HEAT PUMPING AND REVERSIBLE AIR CONDITIONING; RETROFIT OPPORTUNITIES IN A LABORATORY BUILDING

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ABSTRACT

This paper presents the main results coming from a case study performed in the frame of the IEA_ECBCS annex 48 project. A laboratory building is submitted to a detailed energy audit in order to identify the most attractive retrofit potentials.

KEYWORDS

Heat pumping, HVAC, modelling, simulation

INTRODUCTION

The aim of IEA-ECBCS annex 48 is to promote the most efficient combinations of heating and cooling techniques in air conditioning.
Special attention is paid to all possibilities of recovering condenser heat of using the chillers in heat pumping mode.
Several case studies are performed in the frame of this project.
The results presented here are extracted from one of the Belgian case studies, in continuation with a previous paper (Aparecida et al. 2007).

THE BUILDING

The results already presented in the previous paper (Aparecida et al. 2007) are here very briefly summarized.

The present case study is dealing with a laboratory building erected in 2003 in the region of Liège.

This building is subdivided into 3 main zones (Figure 1). Most of the HVAC demanded is concentrated in zones I (the offices) and III (the laboratories).

Description of the HVAC system and power plant

Zone I – Offices

The ventilation of this zone is ensured between 6:00 and 20:00, five days per week, at a rate of 5000 m³/h (full fresh air). Thermal comfort is ensured by 50 heating & cooling terminal units (fan coils) distributed in all the zone.

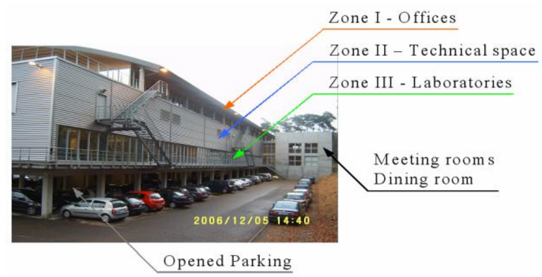

Figure 1: Building south frontage

Zone III – Laboratories

The laboratories are supplied with 33000 m³/h of fresh air fully conditioned through three AHU's, equipped with steam humidifiers, (Figure 2). The high flow is a need for the laboratory work.

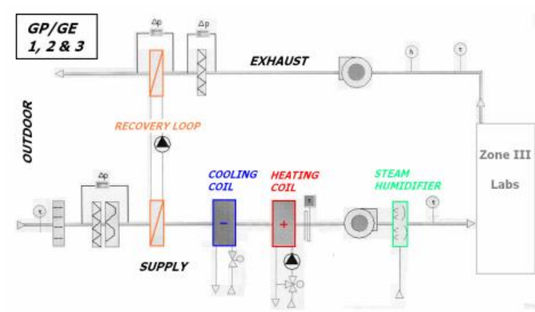

Figure 2: Laboratories AHU's

Ventilation of this zone is ensured 24h per day, 7 days per week , all the year. About 60 people work in the laboratories during day (between 8:00 and 17:00) and, 6 other people are present in the zone 24h/24.

Power Plant

The hot water production is ensured, at a nominal temperature of 80°C, by two modern gas condensing boilers of about 300kW each. The nominal return temperature is about 60°C.

A R134a chiller of 400kW ensures the cold water production at 7°C. This chiller is equipped with an air-cooled condenser.

Retrofit opportunities

The main idea is, in the present case, to use the extracted air as heat source for heat pumping and fresh air heating.

This idea is justified by the very large air flow rate (38000 m³/h) of hot and humid air (about 23°C/50%) extracted from the building. Noting that the high humidity is due to tne comfort exigences of the laboratory work.

First opportunity: replacing the air-cooled chiller by a water-cooled "thermo-frigo pump" (i.e. a same machine usable as heat and cold generator) and supplying all terminal units with hot water coming from the condenser of this machine (at a temperature which shouldn't overpass 55°C).

Second opportunity (in addition to the first one): supplying the heating coils of the different AHU in the same way and introducing a "change-over" process to get additional heat transfer areas.

