# THEORETICAL STUDY OF COOLING TECHNOLOGIES DRIVEN BY GEOTHERMAL ENERGY FOR USE IN TERTIARY BUILDINGS IN BELGIUM

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## ABSTRACT

We investigate the possibility of using geothermal hot water in heat-driven cooling systems, for air conditioning in tertiary buildings in Belgium. Models of two possible cooling systems and of three tertiary buildings are developed in the TRNSYS environment. Results show that hot water temperature level and waste water regulations complies with desiccant cooling system, provided that the fresh air flow rate, related to the occupancy profile of the building, matches the process air flow rate of the cooling system.

## INTRODUCTION

The region of Mons in Belgium has a high potential for geothermal energy because of several particular geological characteristics. Hot water has been produced there for more than 30 years and is mainly used for greenhouse and district heating.

In the near future, a new well will be available with an estimated thermal capacity of 6 MW and an estimated maximum temperature of 72°C. The main use will be heat delivery for buildings of a new business park during the heating season (September to May in Belgium). The question arose if it can be possible to deliver hot water during summer to produce cold by using heat-driven cooling machines.

There exist three main heat-driven cooling technologies: absorption cooling, adsorption cooling and open cycle desiccant cooling (Henning, 2007). Absorption cooling needs a minimum temperature of about 80 to 85°C, which is above the geothermal well temperature. Only adsorption cooling and desiccant cooling are able to work with temperatures below or equal to 72°C.

A survey realized in September 2011 shows that cooling machines available on the market have cooling powers ranging from 8 to 2000 kW for adsorption cooling and from 25 to 200 kW for open cycle air desiccant cooling (Dumont et al., 2012).

Models of adsorption cooling machines and of desiccant cooling machines have been developed and coupled to building models in the TRNSYS environment (Trnsys, 2007). These models are used to evaluate the potentiality of using the available hot water for producing cold, to rate the performance of the two cooling technologies and to rate the thermal comfort obtained in the buildings. Three reference tertiary buildings have been used: a hotel building, an office building and a lecture room.

## SIMULATION

Dynamic models are not used for the following reasons: HVAC equipment is most of the time in quasi static state due to its short time constant; the scope of the project is to define new design rules, which are most of the time static methods; experience with heat pumps shows that annual results are not influenced by the unsteady behaviour of the equipment.

Steady-state models of cooling machines have been developed in the TRNSYS environment. Thermodynamic properties of humid air (h, $c_{PAIR}$, $c_{PW}$) are computed with the TRNSYS humid air library (Trnsys, 2007). Each model is able to compute outputs ($T_{OUT}$, $x_{OUT}$) given the inputs ($T_{IN}$, $x_{IN}$).

The building models have been developed using the TRNSYS model libraries.

### Desiccant cooling model

The desiccant cooling (DC) machine uses the Pennington configuration (Henning, 2007). This configuration does not allow recycling some part of the return air flow rate $q_R$ to the supply air flow rate $q_P$. The DC machine is composed of several devices as shown on Figure 1. Each device is modelled separately.

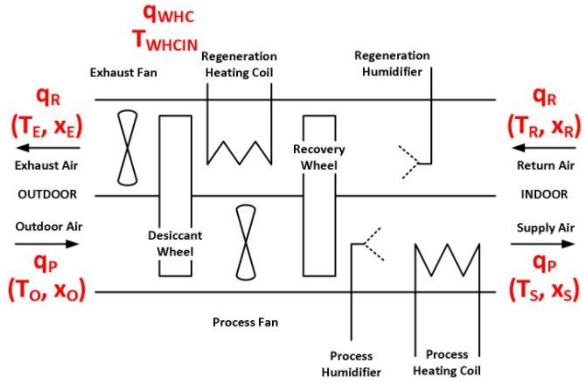


*Figure 1 Pennington configuration desiccant cooling*

- Desiccant wheel

The desiccant wheel is modelled using the F1, F2 potential functions approach of Jurinak (Jurinak, 1982). Each potential function depends on the temperature T and humidity ratio x:

$$F1 = -2865/((T+273.15)^{1.49}) + 4.344\ (x/1000)^{0.8624} \quad (1a)$$

$$F2 = ((T+273.15)^{1.49})/6360 - 1.127\ (x/1000)^{0.07969} \quad (1b)$$

These functions are equivalent to temperature for heat exchangers and allow to define wheel efficiencies.

Solving these equations in T and x for given values of F1 and F2 needs an iterative method. In order to reduce the computation time, correlations have been created to reverse equations (1a) and (1b) and to obtain an explicit equation set in T and x:

$$T = 31.0004997 + 312.343022\ A - 161.83461\ A^2 - 1630.22345\ A^3 + 3467.87795\ A^4 - 1183.00566\ A^5 - 147.450342\ (B + 0.3892398\ B^2)\ (1 + 2.91799317\ C + 0.1049758\ C^2 - 28.4192253\ C^3 + 47.1650402\ C^4) \quad (1c)$$

$$x = 1.07452577 + 28.07929\ A^* + 221.759769\ A^{*2} + 301.636266\ A^{*3} - 1418.32723\ A^{*4} + 1407.69935\ A^{*5} - 0.39154257\ (-6.23263734\ B + B^2 + 7.5508571\ B^3)\ (1 + 7.51782834\ C - 283.172352\ C^2 + 678.930538\ C^3 - 611.863923\ C^4) \quad (1d)$$

with $A = F1+0.7-0.49069028\ (F2+0.06)$
$A^* = F1+0.7-0.68853583\ (F2+0.06)$
$B = 0.2-F2$

The wheel is characterized by two efficiencies, one for each potential function:

$$\varepsilon_{F1} = (F1_{POUT}-F1_{PIN})/(F1_{RIN}-F1_{PIN}) \quad (2a)$$

$$\varepsilon_{F2} = (F2_{POUT}-F2_{PIN})/(F2_{RIN}-F2_{PIN}) \quad (2b)$$

- Recovery wheel

The recovery wheel is modelled as a counter-current heat exchanger with a constant effectiveness $\varepsilon_{RW}$:

$$\varepsilon_{RW} = (T_{ROUT}-T_{RIN})/(T_{PIN}-T_{RIN}) \text{ with } q_R \le q_P \quad (3a)$$

$$\varepsilon_{RW} = (T_{PIN}-T_{POUT})/(T_{PIN}-T_{RIN}) \text{ with } q_P \le q_R \quad (3b)$$

- Humidifier

The humidifiers are modelled in the following way. First, we assume that the humidifying process is adiabatic and that the air comes out of the device in a saturated state:

$$h_{SAT}\ (T_{SAT}, x_{SAT}) = h_{IN}\ (T_{IN}, x_{IN}) \quad (4)$$

$T_{SAT}$ and $x_{SAT}$ are found by iterative process in TRNSYS.

