.m.°C
η_f	2.35 kg/h.m	ρ_f	1036 kg/m³
T_{env}	21°C	$c_{p,f}$	3.84 kJ/kg.°C
ρ	917 kg/m³	T_{st}	17 °C
$c_{p,s}$	2.04 kJ/kg.°C	U	5.672 kJ/h.m².°C
$c_{p,l}$	4.21 kJ/kg.°C	N_p	24
λ_s	8.1 kJ/h.m.°C	L_p	7 m
λ_t	0.25 kJ/h.m.°C	V_{tank}	0.265 m³
λ_l	2.013 kJ/h.m.°C	A_{tank}	2.373 m²
T_{m1}	-2°C	$R_{p,inn}$	0.0004 m
T_{m2}	-0.5°C	$R_{p,out}$	0.0005 m
r_s	333.3 kJ/kg	M	3
ρ_w	1260 kg/m³	N	40
$c_{p,w}$	2.09 kJ/kg.°C	Fr	0.7

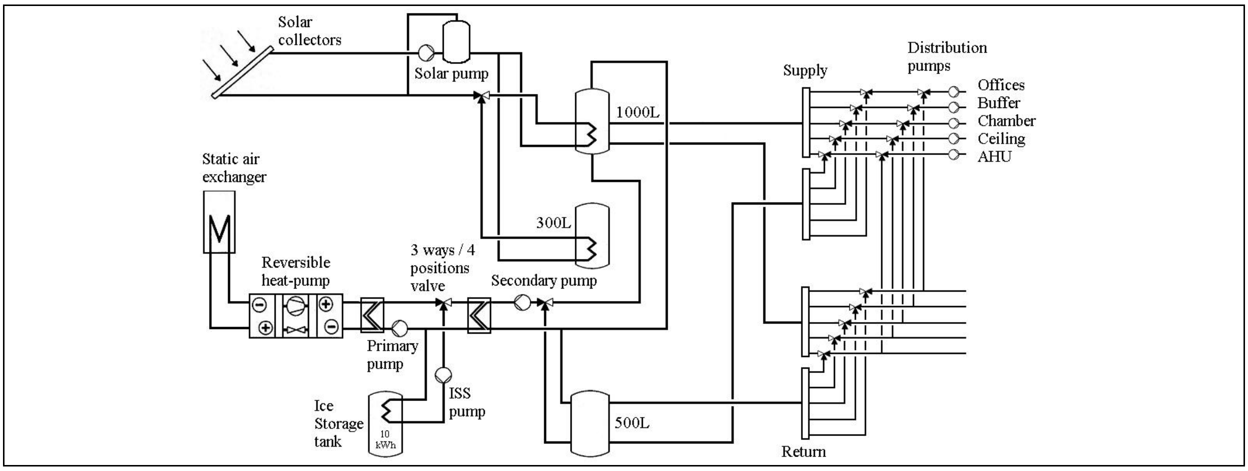

Figure 5 Test bench scheme

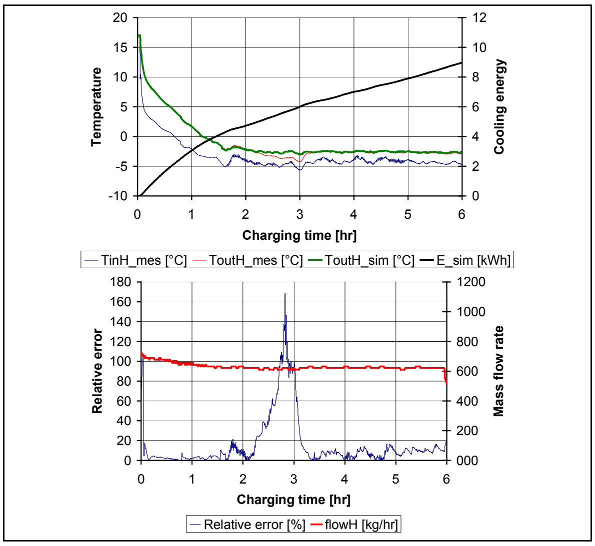

Figure 6 Experimental validation

DISCUSSION AND RESULT ANALYSIS

First, the validation of the design performances has been done. The discharge mode of a conventional ISS of about 700 kWh has been simulated. The result of the comparison with the manufacture data allow to say that the validity of the model is quite good. In fact, the manufacture data aren't well-known and the identification of the parameters has been done without a good feeling of the system (pictures, data extracted from folders,…). This first exercise was nevertheless1 successful.