The "change-over" will consist here in supplying the cooling coils with hot water, when the cooling demand has disappeared.

Other opportunities:

- Installing (stratified) thermal storage to make possible some compensation between heating and cooling demands in mid-season and to allow load-shifting from peak hours to off-peak hours.

- Reducing the air flow rates during low occupation periods (this has still to be discussed because of very strict air quality requirements in laboratories).

Reversible heat pumping

Generally, two heat pumping modes are envisaged (IEA Annex48) :

- the condenser heat recovery (simultaneous cooling and heating demands),
- the reversible heat pumping (non-simultaneous cooling and heating demands).

In spite of this distinction, the principle is unique : a same (water/water) refrigeration cycle should be used to supply both chilled and hot water networks, through its evaporator and condenser respectively.

In case of so-called "condenser heat recovery", the evaporator is already connected to the chilled water network. If the condenser is actually cooled by water, this water is cooled itself with the help of a cooling tower.

In the case of reversible heat pumping, cooling and heating demands are rarely simultaneous. So, the two functioning modes of the machine (chiller and heatpump) are distinct. This makes that more powerful heat source (on evaporator side) and heat sink (on condenser side) are required.

The building considered is characterized by comparable (simultaneous or not) heating and cooling demands, according to the period of the year. The winter is mainly characterized by its heating demand. During this season, the cooling demand is almost or completely null. During summer, the building heating and cooling demands are alternated, according to the day/night cycle. However, cooling and heating demands could be sometimes simultaneous in different rooms or parts of the building.

So, in the present case, according to the period of the year and to the simultaneity of heating and cooling demands, both reversible heat pumping and condenser heat recovery strategies can be used to satisfy the building cooling and heating demands.

To make it possible, the actual air cooled chiller has to be replaced by an air/water or water only cooled chiller. Because of size and practical considerations, the first solution has been chosen. The new machine will be cooled with two condensers, connected in parallel : one air cooled and one water cooled. The first (air-cooled) condenser is a fan-coil assembly as in the existing machine. The second (water-cooled) condenser allows the recovery of condenser heat. A three-way valve ensures the control of the machine and the supplying of one or both condensers (Figure 3).

The following water temperature regimes are considered:

- Between 4 and 10°C on evaporator side;
- Between 40 and 55°C on condenser side.

The exisiting boilers are kept as "boosters", downstream of the heat pump (Figure 3).

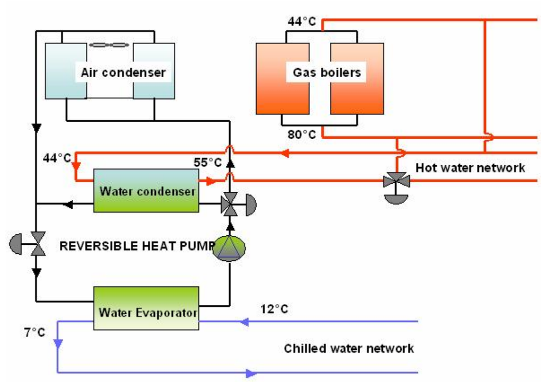

Figure 3: Reversible heat pump scheme

Whenever the heating of the zones considered can be ensured by hot water at 55°C or less, the boilers can still be used in condensing (high efficiency) mode.

On the heat source side, the available heat must be recovered using additional water/air coils. These auxiliary heat transfer surfaces must be installed in the extraction duct, downstream of the existing recovery coils. To recover the largest part of the heat available, the heat pump is supposed to be able to work with water temperatures going down until 9 and 4°C at evaporator supply and exhaust respectively, see Figure 8 hereafter.

The indoor air is extracted at 23°C/50% and at the rate of 33000m³/h. Downstream of the direct recovery coil, the air temperature varies between 12°C (for a cold day with direct recovery in heating mode) and 26°C (or more, for a hot day, with direct recovery in cooling mode). The additional air/water coils must be designed in such way to be able to fully supply the heat pump evaporator. Practical considerations, as space available in extraction ducts, as well as fan and and pump consumptions, must be taken into account.