As a real humidifier cannot deliver saturated air, the model assumes that the output air is a mixing of the input air and of the saturated air. A bypass factor $F_{BPH}$ characterizes the humidifier:

$$q_P\ h_{OUT} = q_P\ F_{BPH}\ h_{IN} + q_P\ (1 - F_{BPH})\ h_{SAT} \quad (5a)$$

$$q_P\ x_{OUT} = q_P\ F_{BPH}\ x_{IN} + q_P\ (1 - F_{BPH})\ x_{SAT} \quad (5b)$$

- Heating coil

The regeneration heating coil is modelled as a crosscurrent heat exchanger with constant effectiveness $\varepsilon_{RHC}$:

$$\varepsilon_{RHC} = (T_{ROUT}-T_{RIN})/(T_{WHCIN}-T_{RIN}) \quad (6)$$
$$\text{with } q_R\ c_{PAIR} \le q_{WHC}\ c_{PW}$$

- Fans

The fan models are very simple: they are supposed to increase the temperature of 0.6°C.

- Desiccant cooling model solving

All device sub-models are connected in series according to Figure 1. The regeneration air mass flow rate $q_R$ and process air mass flow rate $q_P$ are assumed to be equal.

Solving the whole desiccant cooling model is performed by TRNSYS given the outdoor air ($T_O$, $x_O$) and return air ($T_R$, $x_R$) conditions and given the process air mass flow rate $q_P$, regeneration heating coil water mass flow rate $q_{WHC}$ and inlet water temperature $T_{WHCIN}$.

The model computes the supply air ($T_S$, $x_S$) and exhaust air conditions ($T_E$, $x_E$) and the regeneration heating coil outlet temperature $T_{WHCOUT}$. It also computes the DC cooling flow rate $\phi_{CDC}$, the regeneration heating coil flow rate $\phi_{HDC}$, the DC coefficient of performance $COP_{DC}$ and the DC building cooling flow rate $\phi_{CDCBUI}$:

$$\phi_{CDC} = q_P\ (h_S - h_O) \quad (7)$$

$$\phi_{HDC} = q_{WHC}\ c_{PW}\ (T_{WHCIN} - T_{WHCOUT}) \quad (8)$$

$$COP_{DC} = \phi_{CDC} / \phi_{HDC} \quad (9)$$

$$\phi_{CDCBUI} = q_P\ (h_R - h_S) \quad (10)$$

The desiccant cooling model has been fitted to one experimental point provided by the Munters Company (Desiccant wheel manufacturer). The parameters associated with this point are $\varepsilon_{F1}$=0.058, $\varepsilon_{F2}$=0.5391, $\varepsilon_{RW}$=0.785, $F_{BPH}$=0.16 and $\varepsilon_{RHC}$=0.70. The process heating coil is not used in the model: it serves as a backup heating system during the heating season.

**Adsorption cooling model**

The adsorption cooling (AC) system uses a three sources/sinks adsorption machine coupled to a cooling coil located in the supply air duct as shown in Figure 2. This configuration allows recycling some part of the return air flow rate to the supply air flow rate. The AC machine is also composed of several devices, which are modelled separately.

- Cooling coil

The cooling coil is modelled by using the bypass approach. Some fraction of the air flow rate is supposed cooled down (and dehumidified) to the average temperature ($T_{AV}$) of the water flowing through the cooling coil. As a real cooling coil cannot cool down the air to such a temperature, the model assumes that the output air is a mixing of the input air and of the cooled air. A bypass factor $F_{BPC}$ characterizes the cooling coil:

$$q_P\, h_{OUT} = q_P\, F_{BPC}\, h_{IN} + q_P\,(1 - F_{BPC})\, h_{AV} \tag{11a}$$

$$q_P\, x_{OUT} = q_P\, F_{BPC}\, x_{IN} + q_P\,(1 - F_{BPC})\, x_{AV} \tag{11b}$$

$$T_{AV} = (T_{WCCOUT} - T_{WCCIN})/2 \tag{11c}$$

Depending on the input conditions, $x_{OUT}$ is obtained by Equation (11b) if $T_{OUT}$ is above the outlet dew point and by having a saturated air if $T_{OUT}$ is below the outlet dew point. An energy balance allows computing the water outlet temperature $T_{WCCOUT}$ given the water mass flow rate $q_{WCC}$ and the inlet temperature $T_{WCCIN}$. The solution is found by iterative process in TRNSYS.

