The importance of integrally simulating the building, HVAC and control systems, and occupants’ impact for energy predictions of buildings including temperature and humidity control

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The importance of integrally simulating the building, HVAC and control systems, and occupants’ impact for energy predictions of buildings including temperature and humidity control: validated case study museum Hermitage Amsterdam

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For buildings including temperature and humidity control, this study compares the energy prediction accuracy of a ZABES-model (Zone Air Building Energy Simulation) to an IBES-model (Integral Building Energy Simulation), which additionally includes models of the air handling unit (AHU) and controllers. Museum Hermitage Amsterdam served as a case study. For one year, measurements were performed in the main exhibition hall and its AHU. The ZABES-model was developed using heat air and moisture model for building and systems evaluation (implemented in MATLAB). The IBES-model was developed in Simulink and consists of the ZABES-model and models of AHU-components and controllers. Both models have been validated in detail. The IBES-model’s energy prediction errors are well within 10%. However, the ZABES-model underestimated the total annual energy consumption by 84%. Moreover, including occupants’ heat and moisture gains leads to realistic results using the IBES-model, but leads to unrealistic results using the ZABES-model. In conclusion, IBES-models are essential for reliable energy predictions of buildings including humidity control.

Keywords: museum; simulation; building; HVAC; control; occupants

1. Introduction

Humidity control of buildings requires humidification and dehumidification. The former involves a one-step process increasing the air moisture content and slightly affecting the air temperature, for example, by means of steam humidification. The latter mostly involves a two-step process, for example, deep-cooling the air under the dew point temperature and subsequently reheating the air before supplying the air to the building zone. Besides, multiple control strategies exist for dehumidification, for example, controlling the RH directly or indirectly via absolute humidity. Frankly, humidity control significantly complicates BES.

For the sake of clarity, this paper differentiates BES-models into ZABES-models (Zone Air Building Energy Simulation) and IBES-models (Integral Building Energy Simulation). A ZABES-model includes a building envelope model, external loads, and internal loads, for example, from occupants and lighting. The ZABES-model calculates building energy demands by solving the energy and mass balance equations of the zone air. Additionally, an IBES-model includes dynamic models of air handling unit (AHU)-components and their controllers. For example, a ZABES-model calculates the dehumidification demand based on the latent energy of the moisture to be removed from the zone air, whereas an IBES-model includes cooling the air under the dew point temperature and reheating the air to the required supply temperature by an AHU.

The amount of buildings that employ both temperature and humidity control increases, for example, museums, hospitals, libraries, and data centres. Although building simulation models are widely used in the science community and are being ever more adopted by the engineering community (Hensen and Lamberts2011), most of the simulation studies aim to simulate zonal air temperatures, heating, and sensible cooling loads. A minority of studies involves buildings including humidity control. For example, ZABES-models have been applied by Ryhsvendsen et al. (2010), and IBES-models have been applied by Ascione, Bellia, and Capozzoli (2013); Ascione et al. (2009); Ayres et al. (1989); Zannis et al. (2006) to simulate buildings including temperature and humidity control. Moreover, most validation efforts, both test cells and full-scale buildings, only concern thermal simulation (Strachan et al. 2015). Comprehensive validation studies of IBES-models including both temperature and humidity control are lacking.

This paper presents a detailed validation study of an IBES-model for a building including both temperature and humidity control. The energy prediction accuracy of this IBES-model is compared to the energy prediction accuracy...
of only including the ZABES-model. Moreover, the impact of occupants, including visitors and staff, on annual energy predictions is compared for both models. Museum Hermitage Amsterdam served as a case study, in which indoor temperature and RH are strictly controlled. Moreover, the museum’s indoor environment is strongly affected by occupants, including heat gains, moisture gains, and CO₂-production initiating CO₂-controlled ventilation. The methodology included comprehensive measurements of the museum’s indoor climate and AHU for one year, developing a ZABES-model including occupants impact using heat air and moisture model for building and systems evaluation (HAMBASE) (van Schijndel 2007; de Wit 2006), developing an IBES-model using Simulink including the ZABES-model and models of the AHU-components and controllers. Both models have been validated in detail.

Section 2 describes the case study museum Hermitage Amsterdam, Section 3 elaborates on data acquisition, Section 4 concerns the modelling and validation process, Section 5 presents the results, and Section 6 provides a discussion and conclusions.

2. Case study: museum Hermitage Amsterdam

Museum Hermitage Amsterdam is a sister of museum State Hermitage in St. Petersburg, Russia. The museum is located in Amsterdam, the Netherlands. Museum Hermitage Amsterdam has no own collection, but displays loan exhibitions: the artworks mainly belong to the State Hermitage in St. Petersburg, but also to other museums. The most recent renovation dates from the years 2007–2009 when the building was transformed into a state-of-the-art museum building (see Figure 1): only the historical building envelope was preserved and the rest of the building has been newly built inwards. The historical envelope has been insulated from the inside, including thermal insulation and vapour barriers. Floor heating has been applied in the non-exhibition areas such as the restaurant and foyer, and an all-air HVAC system has been installed to condition the exhibition areas. An ATES system has been installed for seasonal heat and cold storage in the ground. The employed indoor climate specifications are 21°C and 50% RH, resulting in a stable museum environment, but unfortunately also in high energy cost.