Change over technique and re-startings

Heating the building with hot water at 55°C could be problematic. Indeed, the actual heat exchangers (AHU and TU heating coils) are designed to be supplied with hot water at 80°C. Lowering the water supply temperature from 80°C to 55°C could reduce too much the heating power available. Two adaptations will be made to ensure the satisfaction of the heating demand : the use of a change over technique and the programming of new re-starting strategies.

The laboratories are only heated by the air handling units GP1,2 & 3, Figure 2, in which additional heating areas must be found in order to ensure sufficient heating power. Hopefully, heating and cooling demands are never simultaneous in a same zone. So, during heating periods, the cooling coil constitutes a (big) un-used heat exchanger. The change over technique consists in using this heat exchanger to post-heat the air. For a given air handling unit and for a parallel supplying of the two coils the change over technique is illustrated in Figure 4. The possibility of series supplying (to increase the counter-flow effect) is also hereafter considered.

The change over technique has another advantage: it allows, thanks to extended heat transfer surface, a lowering of hot water temperature, and so, an increasing of heat pump performances.

The offices are mainly heated by terminal units. The contribution of the air handlers is very limited in this zone (considering the limited air flow rate). The AHU is used to bring the air to an acceptable temperature and not really to heat the zone. The Terminal units are often supplied with hot water at reduced temperature. If the air and water flow rates are maintained at high level, the heating of the zone can be ensured with lower temperature water (55°C). High temperatures (80°C) are only required for heating peaks, for example in early morning.

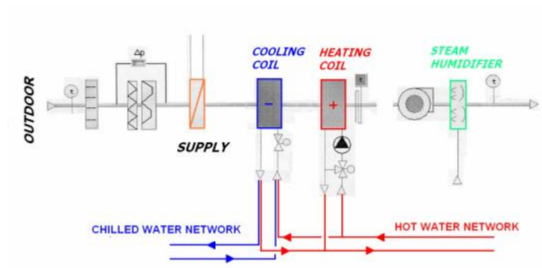

Figure 4 : Change over technique

Supplying the TU heating coils with hot water at 55°C could pose problems during re-starting. An easy adaptation of the control strategy consists in adopting an earlier re-startings during the coldest periods. Example: in winter, instead of beginning to heat at 6 am during the week, the re-starting will be fixed 4 or 5 hours earlier.

Thermal Energy Storage

The idea of adding a thermal energy storage comes from the analysis of the alternation between the heating and cooling power demands: for example during a only-heat demand period, the cold rejection of the evaporator can be stored for further use during a following cooling demand period. This is not a new idea (Dorgan & al. 1999).

A typical example is presented in the Figure 5 between 22:00 and 10:00, the heat production (about 610 kWh) is ensured by the heat pump. At the same time, on the evaporator side, the produced cold energy (about 457.5 kWh) is stored in a storage tank of about 80m³ (corresponding to 465 kWh, for a pure water storage, with a ΔT of 5°C).

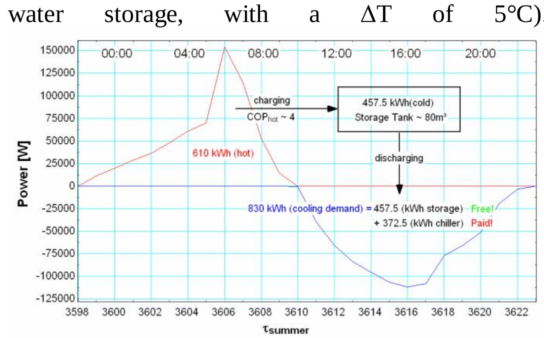

Figure 5: Example of use of thermal storage

From 10:00 until the drain of the storage, the cooling demand is ensured by the stored cold energy. Except for the water pump consumption, this part of the cooling is totally free. The remaining cooling demand (372.5 kWh) is satisfied by the chiller.

Even if the heating demand is not sufficient to satisfy the following cooling demand, it is more interesting to produce the cold energy during the night, in order to use electricity of lower costs. In fact, the use of this storage allows to move the chiller consumption from day to night. This technique is called the "load shifting".