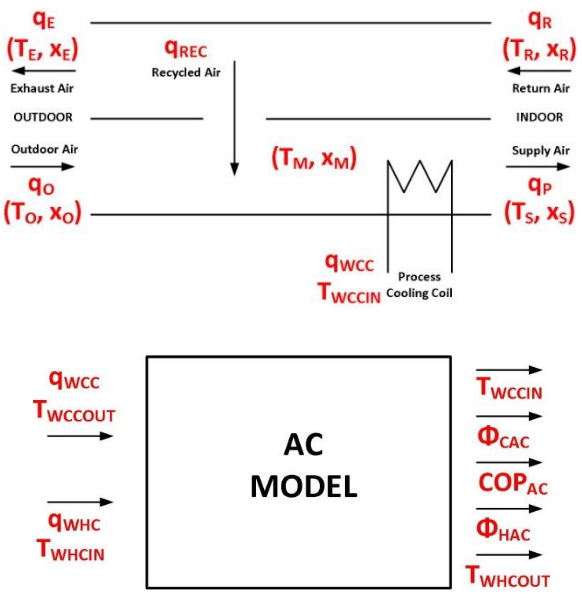


*Figure 2 Adsorption cooling machine model*

- Adsorption machine

The model of the adsorption machine is a lookup table, which contains manufacturer data (Figure 2). Given the hot source inlet temperature ($T_{WHCIN}$), the water temperature returning to the cooling coil ($T_{WCCIN}$), and the temperature coming from the cooling tower ($T_{WCTOUT}$), the table delivers the corresponding outlet temperatures ($T_{WHCOUT}$ and $T_{WCCOUT}$). Actually, the hot source inlet temperature has been fixed to 70°C for the sake of simplicity. $T_{WCTOUT}$ can reasonably be set equal to outdoor temperature $T_O$. The adsorption machine model is coupled to the cooling coil model through the water mass flow rate $q_{WCC}$ and both inlet and outlet temperatures ($T_{WCCIN}$ and $T_{WCCOUT}$).

- Fans

The fans have not been modelled in the AC system.

- Adsorption cooling model solving

The regeneration air mass flow rate $q_R$ and process mass air flow rate $q_P$ are assumed to be equal. Given the recycled air mass flow rate $q_{REC}$, mass and energy balances allow computing the mixing conditions ($T_M$, $x_M$) at the inlet of the cooling coil.

Solving the whole adsorption cooling model is performed by TRNSYS given the outdoor air ($T_O$, $x_O$) and return air ($T_R$, $x_R$) conditions and given the process air mass flow rate $q_P$, fresh air mass flow rate $q_O$, hot source water mass flow rate $q_{WHC}$ and inlet water temperature $T_{WHCIN}$.

The model computes the supply air ($T_S$, $x_S$) and exhaust air conditions ($T_E$, $x_E$) and the hot source outlet temperature $T_{WHCOUT}$. It also computes the AC cooling flow rate $\phi_{CAC}$, the hot source heating flow rate $\phi_{HAC}$, the AC coefficient of performance $COP_{AC}$ and the AC building cooling flow rate $\phi_{CACBUI}$:

$$\phi_{CAC} = q_P\,(h_S - h_M) \tag{12}$$

$$\phi_{HAC} = q_{WHC}\, c_{PW}\,(T_{WHCIN} - T_{WHCOUT}) \tag{13}$$

$$COP_{AC} = \phi_{CAC} / \phi_{HAC} \tag{14}$$

$$\phi_{CACBUI} = q_P\,(h_R - h_S) \tag{15}$$

The adsorption cooling model needs only one parameter value: the bypass factor $F_{BPC}$, which has been set to 0.25. The performance data used in the adsorption machine model comes from the SorTech Company adsorption cooling machine Model ACS08. For design, a scaling factor has been applied.

**Building models**

The three reference tertiary buildings have been modelled with Type56 of the TRNSYS library (Trnsys, 2007). Type56 is a multi-zone detailed building model able to compute the evolution of indoor temperature and humidity in each room or group of rooms of a building, called a zone. The three buildings are described in detail in (Henning, 2007).

*Table 1*
*Hotel building characteristics*

| Wall | Details | e (m) | k (kJ/(hmK)) | $c_P$ (kJ/(kg K)) | ρ (kg/m³) |
|---|---|---|---|---|---|
| Floor level 0: Area 676 m² | Covering | 0.01 | 3.600 | 0.84 | 2000 |
| | Cement | 0.03 | 3.024 | 0.84 | 2000 |
| | Insulation | 0.10 | 0.1224 | 1.47 | 30 |
| | Hollow bricks | 0.20 | 3.600 | 0.84 | 2000 |
| | Soil | 0.30 | 3.240 | 1.00 | 1500 |
| Floor levels 1 to 5: Area 676 m² | Covering | 0.05 | 5.040 | 0.84 | 2000 |
| | Insulation | 0.05 | 0.144 | 1.47 | 30 |
| | Concrete | 0.15 | 5.760 | 0.84 | 2000 |
| Walls levels 0 to 5 E and W walls: Area 44 m² S and N walls: Area 142 m² | Plaster | 0.03 | 0.216 | 0.84 | 200 |
| | Insulation | 0.05 | 0.162 | 0.84 | 180 |
| | Concrete | 0.20 | 4.680 | 0.84 | 1800 |
| | Windows: 1.78 m² (E and W) and 35.4 m² (S and N), U=1.1 W/(m² K), g=0.609 Frame: 20% of window surface area | | | | |
| Roof | Hollow bricks | 0.0 | 3.600 | 0.84 | 2000 |
| | Insulation | 0.20 | 0.1224 | 1.47 | 30 |
| | Zinc | 0.001 | 396 | 0.39 | 7000 |
| | Plaster | 0.02 | 0.250 | 0.84 | 800 |

The hotel is a four-facade, six-storey parallelepiped volume with a flat roof. Each storey has been defined as a Type56 zone. The main characteristics of the walls are described in Table 1. Ventilation rate is 0.33 $h^{-1}$ and infiltration rate is 0.5 $h^{-1}$. There are 19 persons per storey permanently all along the week, except during a 2 hours period at noon.

The office building is a four-facade, three-storey parallelepiped volume with a flat roof. Each storey has been defined as a Type56 zone. The main characteristics of the walls are described in Table 2. Ventilation rate is 0.34 $h^{-1}$ and infiltration rate is 0.2 $h^{-1}$. There are maximum 32 persons per storey.

