CONNECTION WITH HEATING OR HVAC SYSTEM

The building lumped model validated in chapter 5, and the ventilation model described in chapter 6, can both be connected to a traditional heating system including radiators or heating floors, or to an HVAC system. This chapter is aiming to present some applications resulting from the coupling of the building model with its heating or HVAC system.

9.1. Connection with heating system

A two zones house model can be implemented with two hot water emission systems (radiators or heating floor), those being connected to four types of plants: classical boiler, condensing boiler, air/water heat pump, brine water/water heat pump with outdoor horizontal ground water heat exchanger. The model can also be provided with room air conditioners for heating and/or cooling.

One particular application can concern such a high mass emission system as heating floor, connected to a brine water/water heat pump with horizontal ground water heat exchanger.

9.1.1 Heating floor model

The heating floor can be considered as a semi isothermal water-floor exchanger with isothermal conditions on the floor capacity node which is provided with a thermal capacity corresponding to the whole heating floor mass located above the insulation layer (fig. 9.1, left). A resistance can be added at the bottom of the floor in order to account for losses to the room underneath.

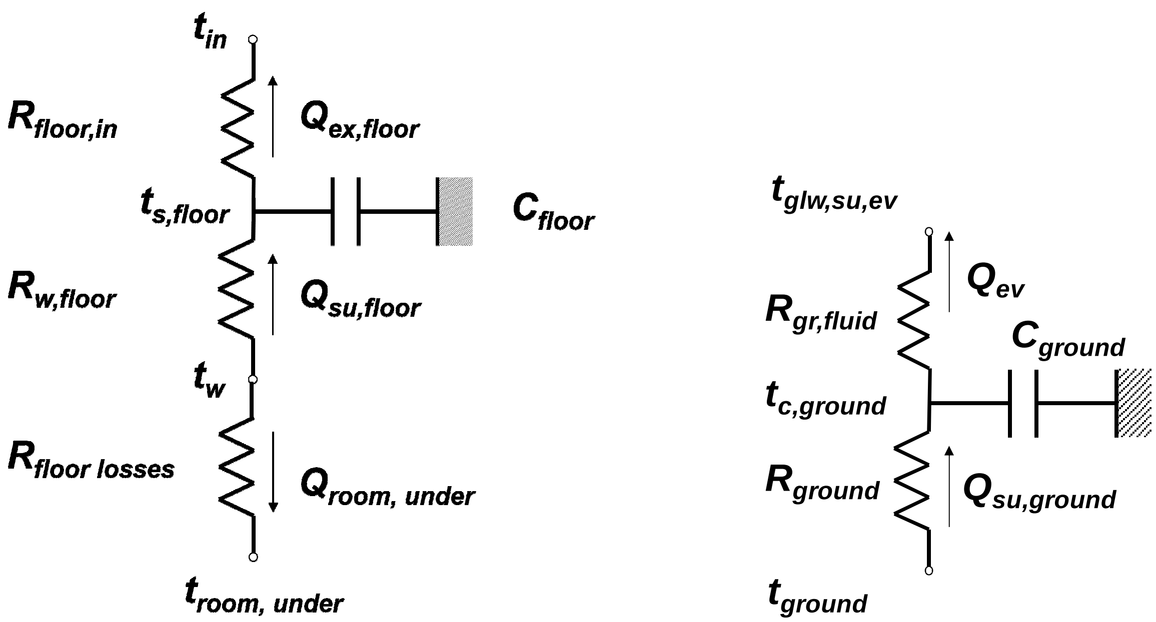

Fig. 9.1: Heating floor model (left) and ground model (right).

$$\dot{Q}_{ex,floor} = \dot{Q}_{su,floor} - \dot{Q}_{floor,storage} - \dot{Q}_{loss,room,under,z1} + \dot{Q}_{gain,room,up,z1} \quad (9.1)$$

$$\dot{Q}_{ex,floor} = \frac{t_{s,floor} - t_{in}}{R_{floor,in}}$$

$$\dot{Q}_{su,floor} = \varepsilon_{floor} \cdot \dot{C}_{w,floor} \cdot (t_{w,su,floor} - t_{s,floor})$$

$$\varepsilon_{floor} = 1 - \mathbf{exp}(-NTU_{floor})$$

$$NTU_{floor} = \frac{1}{\dot{C}_{w,floor} \cdot R_{w,floor}}$$

$$\dot{Q}_{su,floor} = \dot{C}_{w,floor} \cdot (t_{w,su,floor} - t_{w,ex,floor})$$

$$\dot{Q}_{floor,storage} = C_{floor} \cdot dt_{s,floor\backslash dtau}$$

$$t_{s,floor} = t_{s,floor,init} + \int_{\tau_{initial}}^{\tau_{final}} (dt_{s,floor\backslash dtau}) \, d\tau$$

$$\dot{Q}_{loss,room,under,z1} = \frac{\frac{t_{w,su,floor} + t_{w,ex,floor}}{2} - t_{room,under}}{R_{floor,losses}}$$

$$\dot{Q}_{gain,room,up,z1} = \dot{Q}_{loss,room,under,z2}$$

Heating floor parameters can be deduced from the floor sizing so that the heat emission is sufficient to compensate the building losses in nominal steady-state conditions, without exceeding a maximum floor surface temperature equal to *29°C*. Nominal conditions can be defined as $t_{out} = -10°C / t_{in} = 20°C$ to compute building losses. The heat floor nominal working conditions can be $t_{su,floor} = 40°C / t_{ex,floor} = 32 / t_{in} = 20°C$. The nominal heating floor water flow can be deduced from those nominal conditions.

If the floor area needed to compensate the zone nominal heat losses exceeds the zone available floor area, a back-up heat power can be added through an electric radiator during the occupancy hours.

The simulation of the heating floor behavior requires a control strategy (fig. 9.2). The set point temperature of the water supplied to the two zones can be imposed through a feed-forward control related to the outdoor temperature, and corrected by a feedback control on zone 1:

$$t_{w,su,floor,set} = t_{w,su,floor,nom} + C_{ff} \cdot (t_{out} - t_{out,nom}) + C_{fb,z1} \cdot (t_{set,z1} - t_{in,z1})$$