The thermal energy storage system can be used to increase the number of hours during which the chiller operates at high efficiency, thanks to an optimal control strategy (the discharge rate can be controlled in such a way to keep as much as possible the chiller at its optimal operational regime. Of course, the actual impact of such control strategy depends on the part-load characteristics of the chiller (Roth et al., 2006).

Moreover, the thermal losses of the storage tank (currently 1 to 5% per day) must be carefully identified.

A possible scheme of the plant retrofit is presented in Figure 6.

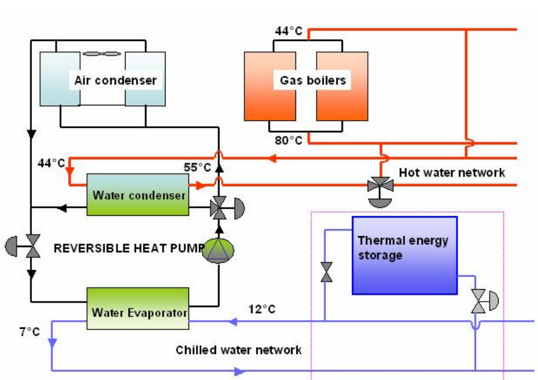

Figure 6: Installation with thermal storage

MODELLING & CONTROL

All the retrofit potentials are evaluated by simulation. The main simulation models are presented hereafter.

Heat Pump Model

The existing chiller is equipped with two classical reciprocating compressors and only one air cooled condenser. The new machine will be equipped with one screw compressor, one air cooled condenser and one water condenser (for heat recovery and reversibility). As it was the case for the air cooled chiller, a limited quantity of data is available for the heat pump.

To simplify the modelling, the machine is supposed to have two functioning modes :

- one air cooled chiller mode (using air condenser),
- one water cooled heat pump mode (using water condenser).

These modes could be sometimes superposed, but the dynamics and the interferences between the two modes (performances variation according to the repartition of the heat load on the two condensers) are not actually taken into account and the functioning modes are modelled by considering two distinct machines.

A correlation model is developed on the basis of (full load) manufacturer data. The bond between the full load and the part load regimes is made by referring to a unique part load curve. This curve is defined by the manufacturer in cooling mode. A third correlation is generated to define the condenser power in heat pump mode,.

The two screw compressors of the new machine are equipped with slide valves controlled by a microprocessor system. This offers the possibility of a more efficient part load control than in the existing chiller. Indeed, these screw compressors have a large operating range (between 25-100% each, so between 12.5 – 100 % for the machine). Part load compressors losses are therefore very reduced and there is no degradation of the corresponding COP. This fact is shown in Figure 7: the curve of electricity consumption part load factor as function of cooling power part load factor is very near to a 45° line. In the calculation, the part load curve is supposed to be equal in heat pump and chiller modes.

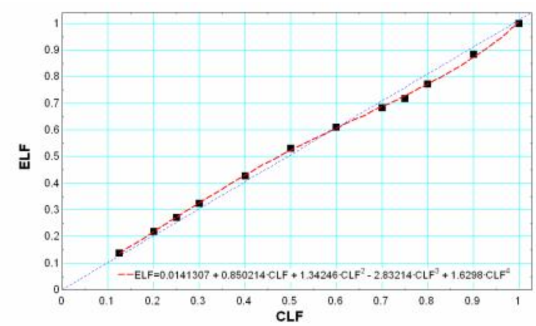

Figure 7: Screw compressor part load law

Heat Source Model

The aim of this model is not the accurate calculation of the recovered heat at the heat source of the heat pump but only the estimation of the available (or recoverable) power by installing an additional water/air coil in the extraction duct and by connecting it to the heat pump evaporator (Figure 8).

Of course, the heat source available power will not always be totally used; it is computed in order to know the limitations of the heat source and the rate of intervention of the boilers (as back boosting heating devices).