*Table 2*
*Office building characteristics*

| Wall | Details | e (m) | k (kJ/(hmK)) | $c_P$ (kJ/(kgK)) | ρ (kg/m$^3$) |
|---|---|---|---|---|---|
| Floor level 0: Area 333 m$^2$ | Covering | 0.08 | 8.280 | 1.08 | 2400 |
| | Insulation | 0.10 | 0.1368 | 0.83 | 30 |
| | Concrete | 0.20 | 8.280 | 1.08 | 2400 |
| | Soil | 0.30 | 5.400 | 2.00 | 1500 |
| Floor levels 1 to 2: Area 333 m$^2$ | Bottom | 0.02 | 0.126 | 0.84 | 100 |
| | Air | R=0.047 K h/kJ | | | |
| | Concrete | 0.20 | 8.280 | 1.08 | 2400 |
| | Insulation | 0.05 | 0.1368 | 0.83 | 30 |
| | Air | R=0.047 K h/kJ | | | |
| | Top | 0.04 | 0.468 | 1.00 | 700 |
| Walls levels 0 to 2 | Insulation | 0.10 | 0.1368 | 0.83 | 17 |
| | Concrete | 0.25 | 8.280 | 1.08 | 2400 |
| E and W walls: Area 62 m$^2$ S and N walls: Area 84 m$^2$ | Windows: 8.04 m$^2$ (E and W) and 40.99 m$^2$ (S and N), U=1.1 W/(m$^2$ K), g=0.609 Frame: 21.5% of window surface area | | | | |
| Roof | Bottom | 0.02 | 0.126 | 0.84 | 100 |
| | Air | R=0.047 K h/kJ | | | |
| | Concrete | 0.20 | 8.280 | 1.08 | 2400 |
| | Insulation | 0.16 | 0.1152 | 0.84 | 30 |

*Table 3*
*Lecture room characteristics*

| Wall | Details | e (m) | k (kJ/(hmK)) | $c_P$ (kJ/(kgK)) | ρ (kg/m$^3$) |
|---|---|---|---|---|---|
| Ground: Area 226 m$^2$ | Covering | 0.001 | 208.8 | 0.48 | 7800 |
| | Insulation | 0.020 | 0.169 | 0.84 | 75 |
| | Concrete | 0.120 | 7.326 | 0.92 | 2100 |
| | Insulation | 0.020 | 0.169 | 0.84 | 75 |
| | Cement | 0.030 | 5.040 | 1.05 | 2200 |
| Common wall: Area 62 m$^2$ | Plaster | 0.009 | 0.760 | 1.00 | 900 |
| | Board | 0.012 | 0.610 | 1.00 | 1000 |
| | Insulation | 0.100 | 0.160 | 0.90 | 80 |
| | Board | 0.012 | 0.610 | 1.00 | 1000 |
| | Plaster | 0.009 | 0.760 | 1.00 | 900 |
| E and W walls: Area 42 m$^2$ S wall: Area 62 m$^2$ | Concrete | 0.100 | 7.326 | 0.92 | 2100 |
| | Insulation | 0.060 | 0.169 | 0.84 | 75 |
| | Plaster | 0.025 | 1.620 | 1.05 | 1300 |
| | Windows: 18 m$^2$ (E and W) and 27 m$^2$ (S and N), U=1.1 W/(m$^2$ K), g=0.609 Frame: 20% of window surface area | | | | |
| Roof Area 226 m$^2$ | Covering | 0.001 | 208.8 | 0.48 | 7800 |
| | Insulation | 0.020 | 0.169 | 0.84 | 75 |
| | Concrete | 0.120 | 7.326 | 0.92 | 2100 |
| | Insulation | 0.020 | 0.169 | 0.84 | 75 |
| | Cement | 0.030 | 5.040 | 1.05 | 2200 |

The lecture room is located on the second level (first floor) of a three-storey, parallelepiped building. The building has three-facade (the fourth facade is common with another building) and a flat roof. Only the lecture room has been modelled as a single Type56 zone. The main characteristics of the walls are described in Table 3. Ventilation rate is 6.17 $h^{-1}$ and infiltration rate is 0.2 $h^{-1}$. There are maximum 100 persons in the room. Details about the occupation schedules and internal gains for the three buildings are found elsewhere (Henning, 2007).

## Cooling system design

Flow rates are obtained during the design of the cooling systems, i.e. when adapting the system to the cooling loads of the buildings.

The process flow rate $q_P$ is obtained when designing the cooling systems. Each system size is based on sensible cooling loads $\phi_{CSENSBUI}$ of the building, calculated in TRNSYS with no ventilation (because ventilation is part of the cooling system), for maximum outdoor temperature in Uccle (Belgium) and indoor temperature $T_R$= 23°C. The sensible cooling loads used are day-average values.

The supply air temperature $T_S$ is determined according to the height of the room (Table 4).

*Table 4*
*Supply air temperature $T_S$*

| Room height (m) | $\|T_S - T_R\|$ (°C) |
|---|---|
| 2.4 | 6 |
| 2.7 | 8 |
| 3.0 | 10 |
| 3.5 | 12 |

Equation 16 allows to calculate the needed process air flow rate $q_P$:

$$\phi_{CSENSBUI} = q_P\, c_{PAIR}\, (T_R - T_S) \qquad (16)$$

In the DC cooling model, regeneration heating coil water mass flow rate $q_{WHC}$=$q_P$/4.905, according to the Munters company design data, which gives a hot water temperature drop à 20°C. This ratio has been varied to have a temperature drop in the range 20-25°C, in order to comply with regulations about hot water temperature release in the sewage system.

In the AC cooling model, hot source water mass flow rate $q_{WHC}$ is a multiple of the cooling coil mass flow rate $q_{WCC}$, according to the manufacturer data. This flow rate is obtained with Equation 17:

$$\phi_{CAC} = q_P\, (h_S - h_M) = q_{WCC}\, c_{PW}\, (T_{WCCOUT} - T_{WCCIN}) \qquad (17)$$

This equation needs the water temperature regime used in the cooling coil (usually 10-15°C), which has also been varied.