$$t_{w,su,floor,nom} = 40 \quad [°C] \qquad C_{fb,z1} = 5 \tag{9.2}$$

$$C_{ff} = \frac{-2}{3}$$

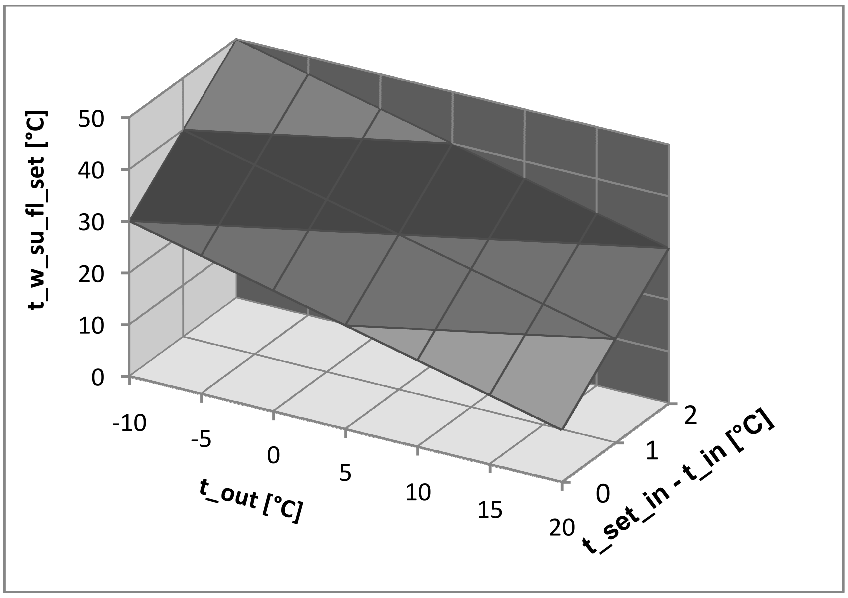

Fig. 9.2 Control low for heating floor emission system.

A proportional feedback control can be applied to maintain this set point water temperature (fig. 9.3):

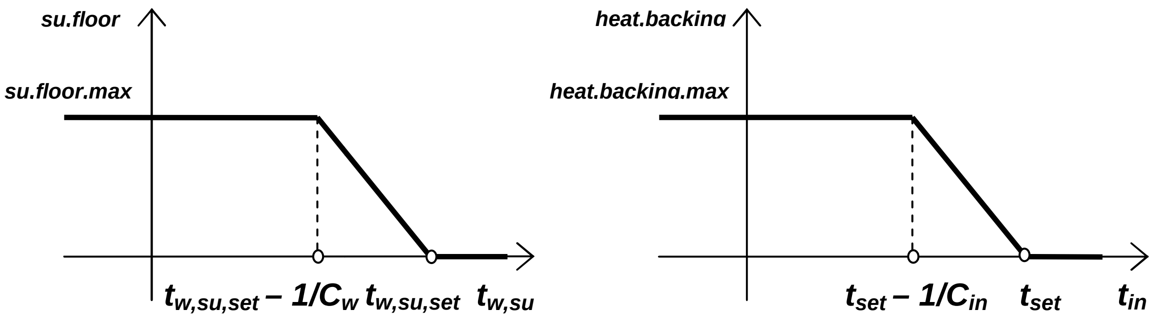

Fig. 9.3 Proportional feedback control on floor water supply temperature for the heating floor (left), and on zone indoor temperature for the backing heating system (right).

$$\dot{Q}_{su,floor} = X_{heating,floor,control} \cdot \dot{Q}_{su,floor,max}$$

$$X_{heating,floor,control} = \mathbf{Max}\ (\ \mathbf{Min}\ (\ C_{w,heating,floor,control} \cdot (\ t_{w,su,floor,set} - t_{w,su,floor}),\ 1\ ),\ 0\ )$$

$$C_{w,heating,floor,control} = 2 \quad [K^{-1}]$$

9.1.2 Brine water - water heat pump

The compressor electricity consumption can be evaluated through a set of order 2 polynomial regressions laws, giving the heat pump COP and condenser heat power as function of:

- The temperature of the water supplied to the evaporator
- The temperature of the water leaving the condenser
- The heat pump nominal heat power and the COP related to EN255 standard conditions (0°C at the evaporator and 35°C at the condenser).

The coefficients of those regression laws can be adjusted through simulations on a detailed model of heat pump (ref. [11]).

A simple dynamic model of the ground can be used to compute the temperature of the brine water supplied to the evaporator (fig. 9.1). A ground temperature node is submitted to a damped sinusoidal temperature signal, and connected to the heat pump evaporator brine supply temperature, through to resistances and a ground heat capacity:

$$\dot{Q}_{ev} = \dot{Q}_{su,ground} - \dot{Q}_{ground,storage} \tag{9.3}$$

$$\dot{Q}_{ev} = \frac{t_{c,ground} - t_{glw,su,ev}}{R_{ground,fluid}} \qquad \dot{Q}_{su,ground} = \frac{t_{ground} - t_{c,ground}}{R_{ground}}$$

$$\dot{Q}_{ground,storage} = C_{ground} \cdot dt_{c,ground\backslash dtau}$$

$$t_{c,ground} = t_{c,ground,init} + \int_{\tau_{initial}}^{\tau_{final}} (dt_{c,ground\backslash dtau}) \, d\tau$$

$R_{ground,fluid} = 0.001$ [K/W] $\qquad R_{ground} = 0.001$ [K/W]

$C_{ground} = 1 \times 10^9$ [J/K]

$$t_{ground} = t_{ground,average} + \Delta t_{ground} \cdot \cos(\omega \cdot \tau + \pi)$$

$$\omega = 2 \cdot \frac{\pi}{365 \cdot 24 \cdot 3600}$$

$t_{ground,average} = 10$ [°C] $\qquad \Delta t_{ground} = 5$ [°C]

9.1.3 Application

The building dynamic model, connected to the heating floor and to a brine/ water heat pump allows us to answer the following question: *is it worth performing a night set back with such a massive heating system?*