Next, a test bench has been installed to compare the behaviour of the experimental system and the simulation model. After tuning some parameters (λ_t, T_{m1}, T_{m2}, M, N), the mean absolute error of the model is about 0.2°C. The maximum absolute error is 1.3°C. The model is globally correct.

The thermal conductivity of the PCM in transition phase has decreased from 5.058 to 0.25 kJ/h.m.°C. The first value is the mean value of the solid and liquid phases. It was used for the validation of the design performances because the transition phase doesn't exist and no thermodynamic parameters are available. The transition temperature range has been widened (from –0.1..0.1°C to –2..-0.5 °C). That is explained by the presence of particles in the water (rust, waste,…) which accelerate the nucleation but decrease the growing effect.

Table 3 Experimental errors

	Relative error [%]	Absolute error [°C]
minimum	0	0.0
maximum	168	1.3
mean	16	0.2

APPLICATION: SIMULATION OF A TYPICAL HVAC PRIMARY PLANT EQUIPPED WITH AN ICE STORAGE UNIT

Presentation of the plant

The typical HVAC plant is the primary HVAC plant of the Fondation Universitaire Luxembourgeoise. The scheme is given in Figure 7. The primary HVAC plant of FUL building is composed of refrigerating units with reciprocating chillers. Three machines are installed.

The TRNSYS library offers several models of buildings and "structures". These models go from a simplified model built on concept of "degree-hours" until a dynamic multi-zone building model. For this study, the simplest model, namely "Type 12" was used.

The building is connected to a chilled water tank of 5m³ volume. A two speed pump identified PFUL on the scheme discharges the storage and supplies the chilled water in the building. The low speed is turned on as soon as the temperature in the building exceeds 15°C; high speed, as soon as this

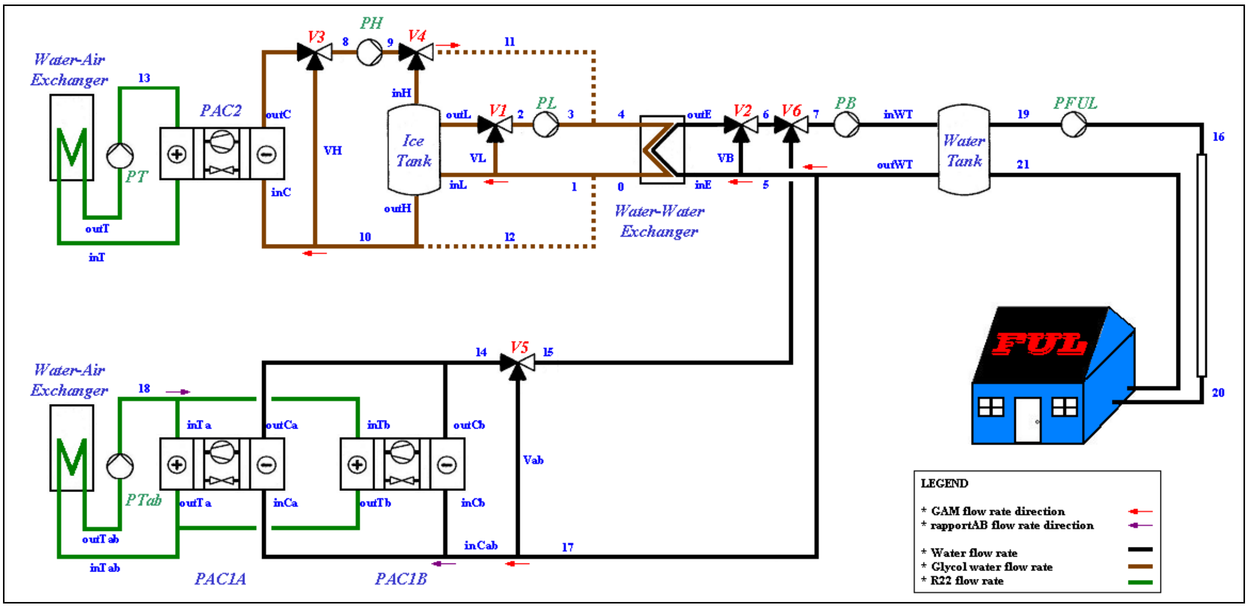

Figure 7 Scheme of the primary HVAC plant of FUL building

temperature exceeds 25°C. At this level, there is no temperature control of the water supplied to the building.