The fresh air flow rate $q_O$ is fixed according to fresh air regulations, depending on the occupation schedule of the building. The values are given above for each building. In most cases, the process air flow rate $q_P$ is larger than the ventilation air flow rate $q_O$.

### Simulation conditions

In order to define completely the simulation conditions, we need numerical values for inlet air humidity and temperature, and hot water temperature.

For both cooling models, $T_O$ and $x_O$ vary along the year according to weather data coming from a TRNSYS library (Trnsys, 2006). We used weather data of Uccle (Belgium). $T_R$ and $x_R$ are computed by the building model through energy balance, according to the building occupancy. The hot water inlet temperature used was in the range $T_{WHCIN}$=65-70°C, depending on the simulation.

The indoor temperature has been controlled, using a controller model available in the TRNSYS libraries: $T_R$=20±0.5°C during the heating season and $T_R$=23±0.5°C during the cooling season, when people are present. Indoor humidity is not controlled.

In order to save energy, the cooling system (DC or AC) was used when $T_O$ is above 18°C. When $T_O$ is below 18°C, free cooling is used, i.e., outdoor air was directly supplied to the building with the design mass air flow rate $q_P$. This free cooling temperature limit FCL has also been varied.

Simulations have been performed for a whole year with a time step of 1 minute.

## RESULTS

### Building heating and cooling loads

For simulation conditions given above, the heating and cooling loads of the three buildings have been computed. In order to obtain latent loads, a range of humidity ratio (40%-55%) has been defined. A humidifying or dehumidifying process is switched on to keep humidity in this range.

Figure 3 shows the maximum sensible cooling loads needed for the cooling system design: $\phi_{CSENSBUI}$=11.8 kW (Hotel, 2$^{nd}$ floor), 22.3 kW (Office building) and 8.2 kW (Lecture room). The load profiles are very different. The hotel building has cooling loads equal to heating loads, and the latent loads are quite high. The office building also has equal cooling and heating loads but the latent loads are very small. In the lecture room, the cooling loads are much higher than the heating loads, due to the large number of persons present. The latent loads are also small.

### Results analysis with DC – reference case

Indoor temperature and relative humidity (RH) for the hottest day of the year (July 29$^{th}$) when using the DC cooling system are presented in Figure 4. Hot water temperature regime (HWR) is 70-50°C, free cooling temperature limit (FCL) at 18°C.

Figure 4 shows that temperature control is correct, noting that control is effective when people are present in the building. As humidity is not controlled, it can cause comfort problems because DC supplies air with high humidity. But Figure 4 shows that during periods when the temperature is controlled, humidity is nearly always below 70%, except in the lecture room. In order to quantify comfort, Predicted Mean Vote (PMV) index has been calculated: it is always lower than 0.4 when indoor temperature is lower than 23°C and RH lower than 70%. We consider these values as the maximum values for indoor comfort.

### Results analysis with DC – parametric study

Annual results for reference case and three other cases are presented in Table 5. The reference case (HWR=70-50°C and FCL=18°C) has values corresponding to the "standard" parameters fitted to the Munters data. The Var1 case (HWR=70-50°C and FCL=20°C) tries to decrease the use of DC by using free cooling more often. Var2 case (HWR=65-45 and FCL=18°C) tries to use a lower hot water temperature. Finally, Var3 case (HWR=65-40 and FCL=18°C) modifies the hot water flow rate trying to release as cold as possible hot water to the sewage system.

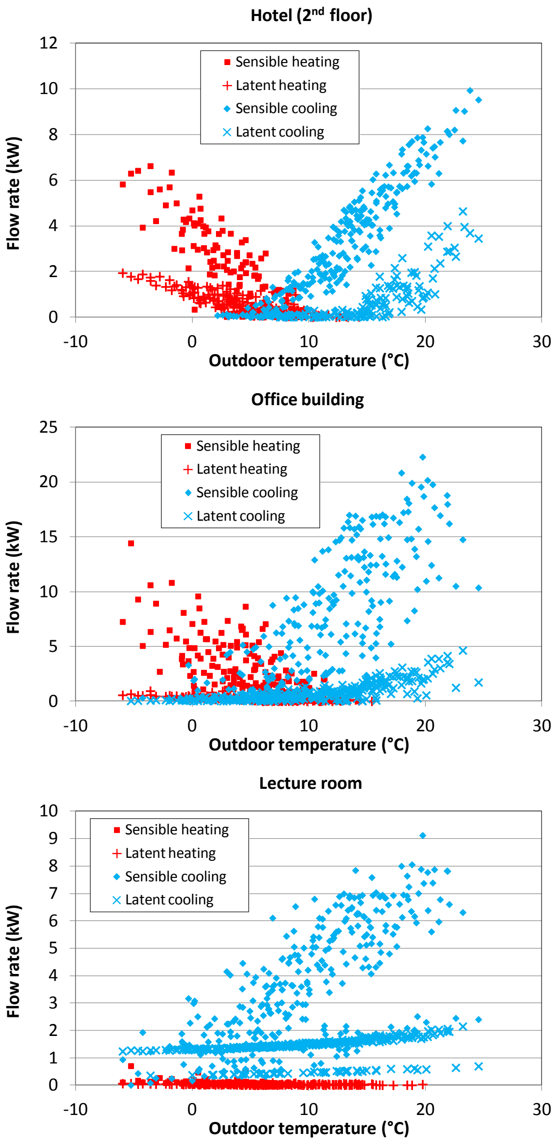


*Figure 3 Heating and cooling loads (day-average)*

Reference case for the Hotel has a COP=0.5 and uses free cooling and DC nearly the same number of hours. The number of hours above 23°C and the number of °C.h above 23°C are small due to the correct temperature control. The number of hours above 70% RH is higher but not too high to make the climate uncomfortable. Cases Var1 to Var3 does not show big differences with the reference case, except Var2. That means that low hot water temperature can be used without losing performance. Var2 case has a higher COP (0.56) and uses more often free cooling, using therefore less energy to cool the building but with less comfort. It is worth to note that the process air flow rate is 11.8 times higher than the fresh air flow rate.