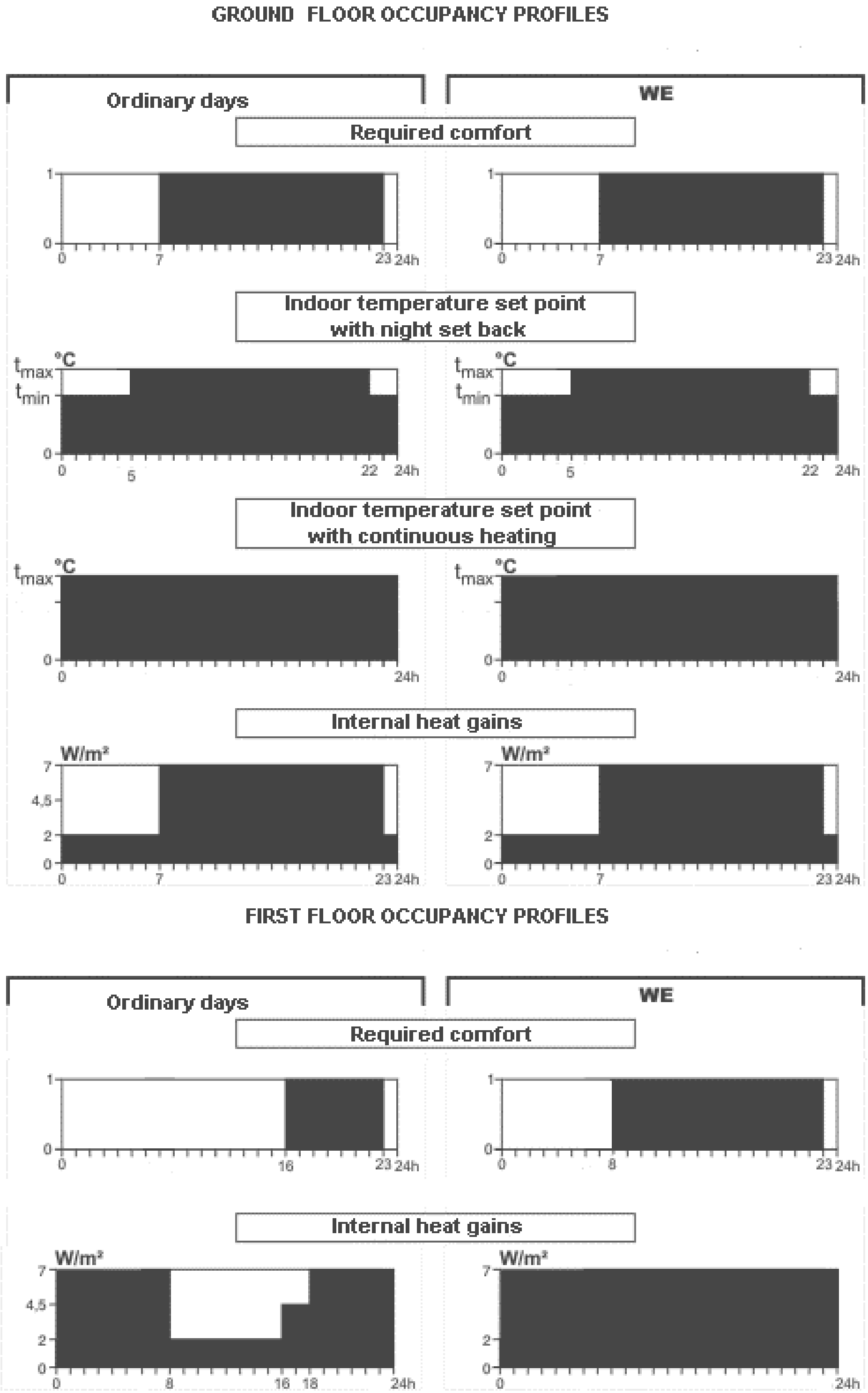

Fig. 9.4 Occupancy profiles for ground floor and first floor of Esneux house. Indoor temperature set points for both simulations (continuous heating and night set back) are represented for ground floor and applied to both floors.

A simulation is performed on a traditional house (annex 2: Esneux house). The house is subdivided into two zones corresponding to the ground floor and first floor. A first simulation maintains the heating floor indoor temperature set point at a constant temperature level, while a second simulation performs a night temperature set back.

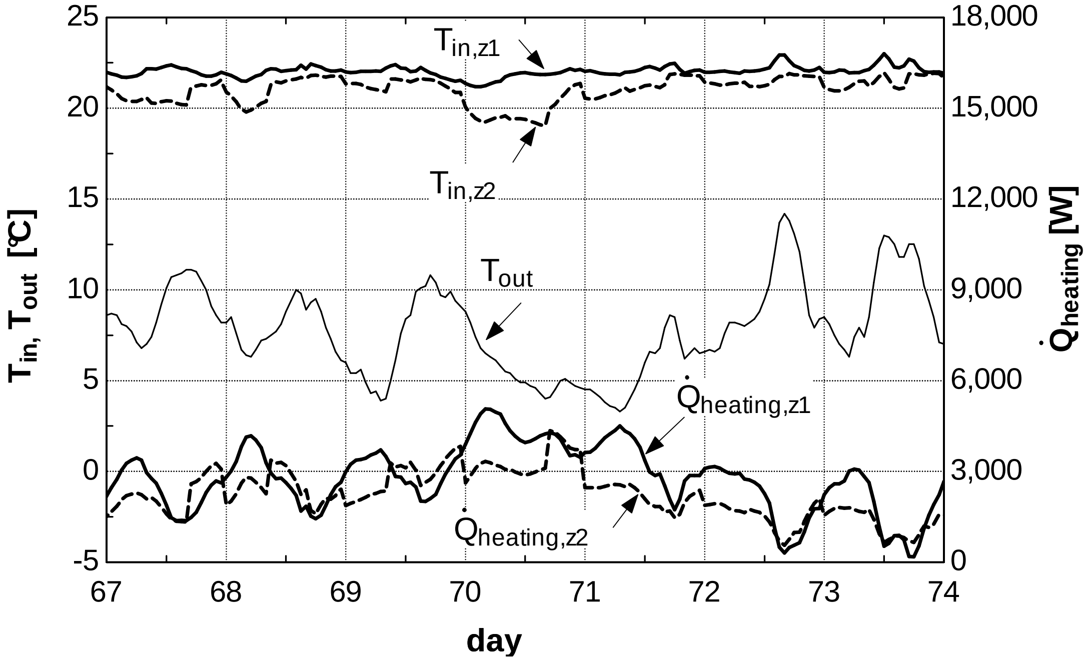

Fig. 9.5a Indoor temperature (upper curves) and heating power (lower curves) related to ground floor (full line), and first floor (dotted line), for a continuous heating strategy. Middle curve shows the outdoor temperature.

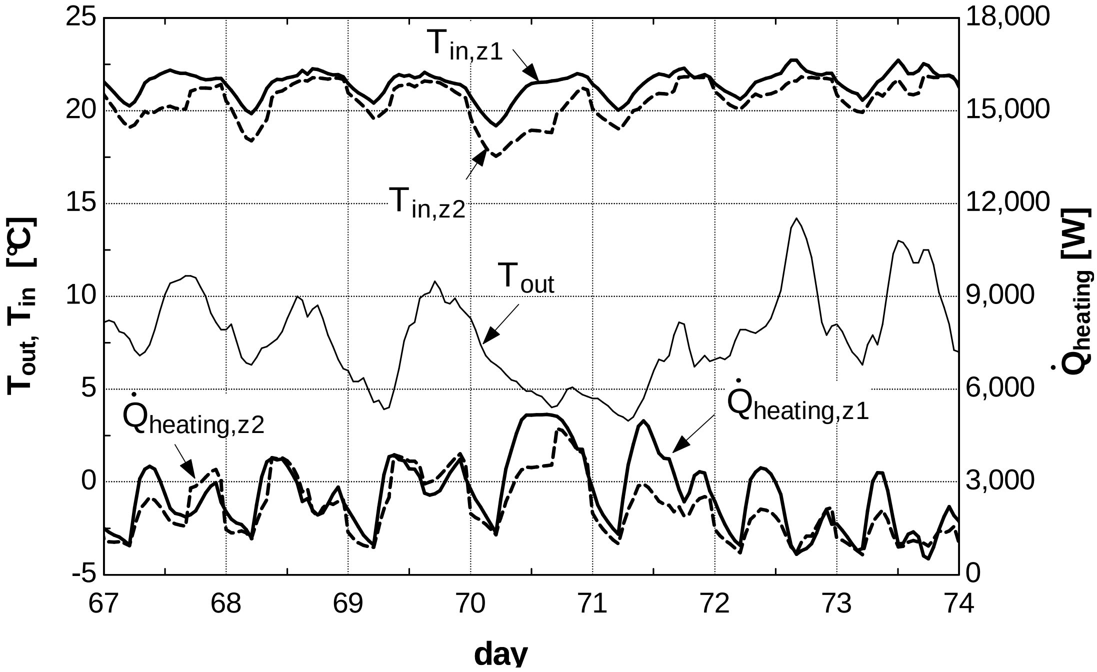

Fig. 9.5b Indoor temperature (upper curves) and heating power (lower curves) related to ground floor (full line), and first floor (dotted line), for a night set back strategy. Middle curve shows the outdoor temperature.