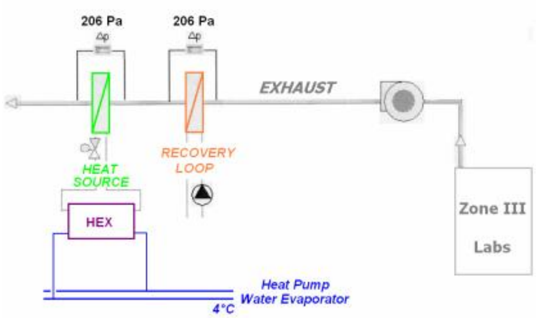

Figure 8: Additional water/air coil

A first part of the analysis consists in identify the extracted air state (temperature and humidity) at the exhaust of the recovery coil. This calculation is made on GP/GE 1,2 & 3.

Then, the amount of heat available is computed by using the extraction air state at the exhaust of each recovery coil as an input for the simulation of the supplementary coil connected to the evaporator (this model is the same as for a classical cooling coil). Of course, this component would disserve to be optimized (in terms of heat transfer area and corresponding pressure drops on air side). But, to make it easier in a first approach, the additional recovery coil of each air handling unit is supposed to be the same as the existing one. This crude choice is justified because of the dimensions of the duct. The thermal effectiveness of the new coil is used to compute the maximal recoverable power. The coil is supplied with chilled water coming from the heat pump evaporator at 4°C. The choice of minimal evaporator exhaust temperature is arbitrary, but it allows to calculate the maximal power recoverable. The water flow rate is also fixed at a value corresponding to a temperature variation of 5 K .

As the existing coils of the recovery loop, the new coil of each air handling unit has to be supplied with glycol-water in order to avoid any freezing risk (when the system is stopped).
Supplying the whole cold distribution network with glycol-water in place of pure water would generate important over-costs and other inconveniences. This can be avoided thanks to the use of a brine/water heat exchanger installed between the glycol-water and the pure water circuits, as shown in Figure 8.

Change Over Modelling

As already explained, the change over technique will allow us to provide to the air a same heating-up, with hot water at 55°C instead of 80°C.

The modelling of the change over technique is made in three steps :

- The change over technique is modelled on each air handling unit (GP 1,2&3) and the characteristics of a larger "equivalent" heating coil are identified;
- The three "extended" heating coils are agregated into one;
- The fictitous heating coil is implemented in the AHU model.

The most important parameters in the change over technique, are related to the assembly in parallel or in series of the two coils of each AHU and to the definition of the water flow rates.

When connected in series on water side, the two coils are, of course, supplied with a same flow rate, but at different temperatures.

When the two coils are connected in parallel, an optimal repartition of both flow rates can be looked for, in such way to get a minimal pumping power.

For both assembly types, the main constrain is to maintain a same exhaust air temperature as in the nominal functioning mode (heating coil supplied with water at 80°C).

The dry heat transfer coefficients of the different coils are already known. The values presented in Table 1 correspond to the coils of the AHU's GP2 and 3.

Table 1: Heat transfer coefficient of different coils

	Heating Coil			Cooling Coil		
Air Flow Rate [m³/h]	Water Flow Rate [m³/h]	AU value [W/K]	Coil Eff. [%]	Water Flow Rate [m³/h]	AU value [W/K]	Coil Eff. [%]
11500	5.4	2371	40	15	6640	78

For all cooling coils, both air-side and water-side thermal resistances are known at nominal water flow rates.

For the heating coils, only the global thermal resistance (equal to the inverse of the AU value) is known and a guess has to be made in order to split it into air-side and water-side resistances.

A classical guess consists in assuming that both resistances are of the same order of magnitude in nominal conditions.