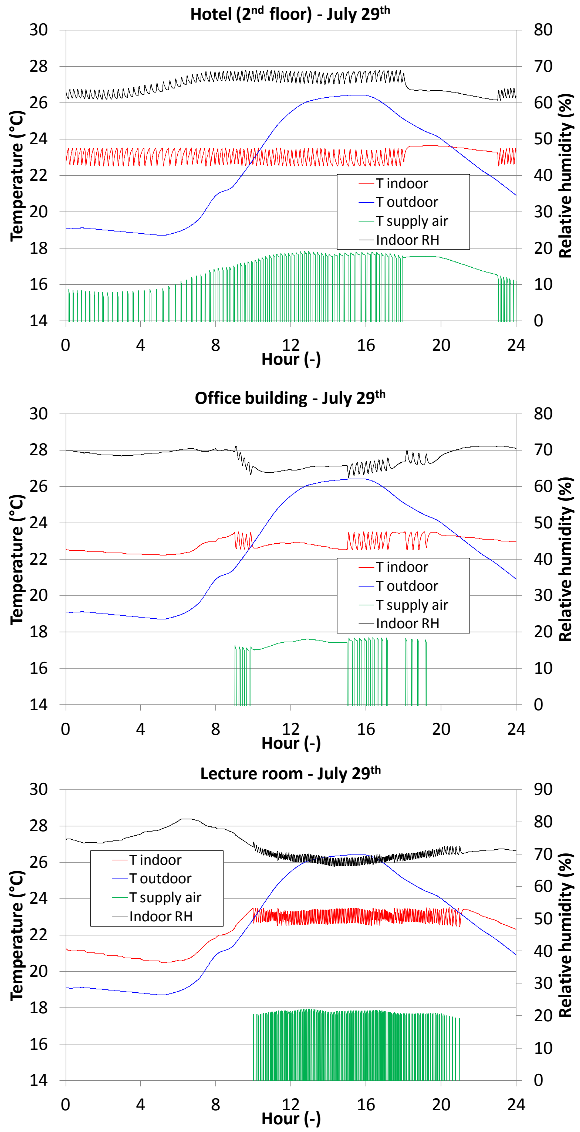


*Figure 4 Temperatures and RH for July 29th*

We can draw nearly the same conclusions for the office building (COP=0.44) as for the hotel, except for temperature control, which is not so good, certainly due to a small under-sizing of the cooling system. Humidity seems to cause fewer problems. Here, the process air flow rate is 9.55 times higher than the fresh air flow rate.

The lecture room reference case (COP=0.45) is intermediate between the hotel and office building reference case. Here the process air flow rate is nearly the same as the fresh air flow rate, due to high occupancy of the room.

**Results analysis with AC – reference case**

Indoor temperature and humidity for the hottest day of the year (July 29th) when using the AC cooling system are presented in Figure 5. The cooling coil is designed for water temperature regime (CWR) 15-18°C.

*Table 5*
*Annual results for DC case studies*

| Results | Reference | Var1 | Var2 | Var3 |
|---|---|---|---|---|
| Hotel (2nd floor) | | | | |
| $q_P/q_{FA}$ (-) | 11.8 | 11.8 | 11.8 | 11.8 |
| $Q_{CSENSBUI}$ (kWh) | 13834 | 13891 | 13831 | 13829 |
| $Q_C$ (kWh) | 11019 | 8570 | 10301 | 10058 |
| $Q_H$ (kWh) | 21931 | 15224 | 20151 | 19594 |
| COP (-) | 0.50 | 0.56 | 0.51 | 0.51 |
| $N_{FC}$ (h) | 330 | 594 | 331 | 331 |
| $N_{DC}$ (h) | 460 | 327 | 471 | 475 |
| $N_{23^\circ C}$ (h) | 41 | 120 | 42 | 43 |
| $I_{23^\circ C}$ (°C.h) | 17 | 43 | 21 | 22 |
| $N_{70\%}$ (h) | 44 | 81 | 60 | 67 |
| $I_{70\%}$ (%.h) | 133 | 285 | 195 | 217 |
| Office building | | | | |
| $q_P/q_{FA}$ (-) | 9.55 | 9.55 | 9.55 | 9.55 |
| $Q_{CSENSBUI}$ (kWh) | 38294 | 37906 | 38267 | 37063 |
| $Q_C$ (kWh) | 19123 | 13963 | 17501 | 17254 |
| $Q_H$ (kWh) | 43234 | 27592 | 39285 | 37852 |
| COP (-) | 0.44 | 0.51 | 0.45 | 0.46 |
| $N_{FC}$ (h) | 579 | 841 | 580 | 554 |
| $N_{DC}$ (h) | 493 | 325 | 498 | 500 |
| $N_{23^\circ C}$ (h) | 239 | 437 | 245 | 246 |
| $I_{23^\circ C}$ (°C.h) | 150 | 340 | 163 | 160 |
| $N_{70\%}$ (h) | 9 | 16 | 12 | 23 |
| $I_{70\%}$ (%.h) | 19 | 45 | 30 | 49 |
| Lecture room | | | | |
| $q_P/q_{FA}$ (-) | 1.22 | 1.22 | 1.22 | 1.22 |
| $Q_{CSENSBUI}$ (kWh) | 6339 | 6785 | 6352 | 5457 |
| $Q_C$ (kWh) | 5785 | 4708 | 5356 | 4952 |
| $Q_H$ (kWh) | 12796 | 9271 | 11789 | 10297 |
| COP (-) | 0.45 | 0.51 | 0.45 | 0.48 |
| $N_{FC}$ (h) | 138 | 363 | 138 | 112 |
| $N_{DC}$ (h) | 396 | 294 | 406 | 372 |
| $N_{23^\circ C}$ (h) | 103 | 298 | 112 | 85 |
| $I_{23^\circ C}$ (°C.h) | 57 | 217 | 70 | 53 |
| $N_{70\%}$ (h) | 27 | 35 | 35 | 60 |
| $I_{70\%}$ (%.h) | 62 | 91 | 89 | 151 |

Figure 5 shows that temperature control is correct. In this case also, humidity is not controlled but the use of a cooling coil can sometimes dehumidify the supply air, causing a lower RH inside the building.