Fig. 9.4 shows the occupancy profiles related to each zone. The indoor temperature set points corresponding to both simulations are represented for the ground floor only, as they are the same for the first floor. The maximum indoor temperature set point equals 22°C, while the minimum is 10°C when night set back control is applied.

Fig. 9.5 shows the evolution of indoor temperatures and heating power in the two zones, for both simulation strategies: continuous heating or night set back.

The model can compute the electricity consumptions of compressor, pumps and back-up heat system cumulated over a whole mean year (Fig. 9.6). The full lines correspond to a continuous heating, while the dotted lines are related to the night set back control.

The model shows that the total electricity consumption is nearly the same for both control strategies, while continuous heating requires a lower portion of high rate kWh, thus providing a financial economy.

Night set back control provides 1.7 % reduction of energy consumption, while continuous heating decreases the proportion of high rate electricity consumption by 12 %, providing 6% financial economy, if the ratio high rate/low rate electricity cost reaches 2. The total electricity consumption is close for both control strategies, because even with a variable set-point, the resulting indoor temperature is damped by the heating floor thermal mass (fig. 9.5b), so that it is close to a constant set point.

So the model can help to define a control strategy related to that heating system as far as energy costs are concerned: for a heating floor system, connected to a brine-water heat pump with a ground heat exchanger, the more interesting control strategy in terms of cost management is to maintain a constant indoor temperature set-point the whole day long.

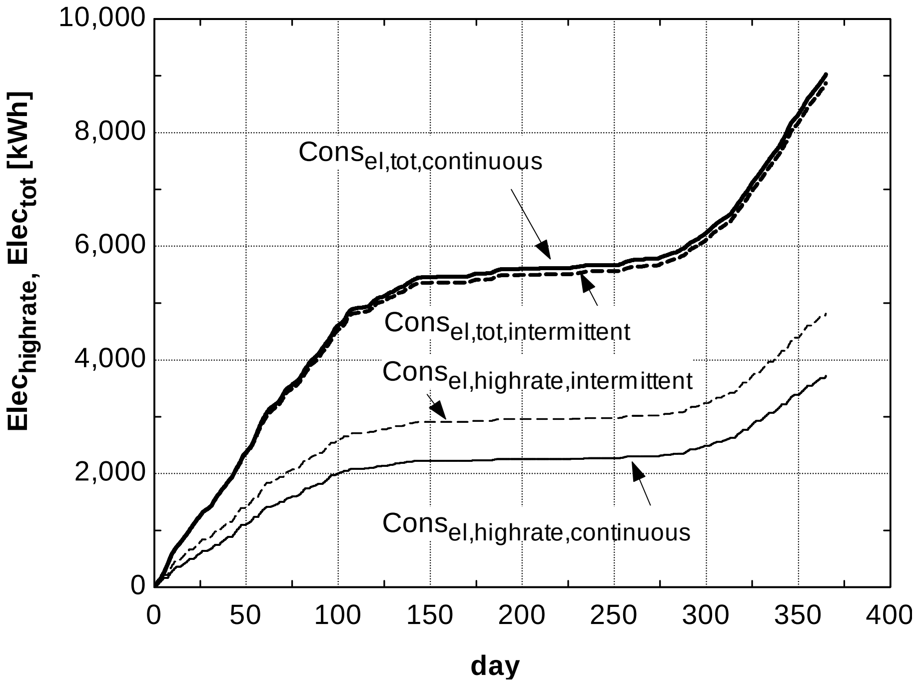

Fig. 9.6 Cumulated electricity consumptions for a continuous heating strategy (full lines) and for a night set back control (dotted lines). Upper curves show the total electricity consumption while lower curves represent the portion of consumption occurring during high rate hours.

9.2. Connection with HVAC unit

The office building lumped model validated in chapter 5 (fig. 5.34), and the office building ventilation model described in chapter 6, can both be connected to an HVAC system (fig. 9.7).

Zone heating can be performed either by AHU, through a *constant air flow system*, or by fan coil terminal units. When terminal units are used, the AHU can only provide the zones with the required fresh air flow.

Zone cooling can be performed either by AHU, through a *constant or variable air flow system*, or by terminal units. In the latter case, only the required fresh air flow can be provided to the zones by the AHU.

The AHU can include the following devices (fig. 9.7):

- Plate heat recovery exchanger;
- Economiser: when it isn't used during occupancy, it can be used during morning start-up of the system;
- Pre-cooling coil working together with an adiabatic humidifier, when it exists; both entities can be controlled by a humidity probe on return air;
- Cooling coil which can be controlled by a temperature probe on supplied air, and by a humidity probe on return air when dehumidification is required;
- Heating coil which can be controlled by a temperature probe on supplied air;
- Steam humidifier which can be controlled by a humidity probe on return air;
- Supply fan, whose rotating speed can be controlled by zone indoor temperature probes in case of variable air flow system;
- Post heating coil which can be controlled by a temperature probe on supplied air, for a variable air flow system.

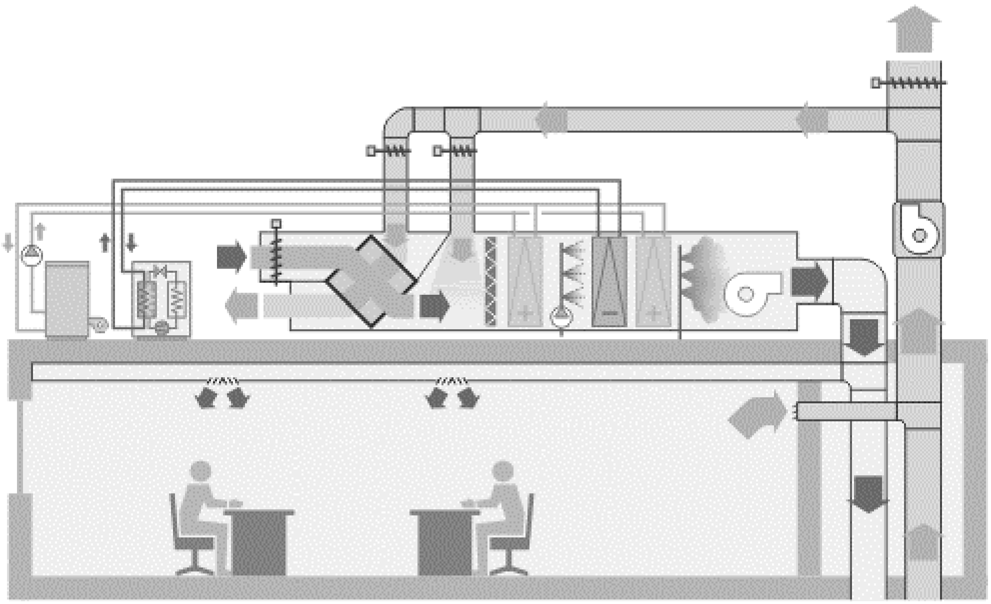

Fig. 9.7: Air Handling Unit connected to a building zone.