Both (convective) thermal resistances must vary with corresponding flow rates.
The following laws are used in the simulation (Lebrun et al. 2007):

$$R_{a,coolingcoil} = R_{a,coolingcoil,n} \cdot \left[\frac{\dot{M}_{a,coolingcoil,n}}{\dot{M}_{a,coolingcoil}}\right]^{0.6}$$

$$R_{w,coolingcoil} = R_{w,coolingcoil,n} \cdot \left[\frac{\dot{M}_{w,coolingcoil,n}}{\dot{M}_{w,coolingcoil}}\right]^{0.8}$$

$$R_{tot,coolingcoil} = R_{a,coolingcoil} + R_{w,coolingcoil}$$

$$AU_{coolingcoil} = \frac{1}{R_{tot,coolingcoil}}$$

Parallel Assembly

The two coils are supplied with hot water coming directly from the hot water network. The principle of the calculations and the main results are presented in Figure 9, that corresponds to the optimal values.

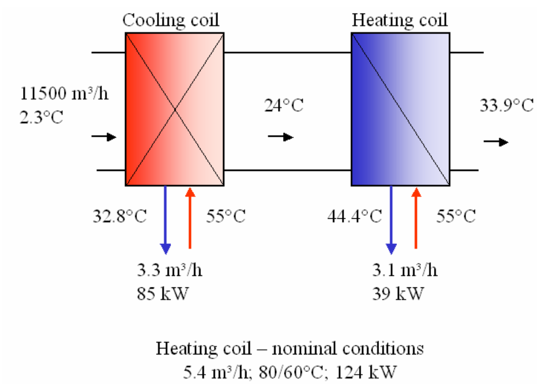

Figure 9: Change Over – Parallel Assembly Model

Imposing the exhaust air temperature, and using the water flow rate in the heating coil as the parameter, the model gives the required cooling coil water flow rate.

Serie assembly

The counter-flow effect is maximized by supplying the two coils in series:

- The heating coil is supplied with hot water coming from the hot water network;

- The cooling coil is supplied with the hot water coming from the heating coil.

The calculation principle is presented in Figure 10. Supply and exhaust air states and air flowrate have to be the same as when having only the heating coil supplied in nominal conditions.

With the new hot water supply temperature (55°C), the water flow rate (4 m³/h) is computed to obtain the wanted exhaust air temperature (33.9°C).

All other temperatures, AU values, effectivenesses, and heat transfers are computed by the model. It appears that the counter-flow effect is maximized and that the water temperature vary between 55 and 28.6°C.

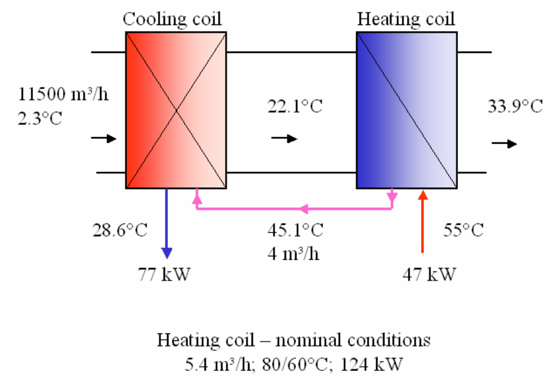

Figure 10: Change Over – Serie Assembly Model

It is also interesting to note that the required water flow rate is lower than the nominal water flow rate.

Water pressure drops and pumping power are computed for the two coils come from manufacturer data as well as the coefficients k_{hc} and k_{cc}. The pressure drops are varying with the water flow according to the following laws:

$$\Delta P_{w,hc} = k_{hc} \cdot \dot{V}_{w,m3h,hc}^2$$

$$\Delta P_{w,cc} = k_{cc} \cdot \dot{V}_{w,m3h,cc}^2$$

For the considered water flow rate (4 m³/h), the pressure loss in the cooling and heating coils are 1873 Pa and 1334 Pa, respectively. Considering a pump efficiency of about 20%, this would give a power of about 18 W. Of course, this corresponds to part of the consumption required by the coils only.

The actual pump consumption is much bigger, because of the dominating effect of other pressure drops (along the whole piping network and across the valves).

Choice and globalisation work

According to practical and technical considerations, the series assembly is here preferred. This assembly mode has the advantage of a reduced total water flow rate (lower than the nominal water flow rate), allowing a reduction of pressure drop in the whole hot water network. This reduction is large enough to compensate the increase of pressure drop due to the series assembling of the two coils.