**Results analysis with AC – parametric study**

Annual results for reference case and two other cases are presented in Table 6. The reference case (CWR=15-18°C) is designed to work in the same conditions as the reference case of the DC. Var1 case (CWR=15-18°C and dehumidifying above 70% RH, no free cooling) is used to investigate the effect of dehumidification on the performance of the machine.

Var2 case (CWR=10-15°C) is used trying to decrease the process air flow rate by decreasing the supply air temperature. Due to design problems, Var2 case was not simulated for the lecture room. For this building, CWR=13-16°C for reference and Var1 cases.

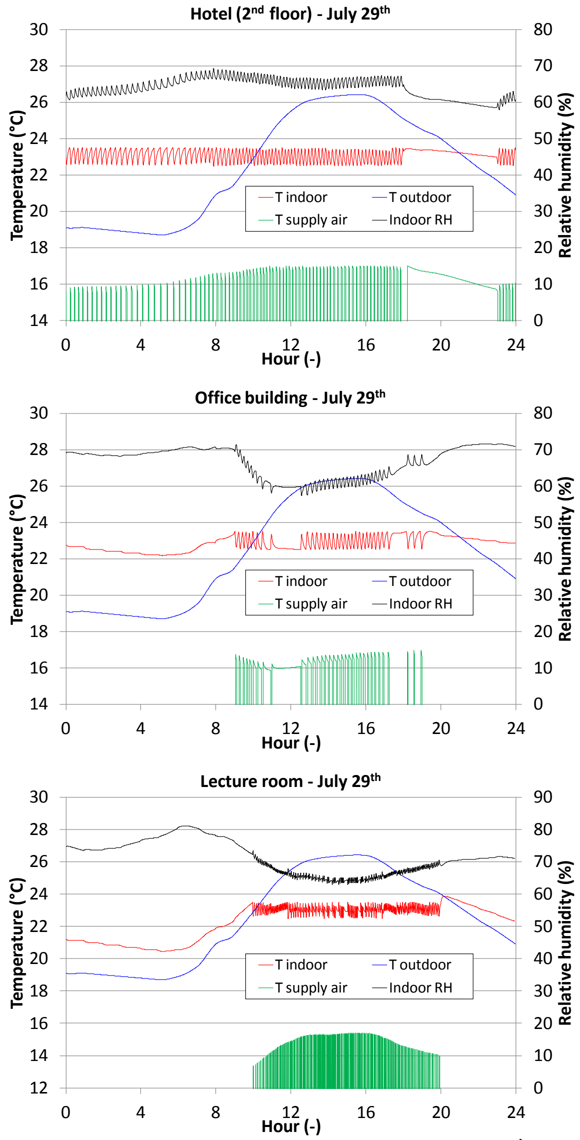


*Figure 5 Temperatures and HR for July 29th*

Reference case for the Hotel has a COP=0.82 and uses free cooling and DC nearly the same number of hours. The number of hours above 23°C and the number of °C.h are small. For RH, the comfort is lower but not dramatic. Var1 case sees its energy consumption increase (1.5 times) due to air dehumidification but keeps the same COP. Var2 case, by using lower supply air temperature uses a little more energy than the reference case due to unavoidable dehumidification but with a worse COP (0.59). Var1 and Var2 cases are effective for RH control compared to reference case but Var2 is less effective in temperature control. The same conclusions apply for the two other buildings.

**Comparison DC and AC cooling systems**

The most successful DC systems are the Var3 case for which the geothermal hot water regime (65-40°C) is favourable while keeping the same COP as in the other cases. For AC systems, the reference case is the best due to lower energy consumption than the two other cases.

*Table 6*
*Annual results for AC case studies*

| Results | Reference | Var1 | Var2 |
|---|---|---|---|
| | Hotel (2nd floor) | | |
| $q_P/q_{FA}$ (-) | 11.75 | 11.75 | 7.35 |
| $Q_{CSENSBUI}$ (kWh) | 13553 | 13787 | 13769 |
| $Q_C$ (kWh) | 9368 | 15812 | 10947 |
| $Q_H$ (kWh) | 11493 | 19286 | 18562 |
| COP (-) | 0.82 | 0.82 | 0.59 |
| $N_{FC}$ (h) | 338 | - | 512 |
| $N_{AC}$ (h) | 452 | 751 | 503 |
| $N_{23°C}$ (h) | 37 | 36 | 191 |
| $I_{23°C}$ (°C.h) | 6 | 23 | 55 |
| $N_{70\%}$ (h) | 42 | 4 | 12 |
| $I_{70\%}$ (%.h) | 96 | 36 | 13 |
| | Office building | | |
| $q_P/q_{FA}$ (-) | 9.65 | 9.65 | 6.02 |
| $Q_{CSENSBUI}$ (kWh) | 13135 | 13092 | 13183 |
| $Q_C$ (kWh) | 19948 | 37560 | 22484 |
| $Q_H$ (kWh) | 25159 | 46589 | 33877 |
| COP (-) | 0.79 | 0.81 | 0.66 |
| $N_{FC}$ (h) | 530 | - | 767 |
| $N_{AC}$ (h) | 499 | 1335 | 542 |
| $N_{23°C}$ (h) | 264 | 172 | 628 |
| $I_{23°C}$ (°C.h) | 173 | 112 | 590 |
| $N_{70\%}$ (h) | 15 | 6 | 10 |
| $I_{70\%}$ (%.h) | 87 | 19 | 61 |
| | Lecture room | | |
| $q_P/q_{FA}$ (-) | 1.23 | 1.23 | / |
| $Q_{CSENSBUI}$ (kWh) | 4892 | 5214 | / |
| $Q_C$ (kWh) | 6640 | 8635 | / |
| $Q_H$ (kWh) | 11686 | 15012 | / |
| COP (-) | 0.56 | 0.57 | / |
| $N_{FC}$ (h) | 74 | - | / |
| $N_{AC}$ (h) | 227 | 290 | / |
| $N_{23°C}$ (h) | 16 | 2 | / |
| $I_{23°C}$ (°C.h) | 3 | 0 | / |
| $N_{70\%}$ (h) | 142 | 25 | / |
| $I_{70\%}$ (%.h) | 604 | 53 | / |