Plate heat recovery exchanger, heating coils and terminal unit coils can be modeled as classical heat exchangers (ref. [10], [11], [46] and [52]):

- Plate heat recovery exchanger can be modeled by a *cross-flow* heat exchanger with equal heat capacity rates on both sides:

(9.4)

$$NTU_{heat,recovery} = \frac{AU_{heat,recovery}}{\dot{C}_{a,heat,recovery}} \qquad \eta_{heat,recovery} = \frac{1}{NTU_{heat,recovery}^{\,0.22}}$$

$$\varepsilon_{heat,recovery} = 1 - \mathbf{exp}\left[\frac{\mathbf{exp}\,(-NTU_{heat,recovery} \cdot \eta_{heat,recovery}) - 1}{\eta_{heat,recovery}}\right]$$

- Heating coils can be modeled by *counter-flow* heat exchangers:

$$NTU_{heatingcoil} = \frac{AU_{heatingcoil}}{\dot{C}_{a,heatingcoil}} \qquad \omega_{heatingcoil} = \frac{\dot{C}_{a,heatingcoil}}{\dot{C}_{w,heatingcoil}}$$

(9.5)

$$\varepsilon_{a,heatingcoil} = \frac{1 - \mathbf{exp}\,(-NTU_{heatingcoil} \cdot (1 - \omega_{heatingcoil}))}{1 - \omega_{heatingcoil} \cdot \mathbf{exp}\,(-NTU_{heatingcoil} \cdot (1 - \omega_{heatingcoil}))}$$

- Terminal unit coils can be modeled by *semi-isothermal* heat exchangers:

$$NTU_{TU} = \frac{AU_{TU}}{\dot{C}_{a,TU}}$$

$$\varepsilon_{a,TU} = 1 - \mathbf{exp}\,(-NTU_{TU})$$

(9.6)

Adiabatic humidifier can be modeled through a given effectiveness:

$$W_{ex,adiabhum} = W_{su,adiabhum} + \varepsilon_{adiabhum} \cdot (W_{s,twb,su,adiabhum} - W_{su,adiabhum})$$

Saturated air humidity ratio $W_{s,twb,su,adiabhum}$ related to air supply wet-bulb temperature can be computed through an exponential correlation presented in annex 9.

Steam humidifier can be modeled through a nominal steam flow rate provided by the humidifier sizing in winter nominal conditions:

$$W_{ex,steamhum} = W_{su,steamhum} + \frac{\dot{M}_{steam,steamhum,n}}{\dot{M}_{a,steamhum}}$$

(9.7)

$$h_{steam,su,steamhum} = h_{fg0} + c_{p,g} \cdot t_{steam,su,steamhum}$$

$$h_{a,ex,steamhum} = h_{a,su,steamhum} + (W_{ex,steamhum} - W_{su,steamhum}) \cdot h_{steam,su,steamhum}$$

9.2.1 Cooling coil model

When working in *dry conditions*, the cooling coil can be modeled as a *counter-flow* heat exchanger (9.5).

When working in *wet conditions*, the cooling coil can be modeled through Threlkeld method, i.e. by considering the heat balance between the air and the cooling coil surface water film, on an elementary area *dA*:

- The convection exchange from air to water can be expressed as :

$$d\dot{Q}_c = h_c \cdot (t_a - t_w) \cdot dA \tag{9.8}$$

- The latent heat provided to the air by the water vapor condensation is:

$$d\dot{Q}_L = (h_m \cdot (c_{p,g} \cdot t_w + h_{fg0}) \cdot (w_{sat} - w_a)) \cdot dA \tag{9.9}$$

The heat balance can yield:

$$d\dot{Q}_{tot} = d\dot{Q}_c - d\dot{Q}_L \tag{9.10}$$

Lewis number Le equals the ratio of thermal diffusivity divided by mass diffusivity. Chilton and Colburn analogy between heat and mass transfer can be expressed as follows for an air-water vapor mixture [46]:

$$h_c = h_m \cdot (c_{p,a} + w_a \cdot c_{p,g}) \cdot Le^{(2/3)}$$

For an air-water vapor mixture, Le $= 0.872$, so that Le$^{2/3} = 0.913 \cong 1$, and:

$$h_c = h_m \cdot (c_{p,a} + w_a \cdot c_{p,g}) \tag{9.11}$$

Equation (9.10) can give:

$$d\dot{Q}_{tot} = \left[\frac{h_c \cdot dA}{c_{p,a} + w_a \cdot c_{p,g}}\right] \cdot (h_a - h_{sat,tw}) \tag{9.12}$$

Where $h_{sat,tw}$ is the enthalpy of saturated air at the surface water film temperature.

The air enthalpy h_a can be considered as nearly equal to the enthalpy of saturated air at a temperature equal to the air wet-bulb temperature, so that:

$$h_a - h_{sat,tw} = h_{sat,twb} - h_{sat,tw} \tag{9.13}$$

Saturated air can then be considered as a fictitious fluid with its own *saturated specific heat* named $c_{p,sat}$ expressed in *J/kg-K* (ref. [47] and [48]):

$$c_{p,sat} = \frac{h_{sat,twb} - h_{sat,tw}}{t_{wb} - t_w} \tag{9.14}$$

For a wet regime cooling coil, characterized by an heat exchange effectiveness, the air-water heat exchange can be expressed as function of the air supply and exhaust wet-bulb temperatures (fig. 9.8):

$$t_{wb,ex,coolingcoil,wet} = t_{wb,su,coolingcoil} - \varepsilon_{a,coolingcoil,wet} \cdot (t_{wb,su,coolingcoil} - t_{w,su,coolingcoil})$$

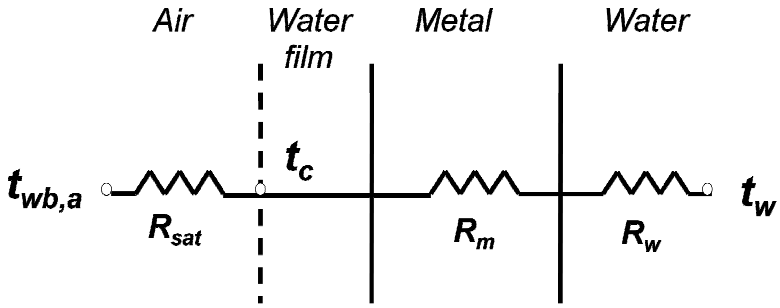

Fig. 9.8: Wet cooling coil surface water film model.