At the contrary, the parallel assembly requires an increase of water flow rate and therfore an increase of pressure drop which cannot be compensated by the reduction of pressure drop across the coils.

Previous results are used to calculate the characteristics of the (fictitous) heating coil equivalent to the assembly of both (heating and cooling) coils. The results of this agregation for one of the AHU's (GP 2 or 3) are presented in Figure 11.

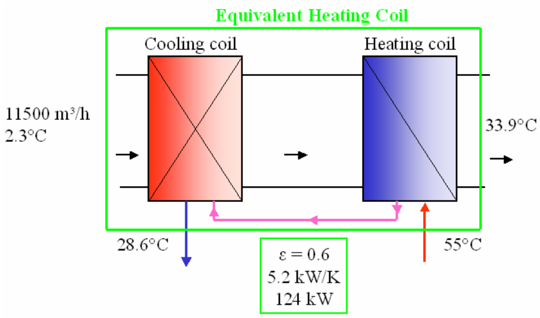

Figure 11: Global Heating Coil Model

The AU value of this fictitious coil corresponds to the sum of the two separated AU values shown in Figure 10. The exchanged power is the sum of the powers exchanged in the two coils and the global efficiency reaches 60%. The same calculation is performed for the other AHU (GP1).

Global results obtained for the different couple of coils are presented in Table 2 .

Table 2: Change Over Results

AHU	Air Flow Rate [m³/h]	Water Flow Rate [m³/h]	Equivalent Heating Coil AU value [W/K]	Equivalent Coil Eff. [%]
GP1	10000	5.2	3440	53
GP2	11500	4.026	5233	60
GP3	11500	4.026	5233	60

The results obtained by aggregation of the three (already fictitious) heating coils into an equivalent one are given in Table 3. The equivalent efficiency of this global heating coil reaches 58%.

Table 3: Global Heating Coil Characteristics

Global Air Flow Rate [m³/h]	Global Water Flow Rate [m³/h]	Global Heating Coil AU value [W/K]	Global Eff. [%]
33000	13.252	13906	58

Thermal Energy Storage Model

The thermal storage tank is modelled as if ideally stratified and adiabatic. During mid-season and summertime, the storage charging periods correspond to the heating demand periods during which the heat pump is working mainly by night. When the heat pump is not working, during peak hours, the tank could only be discharged or not used. If the tank charge is sufficient to ensure the cooling demand, the storage is discharged. If the tank is not enough charged, the complement of cooling production is ensured by the chiller/heat pump. During off-peak hours, the thermal storage can be discharged, not used or re-charged (even if the heat pump/chiller is not solicited for heat/cold production).

The storage charging rate is supposed to be proportional to the level of the storage charge. The charging power is computed supposing that the storage could always be charged in a defined time period. Of course, if the charge is sufficient, the discharging rate corresponds to the cooling demand.

This scenario is described thanks to the following equations, remembering that the thermal energy storage tank is modelled as a charging/discharging tank (Figure 12):

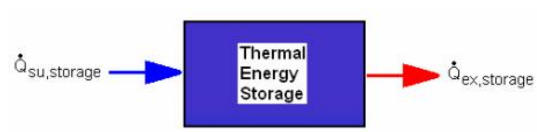

Figure 12: storage model

$$\dot{Q}_{chiller} = \dot{Q}_{cooling} - \dot{Q}_{ex,storage}$$

$$\dot{Q}_{ev} = \dot{Q}_{su,storage} + \dot{Q}_{coldsource}$$

$$\dot{Q}_{storage} = \dot{Q}_{su,storage} - \dot{Q}_{ex,storage}$$

$$Q_{storage} = \int_{\tau_1}^{\tau_2} (\dot{Q}_{storage})\, d\tau$$

where:

$\dot{Q}_{su,storage}$ - storage supply (or charging) power, [W]

$\dot{Q}_{ex,storage}$ - storage exhaust (or discharging) power, [W]

$Q_{storage}$ - charge of the storage at time t, [J].