2.1. The building

Figure 1(a) shows the layout of the building. It is a historical building from the seventeenth century located in between three canals. The historical appearance has been preserved by restoring the façade, but all the remaining parts of the building were rebuilt to accommodate the

Figure 1. (a) Museum Hermitage Amsterdam is located in between three canals. (b) One of the two main exhibition halls with a large glass roof. (c) The entrance stair from the lobby to the main exhibition hall with an air curtain reducing air exchange. (d) A cross section of one side of the building showing the main exhibition hall and adjacent cabinets.

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museum adequately. The building has a symmetrical floor plan: two identical exhibition wings may be recognized by the glass roof on the left and the right side. The central part, shown in the top of Figure 1(a), accommodates the main entrance with the foyer, the restaurant, the auditorium, and the restrooms.

This study focuses on ‘de Keizersvleugel’, which is the exhibition wing on the right side in Figure 1(a). The exhibition area consists of the main hall (Figure 1(b)) and the adjacent cabinets (Figure 1(d)). Visitors enter the exhibition area via the stair from the foyer (Figure 1(c)). The ceilings of the exhibition cabinets adjoin the technical areas that are located on the top floor. The ceiling of the main exhibition hall partly consists of a large glass roof with interior sun blinds that are almost closed permanently. An air curtain reduces the air exchange between the main exhibition hall and foyer. Furthermore, this interzonal air exchange is absent during closing hours due to the closing of the fire protection doors. We refer to Maas (2012) for a comprehensive description of the building materials, floor plans, and detailed drawings.

2.2. Internal heat and moisture gains

The museum is open 7 days per week from 10 until 17 h and welcomes between 7000 and 11,000 visitors per week, depending on the exhibition. The measurement campaign was executed during a period of one year (summer 2014–summer 2015) without change of exhibition, resulting in a repetitive weekly occupants’ presence profile. On Sunday, Tuesday, and Wednesday, most visitors were welcomed, on Monday the least.

The occupants’ impact, including visitors and staff, consists of three factors: heat production, moisture production, and fresh air supply by a CO₂-controlled ventilation system.

Besides, the lighting systems influence the indoor climate by emitting heat via convection and radiation. All lighting systems included halogen lamps at the time of measurements. Although halogen lamps itself generate a significant amount of heat, the overall heat production by lighting systems was limited due to low illuminance levels. Note that dyes and pigments exposed to light fade or change appearance; so illuminance levels are limited in most museums to 200 lux and 50 lux for very sensitive objects. The average heat load by lighting was determined by dividing the total lighting power by the exhibition’s floor area, resulting in 9 W/m².

2.3. Air exchange of the exhibition room

The exhibition room’s air exchange was analysed critically, enabling detailed modelling. The exhibition room’s air exchange includes three processes: (i) interzonal airflow between the exhibition room and the foyer (only from 7 to 19 h when the fire protection doors were open); (ii) infiltration through the building envelope; (iii) fresh air supply by a CO₂-controlled ventilation system (above 1000 ppm only). Table 1 indicates which of these three processes play a role for several time slots of the day.

2.4. Main exhibition hall’s AHU

The main exhibition hall of interest includes comprehensive air conditioning. See Figure 2 for an overview of the AHU. The AHU consists of a mixing section, a dust filter, a high temperature cooling coil (HT-CC) used to cool...
the air and to precool the air in case of dehumidification, a steam humidifier, a low temperature cooling coil (LT-CC) with bypass to dehumidify the air, a fan, a high temperature heating coil (HT-HC), and a filter section including electrostatic, chemical, active carbon, and an end filter.

Most of the time, all air is recirculated. When a CO2-level threshold of 1000 ppm has been exceeded, the outdoor air valve will be controlled to supply fresh outdoor air. The CO2-level was measured in the exhibition hall.

The counter-flow HT-CC has four rows, and its designed cooling capacity is 55 kW at a water supply temperature of 12°C and return temperature of 16°C. The counter-flow LT-CC has eight rows, and its designed cooling capacity is 111 kW at a water supply temperature of 6°C and a return temperature of 10°C, but this is reduced to 74 kW because of the bypass construction. The counter-flow low temperature heating coil (LT-HC) has four rows, and its designed cooling capacity is 128 kW with a water supply temperature of 45°C and a return temperature of 35°C.

The steam humidifier has a maximum capacity of 18 kg/h with an electrical power supply of 15 kW. The amount of steam injection is controlled via PI-controlled modulation.