The chilled water of the 5m³ tank is renewed thanks to the pump PB. It is started as soon as the building is occupied (between 08:00 A.M. and 05:00 P.M.). The V6 valve directs the production of cold according to the resources of the installation and priorities determined by the manager. It is at this level of the installation that the discharge of the ice storage is managed. Three methods of management (operating strategies) were simulated:

1. **"ALTERNATIVE LOAD"** : STORAGE DISCHARGE FOLLOWED BY DIRECT PRODUCTION
   A fixed quantity of cooling energy is stored. As soon as this energy quantity is obtained, the installation shifts to the direct production mode.
2. **"FULL LOAD"** : STORAGE DISCHARGE ONLY (WITHOUT DIRECT PRODUCTION)
   All the cooling energy available is taken from the storage sufficiently charged or not.
3. **"DIRECT LOAD"** : DIRECT PRODUCTION ONLY (WITHOUT STORAGE DISCHARGE)
   All the cooling energy is produced directly by the direct production refrigerating units PAC 1 A and B, whatever storage is charged or not.

The V5 valve then controls the temperature in pipe number 15 around 5°C. The V1 valve maintains a temperature of 4°C at the inlet of the exchanger side storage. At the outlet of the exchanger side chilled water tank, V2 valve controls the temperature of the chilled water around 5°C.The V3 valve maintains a charging temperature of –4°C.

Discussion and result analysis

After having simulated three different operating strategies for an ice storage within a primary HVAC plant, we qualitatively analysed the produced results. Continuing our analysis, we will calculate for each simulation two quantities qualifying the economical state of management. These two quantities are as follows:

- The ***cost*** of the cold energy production. This cost is calculated according to a tariff determined by the electricity distributor: tariff "binôme A force motrice" in this case.
- The cumulated deviation of comfort in °C.h which represents the integral of the variations in temperature (inside the building) compared to two temperatures of comfort (for example 21°C and 24°C) during hours of occupation. It would be more judicious to speak about "***cumulated (deviation of) discomfort***".

Figure 8 shows a graph which represents the performances of each one of these three storage management strategies. On the x axis, we have the cumulated deviation of discomfort; on the y axis, the cost.

On this diagram, the optimal zone is the blue colour region (left bottom corner). The cost is lower and the comfort is higher. The more the point associated with a strategy is located in the right top corner, the more this operating strategy is penalizing at both the points of view of comfort and electrical cost.

The strategies "direct load" and "alternative load" are comparable from the point of view of comfort. From the financial point of view, to manage the cooling load with a storage is better. The electrical peak consumption of the beginning of day is smoothed because storage has the property to develop a significant refrigerating power at the beginning of discharge.

The strategy "full load" is rather far away from the two others in the diagram. Its use is cheaper than the two others. Comfort is also worse. To appreciate this strategy as good as possible, to evaluate its advantages, it would be interesting to carry out a simulation of the installation provided with a storage of greater capacity.

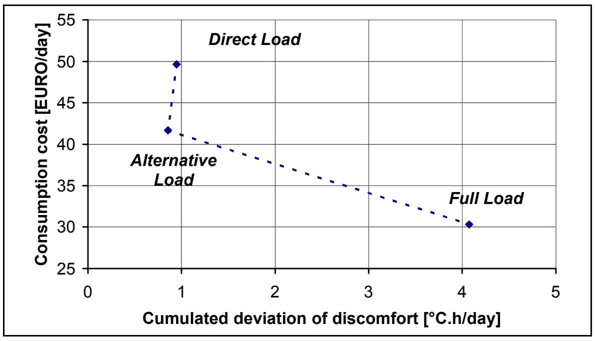

Figure 8 Evaluation of the operating strategies

Finally, after comparison of these three strategies, we can conclude that:

- The use of a storage within the installation is cost-saving. It is possible to reduce the electric power term and thus to strongly decrease the cost of electricity.
- The design of the storage and the refrigerating unit used to charge the storage is very important. In "full load" mode, if the capacity of storage is limited or if the power of the chiller is too low, comfort could not be ensured in the building.