Global effectiveness in wet regime $\varepsilon_{a,coolingcoil,wet}$ can be computed as follows:

$$c_{p,sat} = \frac{h_{a,sat,twbsu,coolingcoil} - h_{a,sat,twbex,coolingcoil}}{t_{wb,su,coolingcoil} - t_{wb,ex,coolingcoil}} \tag{9.15}$$

$$R_{sat,coolingcoil} = R_{a,coolingcoil,n} \cdot \left[\frac{c_{p,a} + W_{su,coolingcoil} \cdot c_{p,g}}{c_{p,sat}} \right] \tag{9.16}$$

$$R_{sat,coolingcoil} + R_{m,coolingcoil,n} + R_{w,coolingcoil,n} = \frac{1}{AU_{coolingcoil,wet}}$$

$$\omega_{coolingcoil} = \frac{\dot{C}_{a,coolingcoil}}{\dot{C}_{w,coolingcoil,n}} \qquad NTU_{coolingcoil,wet} = \frac{AU_{coolingcoil,wet}}{\dot{C}_{a,coolingcoil}}$$

$$\varepsilon_{a,coolingcoil,wet} = \frac{1 - \mathbf{exp}\left(-NTU_{coolingcoil,wet} \cdot (1 - \omega_{coolingcoil})\right)}{1 - \omega_{coolingcoil} \cdot \mathbf{exp}\left(-NTU_{coolingcoil,wet} \cdot (1 - \omega_{coolingcoil})\right)}$$

The $c_{p,sat}$ saturated specific heat values can be correlated to air supply wet-bulb temperature and to air wet-bulb temperature decrease (see annex 9).

The heat exchange from air to water film can be considered as semi-isothermal with a constant contact temperature t_c (fig. 9.8). The *contact effectiveness* can be:

$$R_{sat,coolingcoil} = \frac{1}{AU_{coolingcoil,c}} \qquad NTU_{c,coolingcoil} = \frac{AU_{coolingcoil,c}}{\dot{C}_{a,coolingcoil}} \tag{9.17}$$

$$\varepsilon_{c,coolingcoil} = 1 - \mathbf{exp}\left(-NTU_{c,coolingcoil}\right)$$

The contact temperature t_c can be computed, so that air exhaust temperature can be deduced from:

$$t_{c,coolingcoil,wet} = t_{wb,su,coolingcoil} - \frac{\varepsilon_{a,coolingcoil,wet}}{\varepsilon_{c,coolingcoil}} \cdot \left(t_{wb,su,coolingcoil} - t_{w,su,coolingcoil}\right)$$

$$t_{a,ex,coolingcoil,wet} = t_{a,su,coolingcoil} - \varepsilon_{c,coolingcoil} \cdot \left(t_{a,su,coolingcoil} - t_{c,coolingcoil,wet}\right) \tag{9.18}$$

9.2.2 Application

The office room described in chapter 4 §4.1.3 can be repeated as described in chapter 7 § 7.2.2 to generate a four floors office building. The building and ventilation models, combined with a model of Air handling Unit, can be used to perform simulations regarding humidity control.

The office rooms can be submitted to 1976 hot wave. Fan coils can be used in offices to provide sensible cooling. Fresh air can be centrally cooled and dehumidified before being distributed to offices at an 18°C supply temperature. The temperature of the cold water supplied to the cooling coil can be 10°C. The central cooling coil can be controlled by a probe on supplied air temperature.

The evolution of the *outdoor air dew point*, with the corresponding *cooling coil sensible heat ratio* can be computed by the model (fig. 9.9). The model shows the cooling coil working in wet regime as soon as the air supplied dew point is lower than the supplied cold water temperature.

The model can estimate *indoor relative humidity* in parallel with indoor and air supply *humidity ratios* (fig. 9.10). The evolution of the humidity ratio is more continuous than the evolution of relative humidity, which is very sensible to the indoor temperature: relative humidity decreases as soon as air temperature increases, and vice versa.

Fresh outdoor air is dehumidified by the cooling coil so that the indoor relative humidity doesn't exceed 60%, which is sufficient for comfort. In that case, the control of the cooling coil through a supplied air temperature probe is sufficient to achieve comfort. A relative humidity probe isn't necessary.

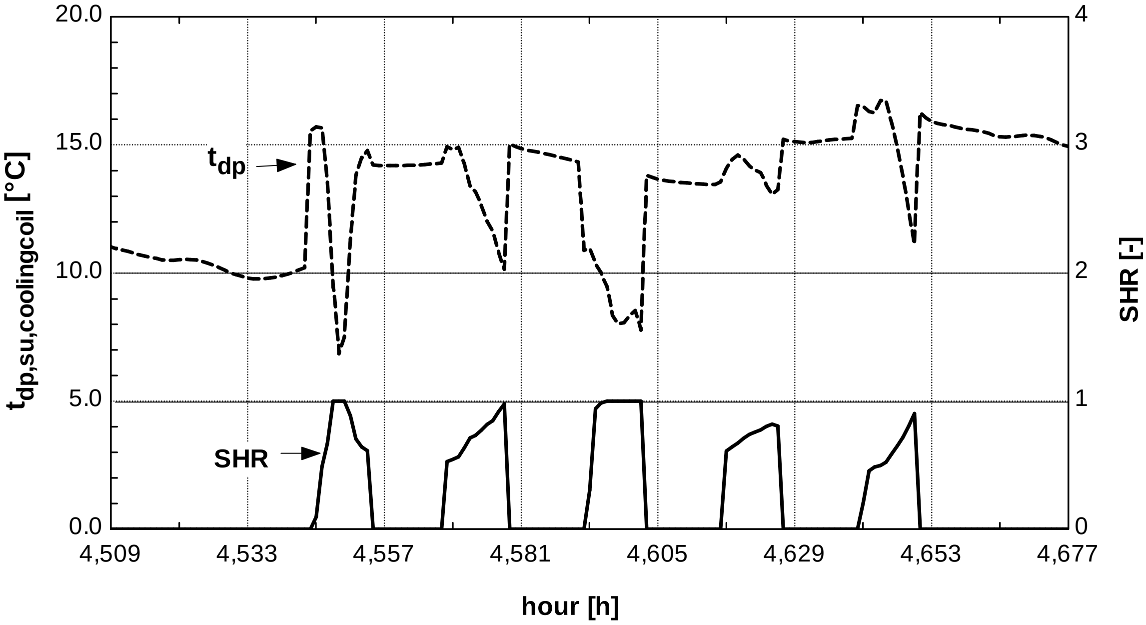

Fig. 9.9: Outdoor air dew point temperature, with the corresponding cooling coil Sensible Heat Ratio, for hot wave conditions.