The belt-driven centrifugal fan with backward-curved blades has a maximum shaft power of 9.74 kW and a maximum air displacement of 6.11 m³/s. It provides an air volume flow of 16,000 m³/h, resulting in an ACR of 7.5 h⁻¹.

A comprehensive filter section completes the AHU: an electrostatic filter to remove mould, micro dust, pollen, and other allergens; an active carbon filter to remove contaminant gasses as ozone and formaldehyde by adsorption, absorption, and oxidation; end particle filter.

3. Data acquisition

3.1. Indoor T, RH, and CO2

The exhibition room of interest is equipped with four sensors that are connected to the BMS: three Hanwell Radiologgers ML4106 combined T (± 0.2°C) and RH (± 2% RH) measurement; one Catec EE80 series combined CO2 (± 50 ppm + 2% of the measuring value), T (± 0.3°C), and RH (± 3% RH) measurement. The sampling interval of the indoor measurement data is 16 min. The four sensors were attached to the four walls of the exhibition room at a height of 2 m. Spatial gradients were found to be very small: a temporary measurement grid including 12 locations in the exhibition hall provided data on Ta and RHa at a sampling rate of 1 Hz, showing standard deviations (among different sensor locations) of only 0.37°C and 1.7% RH. Therefore, the average of the four wall-mounted sensors has been used as indoor Ta and RHa. Spatial gradients of indoor conditions have not been taken into account further.

3.2. Outdoor climate data

Outdoor Ta and RHa were measured at the museum site in Amsterdam. The sampling interval was 16 min, but this was converted to hourly values for simulation by nearest point interpolation. Diffuse solar radiation [W/m²] and direct solar radiation [W/m²] were retrieved from the Royal Netherlands Meteorological Institute’s database (KNMI 2015) with a sampling interval of 60 min. These data were measured by weather station ‘Schiphol Airport’, approximately 15 km southwest of the museum site. Climate year 2014 was used for validation.

3.3. AHU measurement campaign

The AHU that air conditions the main exhibition hall of interest was monitored comprehensively. Figure 2 shows the measurement campaign. This section describes the measurement positions, methods, and post-processing of data.

Energy consumptions of heating (LT-HC), cooling (HT-CC), and dehumidification (LT-CC) were calculated based on the energy exchange between the water side and air side of the coils, according to

\[ P_{c,a} = m_w C_p \left(T_{w,o} - T_{w,i}\right), \]  

where \( P_{c,a} \) [kW] is the thermal heat exchange rate between the coil and air; \( m_w \) [kg/s] is the water mass flow; \( C_p \) is the specific heat of water (4.0 kJ/kg K for a mixture of 75% water and 25% glycol); \( T_{w,o} \) [°C] and \( T_{w,i} \) [°C] are the temperatures of the outlet and inlet water flows. The water mass flow was calculated from measurements of the pressure drop across the balancing valves according to

\[ m_w = \frac{K_v}{36} \sqrt{\Delta P}, \]  

where \( m_w \) is the water mass flow [kg/s], \( K_v \) is the coefficient of flow (from manufacturer’s tables), and \( \Delta P \) is the pressure drop across the balancing valve [kPa]. The pressure drop was measured using TA Hydronics’ TA Link (see Figure 2) with an inaccuracy of < 1 kPa and measuring range of 0–100 kPa.

The temperatures of the supply and return water flow of the LT-HC, HT-CC, and LT-CC were measured by Grant thermistors with an accuracy of ± 0.1°C. The measuring tips were positioned at the external surface of the piping, right under the insulation material. Temperature [°C] follows from the Steinhart–Hart equation according to

\[ T = \frac{1}{a + b \ln(R) + c \ln(R)^3} - 273.15, \]  

where \( \ln \) is the log to base \( e \), \( R \) is the resistance [Ohm]. The values of the coefficients are: (a) 1.498872e-3, (b) 2.379047e-4, (c) 1.066953e-7.

Ta and RHa were measured by E+E Elektronik EE160 with an error after calibration < 0.1°C and < 1.5%
(standard accuracy of ±0.2°C and ±2.5%). These sensors were installed at four positions (see Figure 2): before the HT-CC, between the HT-CC and steam humidifier, between the fan and LT-HC, and after the LT-HC. Since the steam humidifier and LT-CC were not active simultaneously, this setup resulted in a dataset including inlet and outlet air conditions of all active components. Air moisture content \( w_a \) [g/kg] was calculated from measured \( T_a \) [°C] and \( RH_a \) [–] according to

\[
w_a = P_{\text{sat}}(T_a) \cdot RH_a \cdot 0.0062, \tag{4}
\]

where \( P_{\text{sat}} \) [Pa] (Künzel 1995) was calculated for \( T_a \geq 0°C \) according to

\[
P_{\text{sat}} = 611 \cdot e^{17.67 T_a/(234.18+T_a)}, \tag{5}
\]

and for \( T_a < 0°C \) calculated according to

\[
P_{\text{sat}} = 611 \cdot e^{22.44 T_a/(273.15+T_a)}. \tag{6}
\]

The measurement data were logged at an interval of 30 s by a Grant dataTaker® DT85. The data were sent via File Transfer Protocol once a day to a server located at the University.