Supply and exhaust powers are defined as functions of several parameters:
- the building cooling demand at time t.
- the storable cold power produced by the heat pump at time t .
- the charge of the storage at time t.

These functions are described hereunder:

being,

$$\dot{Q}_{su,storage} = \text{If}\,(Q_{storage}, Q_{storage,max}, \dot{Q}_{storage,charge}, 0, 0)$$

$$Q_{storage,max} = Q_{storage,max,kWh} \cdot 3.6 \times 10^{6}$$

$$\dot{Q}_{ex,storage} = \text{If}\,(\dot{Q}_{cooling}, \dot{Q}_{storage,available}, \dot{Q}_{cooling}, \dot{Q}_{cooling}, \dot{Q}_{storage,available})$$

$$\dot{Q}_{storage,available} = \frac{Q_{storage}}{3600}$$

$\dot{Q}_{storage,charge}$ - storage charging power, [W]

$Q_{storage,max}$ - maximal charge of the storage, [J]

$\dot{Q}_{cooling}$ - cooling demand of the building, at time t, [W]

$\dot{Q}_{storage,available}$ - cooling power, available at the storage tank exhaust, [W]

$Q_{storage}$ Q storage is the charge of the storage at time t, [J].

The discharge of the storage is possible only if the actual charge of the storage is sufficient to satisfy the cooling demand. If the charge is too low to satisfy a given cooling demand, the storage is not discharged and wait to be charged again.

The storage model is connected to the chiller/heat pump model using the two first equations.

The first equation is used to compute the cooling load allowed to the chiller (equal to the difference between the building cooling demand and the cooling power provided by the storage). The second formula is used to compute the heat pump evaporator power (equal to the sum of the part of the cold source available power and the storage charging power).

This concerns the use of the storage system to store cooling energy only, but the same system could be also used to store heat during no-cooling season…

GLOBAL SAVINGS

After retrofit, the gas consumption is almost cancelled; the electricity consumptions and costs (computed using actual average belgian costs and emissions) of the two zones considered are given in Table 4.

Table 4: Electricity and Gas Consumptions

	Consumptions	**Cost [€]**
New Situation		
Elec. Cons. [kWh]	1448103	136395
Nat. Gas Cons. [m³]	1702	733
Actual Situation		
Elec. Cons. [kWh]	1258382	118578
Nat. Gas Cons. [m³]	77249	33238
Money Saving / year : 14688 [€]		

The results presented in Table 4 give:

- 75547 m³ of Natural Gas spared each year, corresponding to a money saving of 32505 €;

- an over-consumption 189721 kWh of Electricity each year, corresponding to an overcost of 17817 €;

- 14688 € of energy costs, spared each year, corresponding to a payback time of about 7 years;

- 109115 kg/year of CO_2, i.e. 18 % of the present emissions (591641 kg/year);

- a global COP equal to 3.9.

These results correspond only to the two zones considered (laboratories and offices). The consumptions of the other zones of the building are not taken into account here.

CONCLUSIONS

Using a chiller in heat pump mode appears as very valuable retrofit. This is even more efficient if combined with a change-over process.

The payback time estimated in this case study is of 6 years.

ACKNOWLEDGMENT

This work is financially supported by the Walloon Region of Belgium.

REFERENCES

[1] Aparecida Silva C., Bertagnolio S., Hannay J., Lebrun J. 2007. Heat pumping and reversible air conditioning; first case study results. Heat Set conference April 2007, Chambery, France.

[2]Lemort V., Cuevas C., Lebrun J.2007. “Towards amore easy-to-use cooling coil model”, Paper submitted for ASHRAE Transaction.

[3]Dorgan C.B., Linder R. J., Dorgan C.E.1999. “Chiller Heat Recovery Application Guide”, Atlanta: American Society of Heating, Refrigeration and Air Conditioning Engineers, Inc.

[4]Roth K., Zogg R., Brodrick J. 2006. “Cool Thermal Energy Storage”, ASHRAE Journal, 48, 94 - 96.