Comparing the DC system with the AC system, the highest performance is the AC system: higher COP (0.82 compared to 0.51 for the Hotel, but 0.56 compared to 0.48 for the lecture room). However, the DC system can work with a temperature regime more adapted to geothermal energy. AC systems cannot work with a hot water temperature drop of more than 10°C. Knowing that the geothermal energy is a “free” energy, performance of the system seems less important than the matching of the system to the hot water source temperature regime. We can conclude that DC is certainly more adapted to the use of geothermal energy, when hot water has to be disposed in the sewage system. An important remark for both cooling systems is that they use a process air flow rate much higher than the fresh air flow rate, which can be a drawback compared to standard compression cooling systems. Only the lecture room seems adapted to the correct process air flow rate.

## CONCLUSION

Simulations performed for three reference buildings with two different heat-driven cooling systems (DC and AC) have shown that DC is the most adapted to cooling loads for the Belgian climate, if the process air flow rate is the same as the fresh air flow rate, which is the case for buildings with a high occupancy rate. It should be the case for lecture rooms, schools and supermarkets.

## NOMENCLATURE

$c_P$ = wall layer heat capacity [kJ/(kg.K)]
$c_{PAIR}$ = air heat capacity [J/(kg.K)]
$c_{PW}$ = water heat capacity [J/(kg.K)]
$COP_{AC}$ = COP of AC system [-]
$COP_{DC}$ = COP of DC system[-]
$CWR$ = cold water temperature regime [°C]
$e$ = thichness of the wall layer [m]
$F_{BPC}$ = by-pass factor of the cooling coil [-]
$F_{BPH}$ = by-pass factor of the humidifier [-]
$FCL$ = free cooling temperature limit [°C]
$F1$ = potential function F1 for the desiccant wheel
$F2$ = potential function F2 for the desiccant wheel
$g$ = window solar factor [-]
$h$ = specific enthalpy [J/kg]
$HWR$ = hot water temperature regime [°C]
$I_{23°C}$ = annual number of °Chours above 23°C [°C.h]
$I_{70\%}$ = annual number of %hours above 70% RH [%.h]
$k$ = thermal conductivity o the wall layer [kJ/(h.m.K)]
$N_{AC}$ = annual number of hours AC [h]
$N_{DC}$ = annual number of hours DC [h]
$N_{FC}$ = annual number of hours free cooling [h]
$N_{23°C}$ = annual number of hours above 23°C [h]
$N_{70\%}$ = annual number of hours above 70% RH [h]
$PMV$ = predicted mean vote index [-]
$q_E$ = exhaust air mass flow rate [kg/s]
$q_{FA}$ = fresh air mass flow rate [kg/s]
$q_O$ = outdoor air mass flow rate [kg/s]
$q_P$ = supply air mass flow rate [kg/s]
$q_R$ = return air mass flow rate [kg/s]
$q_{REC}$ = recycled air mass flow rate [kg/s]
$q_{WCC}$ = cooling coil water mass flow rate [kg/s]
$q_{WHC}$ = heating coil water mass flow rate [kg/s]
$Q_C$ = annual cold of the cooling system [kWh]
$Q_H$ = annual heat of the cooling system [kWh]
$T$ = temperature [°C]
$T_E$ = exhaust air temperature [°C]
$T_{IN}$ = inlet temperature [°C]
$T_M$ = mixed air temperature [°C]
$T_O$ = outdoor air temperature [°C]
$T_{OUT}$ = outlet temperature [°C]
$T_P$ = process air temperature [°C]
$T_R$ = return air temperature [°C]
$T_S$ = supply air temperature [°C]
$T_{WCC}$ = cooling coil water temperature [°C]
$T_{WCT}$ = cooling tower water temperature [°C]
$T_{WHC}$ = heating coil water temperature [°C]
$U$ = overall heat transfer coefficient [W/(m$^2$.K)]
$x$ = humidity ratio [g water/kg dry air]
$x_E$ = exhaust air humidity ratio [g water/kg dry air]
$x_{IN}$ = inlet humidity ratio [g water/kg dry air]
$x_M$ = mixed air humidity ratio [g water/kg dry air]
$x_O$ = outdoor air humidity ratio [g water/kg dry air]
$x_{OUT}$ = outlet humidity ratio [g water/kg dry air]
$x_P$ = processair humidity ratio [g water/kg dry air]
$x_R$ = return air humidity ratio [g water/kg dry air]
$x_S$ = supply air humidity ratio [g water/kg dry air]
$\varepsilon_{F1}$ = effectiveness 1 of the desiccant wheel [-]
$\varepsilon_{F2}$ = effectiveness 2 of the desiccant wheel [-]
$\varepsilon_{RHC}$ = effectiveness of the regeneration heating coil [-]
$\varepsilon_{RW}$ = effectiveness of the recovery wheel [-]
$\phi_{CAC}$ = AC cooling flow rate [W]
$\phi_{CACBUI}$ = DC cooling flow rate delivered to the building [W]
$\phi_{HAC}$ = DC heating flow rate [W]
$\phi_{CDC}$ = DC cooling flow rate [W]
$\phi_{CDCBUI}$ = DC cooling flow rate delivered to the building [W]
$\phi_{CSENSBUI}$ = sensible cooling needs of the building [W]
$\phi_{HDC}$ = DC heating flow rate [W]
$\rho$ = wall layer density [kg/m$^3$]

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