The electric power consumptions of the fan and steam humidifier were measured using ND Metering Solutions’ Rail 350 (Figure 2), with a resolution of 10 pulses/kWh.

Furthermore, measurement data were available from the BMS: control signals of all components and controllers’ settings.

4. Modelling and validation

4.1. Building model (ZABES)

A multi-zone building model was developed using HAMBASE (van Schijndel 2007; de Wit 2006), a Heat Air and Moisture modelling and simulation tool developed in the scientific programming environment MATLAB at Eindhoven University of Technology. The model was used to simulate indoor air temperature, relative air humidity, and energy consumption for heating, cooling, humidifying and dehumidifying the multi-zone building. See van Schijndel (2007) and de Wit (2006) for extensive information on HAMBASE. Validation exercises of the thermal and hygric part are provided by Kramer et al. (2015). Here, only a brief explanation is included.

HAMBASE includes for every zone an indoor model that is coupled to an envelope model. The thermal indoor model (Figure 3) consists of two coupled equations: the heat balance of the air temperature \( (T_a) \), and the heat balance of the resultant temperature \( (T_x) \). The latter is a combination of air and radiant temperature and may be interpreted as the temperature experienced by the walls.

\( T_x \) is needed to calculate transmission heat losses with a combined surface coefficient. \( h_c \) and \( h_{cv} \) are the area weighted mean surface heat transfer coefficients for radiation and convection. \( \Phi_r \) and \( \Phi_{cv} \) are the radiant and convective part of the total heat input consisting of heating or cooling, as well as casual gains and solar gains.

For each heat source, a convection factor may be provided between 0 and 1 by the user. For example, for air heating, this factor is close to 1 and for radiative heat sources, a factor ranging from 0.4 to 0.6 may be used. The factor for solar radiation depends on the window system and the amount of radiation falling on furniture. \( C_a \) is the heat capacity of the air. \( L_{xa} \) is a coupling coefficient:

\[
L_{xa} = A_{tot} h_{cv} \left(1 + \frac{h_{cv}}{h_r}\right). \tag{7}
\]

\( \sum \Phi_{ab} \) is the heat loss by air entering the zone with an air temperature \( T_b \). \( A_{tot} \) is the total area. In the case of ventilation, \( T_b \) is the outdoor air temperature. \( \sum \Phi_{xy} \) is transmission heat loss through the envelope part \( y \). For external envelope parts, \( T_y \) is the sol-air temperature for the particular construction, including the effect of atmospheric radiation.

The admittance for a particular frequency can be represented by a network of a thermal resistance \((1/L_x)\) and capacitance \((C_x)\) because the phase shift of \( V_x \) can never be larger than \( \pi/2 \). To cover the relevant set of frequencies (period 1 and 24 h), two parallel branches of such a network are used, giving the correct admittances for cyclic variations with a period of 24 and 1 h. For outdoor air entering the room with temperature \( T_b \), a loss coefficient \( L_v \) is introduced. Figure 4 shows the complete thermal model for one zone.

In a similar way, a model for air humidity is developed. Vapour transfer through a wall is negligible compared to vapour transfer by ventilation; so only the latter is taken into account. The hygric indoor model takes moisture storage in the air and moisture storage in furniture into account. The hygric model takes moisture storage in furniture into account. The hygric indoor model takes moisture storage in furniture into account. The hygric indoor model takes moisture storage in furniture into account.

The hygroscopic curve is linearized between 20% RH and 80% RH: inaccuracy becomes significant towards 95% RH, which occurs very rarely in a museum. The vapour permeability is assumed to be constant. The main differences are: (i) there is only one room node (the vapour pressure); (ii) the moisture storage in walls, furniture, and carpets depends on RH and \( T \).
HAMBASE uses a delta-star transformation, analogous to electrical systems, reducing radiant temperatures to one node (de Wit 2006). It implies that radiation is spread evenly over all surfaces. This approximation eliminates the need of view factors and therefore it has the advantage that complicated geometries can be modelled easily. The introduced error due to this approximation is particularly small for insulated buildings in which surface temperatures are nearly equal and do not differ significantly from air temperatures. Although surface temperatures may differ significantly from air temperature in historical buildings, the error is acceptably small because radiation contributes far less to heat exchange than for example air infiltration.