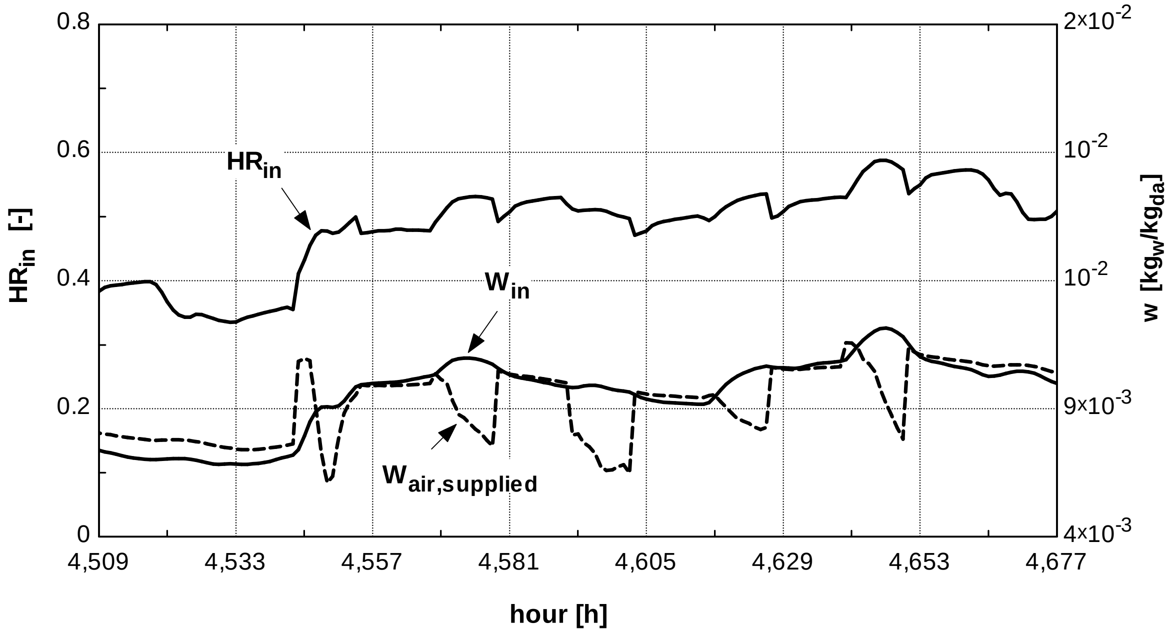

Fig. 9.10: Indoor relative humidity and humidity ratio related to the East offices, with the corresponding air supply humidity ratio, for hot wave conditions.

The offices rooms can also be submitted to the cold wave that occurred in December 1996 and January 1997. Fan coils can be used in offices for cooling. Fresh air can be centrally heated before being distributed to offices at an 18°C supply temperature.

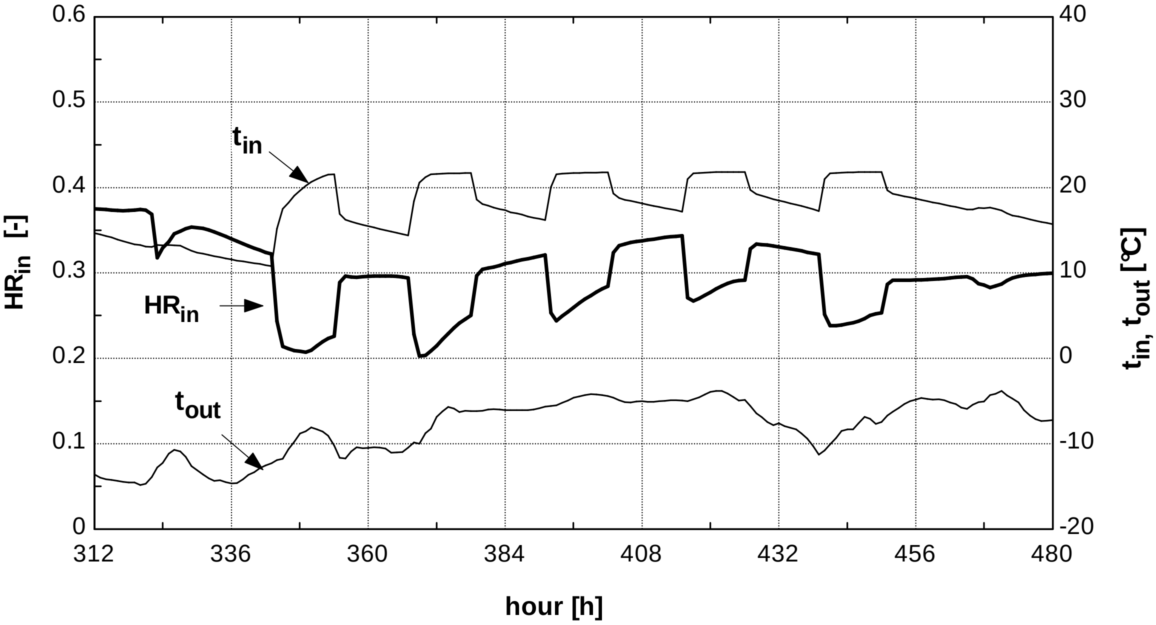

Fig. 9.11: Temperature and relative humidity in East office, with the corresponding outdoor temperature, for cold wave conditions.