CONCLUSION

The ice storage systems are generally used to decrease the electrical cost of the HVAC plant consumption. They shift the peak power and the consumption from on-peak hours to off-peak hours. Sometimes, they are used to increase the refrigerating power plant. But, their main objective is to realise cost-savings.

Before the installation of such systems, the design of the chillers and the storage tanks is crucial. It determines the effectiveness of the cost-saving. The cooling power of the chillers is calculated to produce the whole cold energy of the day during the hours concerned by this purpose:

$$Power = \frac{Energy}{Hours}$$

The storage capacity is equal to the amount of energy which could not be produced and consumed at the same time.

This conventional design of a ISS has same limitations. First of all, it doesn't take into account the variation of the COP during the 24 hours cycle. The temperature levels of the condenser and the evaporator of the chiller and the weather conditions change during this period are not considered either. Moreover, the cooling load changes throughout the seasons. The nominal conditions are reached during summer but not in winter and in autumn. In such cases, the operating strategies had to be adapted continually to be cost effective.

Before design, the simulation of some operating strategies of some ISS configurations should be more advantageous. To determine the operating strategy adapted to the cooling load and to the season allows to better carry out the design. And, ultimately, the simulation could be used to validate the design or to verify the balance between:

- the design of the ISS and the chiller,
- the design of the operating strategy.

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APPENDIX: NOMENCLATURE OF THE MODEL

Inputs

$T_{in,H}$	temperature of the refrigerating fluid at the entry of the tank
$\Phi_{m,in,H}$	mass throughput of the refrigerant at the entry of the tank
$T_{in,L}$	temperature of the coolant at the entry of the discharge
$\Phi_{m,in,L}$	mass throughput of the coolant at the entry of the discharge
λ_f	thermal conductivity of the refrigerating fluid
η_f	dynamic viscosity of the refrigerating fluid
T_{env}	ambient temperature

Parametres

mode = 2

ρ	density of the PCM
$c_{p,s}$	specific heat of the PCM in solid phase
$c_{p,t}$	specific heat of the PCM in phase of transition
$c_{p,l}$	specific heat of the PCM in liquid phase
λ_s	thermal conductivity of the PCM in solid phase
λ_t	thermal conductivity of the PCM in phase of transition
λ_l	thermal conductivity from the PCM in liquid phase
T_{m1}	low temperature of fusion
T_{m2}	high temperature of fusion
r_s	latent heat of fusion of the PCM
ρ_w	density of material constitutive of the pipe
$c_{p,w}$	specific heat of material constitutive of the pipe
λ_w	thermal conductivity of material constitutive of pipe
ρ_f	density of the refrigerating fluid
$c_{p,f}$	heat specific of the refrigerating fluid
T_{st}	initial temperature of the reserve
U	coefficient of total transfer of heat of the tank (external wall)
Np	numbers pipes (parallel flows)
Lp	length of a pipe
$Vtank$	volume of the tank
$Atank$	surface total tank
$R_{p,inn}$	interior radius of a pipe
$R_{p,out}$	radius external of a pipe
M	numbers axial elements
N	numbers radial elements
Fr	coefficient of relaxation (0.1).

Outputs

$T_{out,H}$	temperature of the refrigerating fluid on the outlet side of the tank
$\Phi_{,out,H}$	mass throughput of the refrigerant on the outlet side of the tank
$T_{out,L}$	temperature of the coolant at the output of the discharge
$\Phi_{,out,L}$	mass throughput of the coolant at the output of the discharge
$P_{out,H}$	Quantity of energy which is withdrawn from the cold source
P_{env}	Quantité of energy lost with the environment
ΔE_{pcm}	balances energy (latent and sensitive) contents in the PCM (compared to the initial situation T_{st})
ΔE_w	balances energy (sensitive) contents in the pipes (compared to the initial situation T_{st})
ΔE_f	balances energy (sensitive) contents in the coolant (compared to the initial situation T_{st})
ΔE_{tank}	total balance energy (compared to the initial situation T_{st})
Per_s	fraction of PCM in solid phase
Per_t	fraction of PCM in phase of transition
Per_l	fraction from PCM in liquid phase
$T_{w,mean}$	average temperature of the pipes
$T_{f,mean}$	mean average temperature of the refrigerating fluid