The model of the museum consists of nine zones: one main exhibition hall and all adjacent rooms. The artworks exhibited during the measurement campaign only included paintings. So, the moisture capacity of the indoor model is predominantly determined by the moisture capacity of the air. The moisture capacity of the walls is taken into account by the envelope model. The ZABES-model was validated by comparing air-sided energy supply to simulated energy supply. Air-sided energy supply was calculated based on the specific enthalpy difference between measured supply air conditions \((T_{a}, w_{a})\) and measured indoor air conditions \((T_{a}, w_{a})\), multiplied by measured supply air mass flow \((\dot{m}_{a})\). The validation of the ZABES-model’s dynamics were validated by comparing measured to simulated indoor \(T_{a}\) and RH\(_{a}\). The validation of the ZABES-model is comprehensively described in Kramer et al. (2015).

The ZABES-model was implemented using an S-function, retaining the flexibility required for multi-domain modelling and avoiding the use of the block-oriented approach in complex models (van Schijndel 2013). After all, the block-oriented approach may become cumbersome if the model includes many ODEs. Figure A1 in the Appendix shows the implementation of the ZABES-model in Simulink using the S-function of HAMBASE.

4.2. Occupants’ impact and air exchange based on CO\(_{2}\)-data

CO\(_{2}\)-data of one year was converted to an hourly week profile: each CO\(_{2}\)-datapoint was labelled with information including the day of the week and hour of the day when the data point was measured. This enabled the construction of a week profile of measured CO\(_{2}\)-concentrations (see Figure 5). The grey plots show the mean measured CO\(_{2}\)-concentration and the standard deviation of the measurements for each hour of the week. The base concentration was relatively high because outdoor concentration in Amsterdam was approximately 600 ppm. The increase of CO\(_{2}\)-level, starting at 10 h, resulted from CO\(_{2}\)-production of occupants. The decay of the CO\(_{2}\)-level resulted from three processes: infiltration of outdoor air, interzonal air exchange, CO\(_{2}\)-controlled ventilation (active above 1000 ppm). Table 1 summarizes the time slots that these processes were active.

The model HAMBASE was used to calculate CO\(_{2}\)-concentration according to

\[
\frac{dC(t)}{dt} = \frac{\dot{V}_{\text{mech}} + \dot{V}_{\text{inf}}}{V}(C_{o} - C(t)) + \frac{\dot{V}_{\text{intz}}}{V}(C_{i} - C(t)) + \frac{G}{V},
\]

(8)

where \(C(t)\) is indoor CO\(_{2}\)-concentration at time \(t\), \(C_{o}\) is outdoor concentration, \(C_{i}\) is the concentration in adjacent zones [kg/m\(^3\)]. \(\dot{V}_{\text{mech}}\) is the mechanical ventilation rate (i.e. fresh air supply rate), \(\dot{V}_{\text{inf}}\) is the infiltration rate, and \(\dot{V}_{\text{intz}}\) is the interzonal rate [m\(^3\)/s]. \(V\) is zone volume [m\(^3\)] and \(G\) is the CO\(_{2}\)-production rate [kg/s]. The concentration was converted to ppm\(_{v}\), according to

\[
\text{ppm}_{v} = \frac{24.12}{M}\frac{\text{mg}}{\text{m}^3},
\]

(9)

where \(M\) is the relative molar mass, which is 44 for CO\(_{2}\) (ASHRAE 2009).

4.2.1. Infiltration and interzonal airflow

The decay curves show a transition occurring every day between 18 and 19 h when security walks through the museum and finishes around 19 h closing the fire protection doors. Interzonal air exchange is eliminated, leaving only infiltration resulting in a slower CO\(_{2}\)-decay. This is indicated by the red arrow in Figure 5 on Sundays. Therefore, CO\(_{2}\)-decay from 19 until 24 h was used to determine the infiltration rate. Firstly, measured CO\(_{2}\)-levels at 19 h were set as initial values in HAMBASE for each day of the week. Secondly, measured decay curves were reproduced by calibrating the infiltration rate for each day. The infiltration rate was found to be nearly identical for each day of the week: 0.11 h\(^{-1}\). The calibrated infiltration rate is considered to be a mean infiltration rate since it was found by fitting simulated CO\(_{2}\)-levels to mean CO\(_{2}\)-data.
After determining the infiltration, it was possible to determine the interzonal air exchange between the main exhibition hall and the foyer. The museum was closed at 17h for visitors and the fire protection doors were closed at 19h, leaving time slots from 17 until 19h to determine the amount of interzonal airflow. Auxiliary measurements at one day showed that the CO2-level in the foyer reached 750 ppm around 17h. Firstly, CO2-levels measured in the main exhibition hall and the foyer at 17h were set as initial values in HAMBASE for each day of the week. Secondly, measured decay curves were reproduced by calibrating the interzonal ACR for each day. It was found to be nearly identical for each day of the week: 1180 m3/h. In the final model, which also included the visitors’ CO2-production for every day of the week (see Section 4.2.2), the simulated CO2-level in the foyer at 17h varied between 650 and 850 ppm, depending on the day of the week.