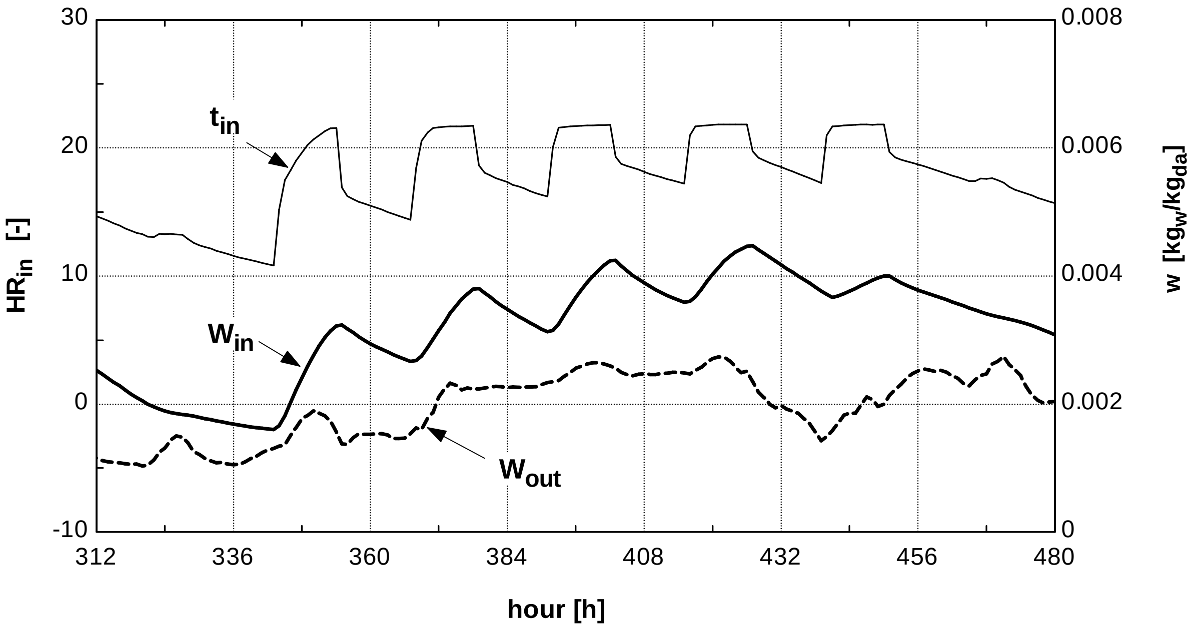

Fig. 9.12: Temperature and humidity ratio in East office, with the corresponding outdoor humidity ratio, for cold wave conditions.

The evolution of temperature and relative humidity in east office can be computed by the model, with the corresponding outdoor temperature. The indoor temperature set point can not be reached on time on Monday morning. When no humidification is performed, the relative humidity is too low fort comfort (fig. 9.11).

Here again, the relative humidity can be seen to be very sensible to indoor temperature. The relative humidity is lower during occupancy hours than during night time, while the humidity ratio is increasing during occupancy hours and decreasing during night time. So relative humidity isn't a good humidity level indicator and humidity control should better be performed on the basis of humidity ratio probes, instead of relative humidity probes.

Adiabatic humidification can be performed with a 40% relative humidity set point, in order to fulfill humidity comfort requirements (fig. 9.13).

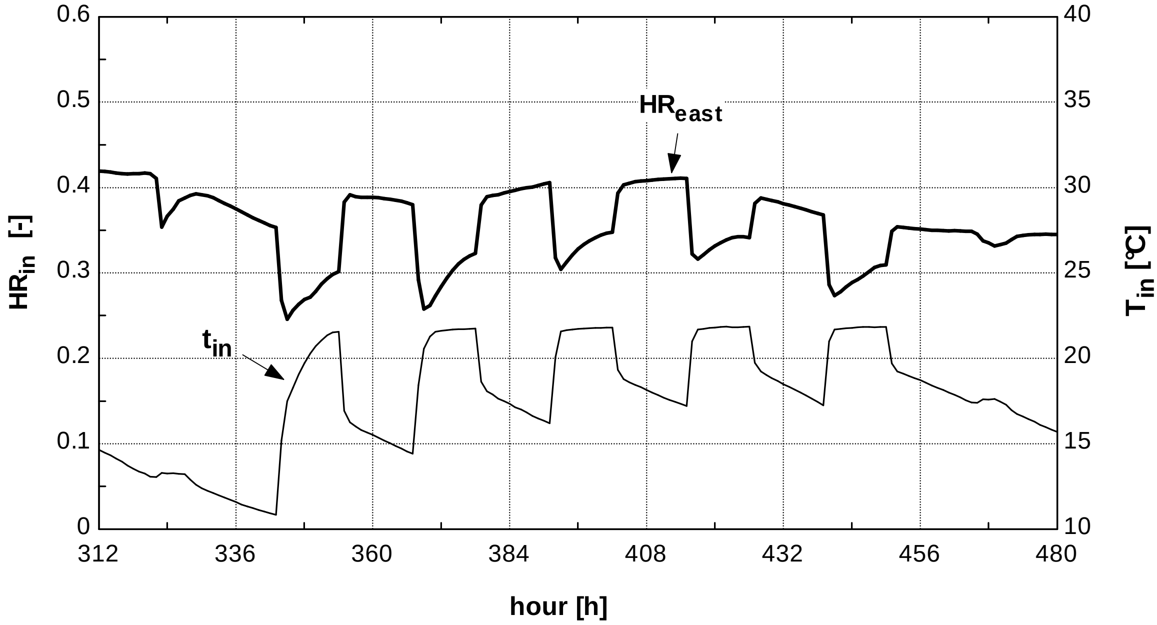

Fig. 9.13: Temperature and relative humidity in East office, when adiabatic humidification is performed, for cold wave conditions.

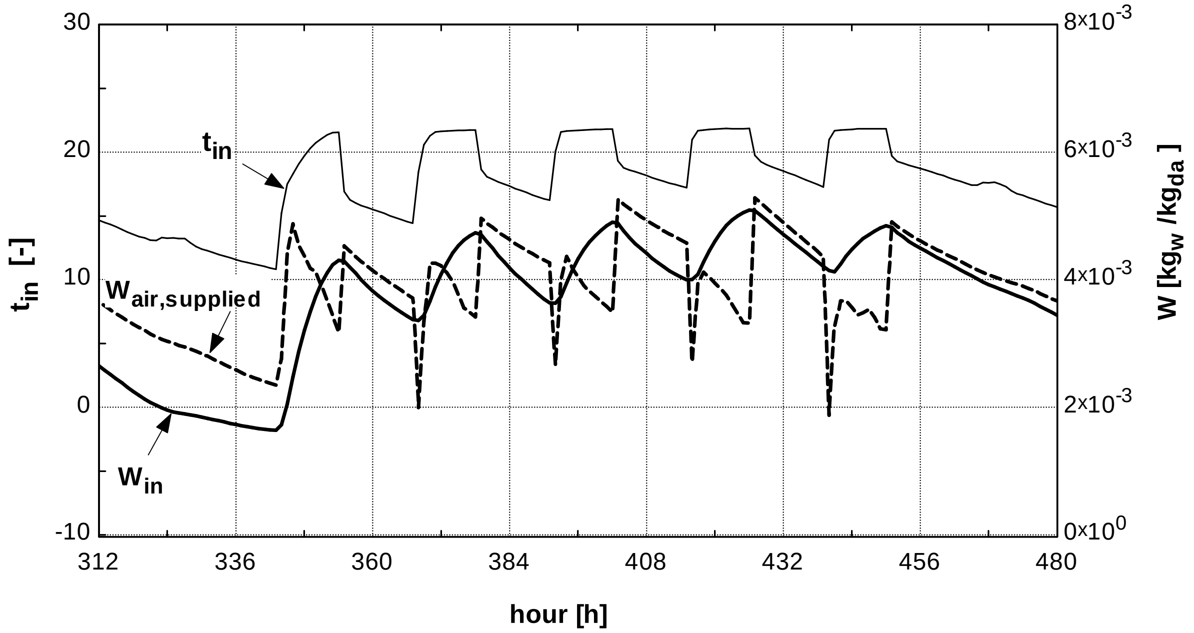

Fig. 9.14: Temperature and humidity ratios in East office, when adiabatic humidification is performed, for cold wave conditions.