4.2.2. Occupants’ presence

Now that infiltration rates and interzonal air flows were known, the number of occupants could be determined. Occupants’ CO2-production was used to fit the simulated CO2-level to mean measured CO2-levels as shown in Figure 5. CO2-production for light activity, for example, walking slowly through the museum, was assumed to be 9.35 mg/s per person (Schoeller et al. 1986). Time slots from 10 to 17h were used to calibrate the number of occupants for each hour of the week.

Figure 5. For each weekday, mean CO2-concentration and its standard deviation are shown (grey plots). Blue curves represent the simulated CO2-concentration by HAMBASE, as a part of the IBES-model. Closing the fire protection doors at 19h eliminates interzonal air exchange, clearly affecting the decay curve (indicated by the red arrow). The resulting occupants’ presence profile is shown at the bottom right.
The mean CO₂-concentration curves were reproduced accurately, see Figure 5 (blue curves). This resulted in an occupants’ presence profile, as shown in Figure 5 (bottom right). Also, the presence of security staff is visible around 19 h.

4.2.3. Occupants’ heat and moisture gains
Calculation of heat and moisture production was based on the calibrated number of occupants and indoor air temperature (see Figure 6). Heat production was based on sensible heat loss per person, and moisture production was based on latent heat loss per person for light activity. The relationships are based on the ASHRAE-standard 55:2010 model for thermal comfort (ASHRAE 2010).

Sensible heat gains by occupants were implemented according to

\[ \Phi_{vis} = (-4.46 T_i + 176) N_{occ}, \]  

where \( \Phi_{vis} \) is the sensible heat flux by occupants [W], \( T_i \) is the indoor temperature [°C], and \( N_{occ} \) the number of occupants. The moisture production was calculated based on latent heat gains by occupants according to

\[ \dot{m}_{w,vap,occ} = \frac{4.46 T_i - 46}{\Delta h_{vap}} N_{occ}, \]  

where \( \dot{m}_{w,vap,occ} \) is the moisture production rate by occupants [g/s] and \( \Delta h_{vap} \) is the specific enthalpy of evaporation (\( \approx \) 2257 J/g for water).

4.3. Air handling unit
This section elaborates on the model development of the AHU-components. Figure A2 in the Appendix provides an overview of the AHU-model in Simulink, and Figure A4 shows the hygrothermal coupling of the AHU-model and the ZABES-model.

4.3.1. Cooling, dehumidification, and heating coils
The coils, HT-CC, LT-CC, and LT-HC, were modelled according to ASHRAE RP-1194 (Zhou and Braun 2007a, 2007b). For each row, two ODEs were solved: heat transfer from the water to the coil and heat transfer from the coil to the air. Moreover, two scenarios were modelled: dry surface condition and wet surface condition.

Water to coil transfer was modelled according to

\[ C_w \frac{dT_{o,w}}{dt} + \dot{C}_w(T_{o,w} - T_{i,w}) + \frac{1}{R_w}(T_{i,w} - T_c) = 0, \]  

where \( C_w \) [J/K] is the total heat capacitance of the water, \( \dot{C}_w \) [W/K] is total heat capacitance of the water flow, \( R_w \) [K/W] is the total thermal resistance between water and coil, according to

\[ R_w = \frac{1}{\epsilon_w C_w}, \]  

where \( \epsilon_w \) is the water side heat transfer effectiveness of a row, according to

\[ \epsilon_w = 1 - e^{-\eta_{NTU_w}}, \]  

where \( NTU_w \) is the number of transfer units of the water side, according to

\[ NTU_w = \frac{h_w A_w}{C_w}, \]  

where \( A_w \) [m²] is the total internal tube surface, \( h_w \) [W/m²K] is the water side heat transfer coefficient.

Coil to air heat transfer in dry conditions was modelled according to

\[ C_a \frac{dT_c}{dt} + \frac{1}{R_a}(T_c - T_{i,a}) + \frac{1}{R_w}(T_{i,w} - T_c) = 0, \]  

where \( C_a \) [J/K] is the total capacitance of the coil, \( R_a \) [K/W] is the total thermal resistance between air and coil, according to

\[ R_a = \frac{1}{\epsilon_a C_a}, \]  

where \( \epsilon_a \) is the air side heat transfer effectiveness of a row, according to

\[ \epsilon_a = 1 - e^{-\eta_{NTU_a}}, \]  

where \( NTU_a \) is the number of transfer units for the air side, according to

\[ NTU_a = \frac{\eta_a h_a A_a}{C_a}, \]  

where \( A_a \) [m²] is the total external tube surface, \( h_a \) [W/m²K] is the air side heat transfer coefficient, calculated according to

\[ h_a = a \frac{m_a}{A_a \times \rho_a} + b, \]

where \( m_a \) [g/s] is the mass flow rate of the air, \( A_a \times \rho_a \) [m²] is the cross sectional area of the AHU, and \( \rho_a \) [kg/m³] is the
density of air. The coefficients $a$ and $b$ are available from Zhou and Braun (2007a).

$\eta_{\text{tot}}$ is the overall fin efficiency for heat transfer, according to

$$\eta_{\text{tot}} = 1 - r_{\text{fin}}(1 - \eta_{\text{fin}}), \quad (22)$$

where $\eta_{\text{fin}}$ is the individual fin efficiency and $r_{\text{fin}}$ is the ratio of fin area $A_{\text{fin}}$ to total surface area $A_{\text{tot}}$, according to

$$r_{\text{fin}} = \frac{A_{\text{fin}}}{A_{\text{tot}}}. \quad (23)$$

If the coil is wet, Equations (16) and (17) are rewritten as

$$C_{c} \frac{dT_{c}}{dt} + \frac{1}{R_{s}}(h_{s,c} - h_{i,a}) + \frac{1}{R_{w}}(T_{c} - T_{i,w}) = 0, \quad (24)$$

$$h_{o,a} = h_{i,a} + \varepsilon_{a}^{*}(h_{s,c} - h_{i,a}), \quad (25)$$

where $h_{s,c}$, $h_{i,a}$, and $h_{o,a}$ are the enthalpies [kJ/kg] of coil surface, inlet air, and outlet air, respectively. $R_{a}^{*}$ [s/g] is the total heat and mass transfer resistance, according to

$$R_{a}^{*} = \frac{1}{\varepsilon_{a}^{*} \dot{m}_{a}}, \quad (26)$$

where $\dot{m}_{a}$ [g/s] is the mass flow rate of the air, and $\varepsilon_{a}^{*}$ is the effectiveness of combined heat and mass transfer, according to

$$\varepsilon_{a}^{*} = 1 - e^{-\text{NTU}_{a}^{*}}, \quad (27)$$

where $\text{NTU}_{a}^{*}$ is the number of transfer units for the air side, according to

$$\text{NTU}_{a}^{*} = \frac{\eta_{\text{tot}}^{*} h_{s,c}^{*} A_{a}}{C_{a}}, \quad (28)$$

where $h_{s,c}^{*}$ is the convection coefficient for air side heat transfer and dehumidification, $\eta_{\text{tot}}^{*}$ is the overall fin efficiency of combined heat and mass transfer, according to

$$\eta_{\text{tot}}^{*} = 1 - r_{\text{fin}}(1 - \eta_{\text{fin}}^{*}), \quad (29)$$

where $\eta_{\text{fin}}^{*}$ is the individual fin efficiency under wet conditions.

The HT-CC of the exhibition hall’s AHU was modelled and the following parameters have been calibrated in the Simulink environment: $r_{\text{fin}}$, $A_{\text{a,cross}}$, $A_{a}$, $h_{w}$, $A_{w}$, $C_{w}$, $C_{c}$. Simulink’s parameter-estimation toolbox was used to calibrate the parameters using the Levenberg–Marquardt algorithm (Levenberg 1944; Marquardt 1963). It can solve nonlinear problems via least-squares and was found to be the most efficient and robust algorithm for the optimization problems in this study (other tested algorithms: Trust-Region-Reflective, Pattern-Search, Simplex Search). The algorithm was used to minimize the total simulation error by optimizing the model’s parameters. The following measurements were used as inputs: air temperature and air humidity before the cooling coil, water supply temperature, and water mass flow. The following simulation results were compared to measurements (comprising the total simulation error): air temperature after the coil, air humidity after the coil, and the water return temperature. Figure 7 shows the results of the calibration for air temperatures (a), air moisture content (b), water temperatures and water mass flow (c). The model proved to be able to capture the dynamics accurately: $\varepsilon < 0.10^\circ\text{C}$ for temperature and $\varepsilon < 0.15$ g/kg for moisture content. The cooling process remained in the dry regime, so without dehumidification. Hence, the simulated moisture content of the outlet air is equal to the measured inlet moisture content. However, the measured moisture content of the outlet air slightly deviates from the measured inlet moisture content, but this is predominantly due to the measurement inaccuracy of RH.

The LT-HC is identical to the HT-CC (four rows): the model was duplicated, but ODEs for the wet regime were excluded to improve simulation speed. The LT-CC for dehumidification is equal to the HT-CC, but it includes eight rows instead of four. So the LT-CC was modelled by increasing the number of rows to eight.
4.3.2. Steam humidifier

Enthalpy balance was used to calculate the specific enthalpy of the outlet air \( h_{o,a} \) [kJ/kg], according to

\[
h_{o,a} = h_{i,a} m_a + h_{st} m_{st}, \tag{30}
\]

where \( m_a \) is air mass flow [kg/s], \( h_a \) [kJ/kg] is specific enthalpy of the injected steam, \( m_{st} \) [kg/s] is mass flow of the steam, and \( h_{i,a} \) [kJ/kg] is specific enthalpy of the inlet air, according to

\[
h_{i,a} = \frac{C_p a T_{i,a} + w_{i,a} (C_p w T_{i,a} + h_{w,evap})}{m_a}, \tag{31}
\]

where \( C_p a \) [kJ/kg K] is specific heat of dry air, \( T_{i,a} \) is the inlet air temperature [°C], \( w_{i,a} \) [kg/kg] is the humidity ratio of the inlet air, \( C_p w \) is specific heat of water [kJ/kg K], and \( h_{w,evap} \) is specific evaporation energy of water [kJ/kg].

Specific moisture content of the outlet air, \( w_{o,a} \) [kg/kg], was calculated according to

\[
w_{o,a} = \frac{w_{i,a} m_a + m_{st}}{m_a + m_{st}}, \tag{32}
\]

which, together with \( h_{o,a} \), enables calculating outlet air temperature \( T_{o,a} \) [°C], according to

\[
T_{o,a} = \frac{h_{o,a} - w_{o,a} h_{w,evap}}{C_p a + C_p w w_{o,a}}. \tag{33}
\]

The steam humidifier was modelled and calibrated in the Simulink environment. In Equations (30) and (32), the steam injection mass flow (\( m_{st} \)) was unknown from measurements. However, the control signal [%] was known from measurements. Simulink’s parameter-estimation toolbox was used to relate the steam injection mass flow (\( m_{st} \)) to the control signal. The injection valve was modelled as described in Section 4.3.4 and the coefficients (a–d) were calibrated using the Levenberg–Marquardt algorithm. The following measurements were used as inputs: air temperature and air humidity before the steam humidifier, and the control signal of the steam injection. The following simulation results were compared to measurements: air temperature and air humidity after the humidifier. Figure 8 shows the results after the calibration of air moisture content (a), air temperatures (b), measured control signal (c), and identified relation between the control signal and the steam injection rate (d). The model proved to be able to capture the dynamics accurately: both moisture content (\( \varepsilon < 0.2 \) g/kg) and temperature (\( \varepsilon < 0.15 \)°C) were reproduced accurately.

4.3.3. Fan

The fan provides airflow at a constant rate of 16,000 m³/h. Therefore, the fan is simply modelled as a two-parameter model: one parameter for air volume rate, and one parameter for temperature increase due to fan inefficiency. If the fan is not working, both volume rate and temperature increase are zero.

![Figure 8](image) Validation of steam humidifier: (a) moisture content of inlet (\( w_{i,a} \)), outlet (\( w_{o,a} \)), and simulated outlet air (\( w_{o,a \text{ sim}} \)); (b) temperature of inlet (\( T_{i,a} \)), outlet (\( T_{o,a} \)), and simulated outlet air (\( T_{o,a \text{ sim}} \)); (c) control signal; (d) relationship of control signal (measured) and steam injection rate (found by optimization).
Figure 9. The temperature increase of the fan ($\Delta T$), proportional to control.

The parameter ‘air flow rate’ was known from measurements published in commissioning reports. The parameter ‘temperature increase’ was determined using data from the measurement campaign (see Figure 9): air temperature difference across the fan is plotted together with the control signal (percentage of maximum rpm). The results show that temperature increase is 0.4°C at a control signal of 35% and 1.8°C at a control signal of 85%. The latter being the normal operating condition, an increase of 1.8°C was adopted in the model.

4.3.4. Valves

The AHU includes hydraulic valves that control water flow through the coils and a valve to control outdoor air flow.

Valves’ mass flow as a function of valve position is modelled by fitting an adjusted error function to measurements. The adjusted error function was calculated according to

$$m = \frac{\text{erf}(\frac{x - b}{a})}{c + d}, \quad (34)$$

where erf($x$) was calculated according to

$$\text{erf}(x) = \frac{2}{\sqrt{\pi}} \int_0^x e^{-t} \, dt. \quad (35)$$

The coefficients determine the slope of the linear part ($a$), horizontal displacement ($b$), amplitude ($c$), and vertical displacement ($d$).

Figure 10 shows the valves’ characteristic curves of the HT-CC (a), LT-CC (b), LT-HC (c), and fresh air supply (d). Especially the valve of the LT-HC shows a narrow control range: Mass flow is determined by only 20% of the control range.

4.4. Control algorithms

AHU’s components were controlled by PI-controllers implemented in BMS Priva. PI-controllers were modelled exactly according to their implementation in the BMS. Inputs to PI-controllers are converted from analogue to digital. This was modelled by a zero-order-hold with a sampling time of 10 s. Moreover, PI-controllers were modelled as being ‘ideal’, and ‘discrete time’ with a sampling time of 10 s. ‘Backward Euler’ was adopted as the integration method. Hence, the $z$-transform of the transfer function is

$$C(z) = K_r \left(1 + \frac{t_i}{t_s} \frac{z}{z - 1}\right), \quad (36)$$

where $K_r$ is the proportional gain, $t_i$ is the integration time, and $t_s$ is the sampling time.

Sampling times for all other components were set to ‘inherited’; so Simulink may adjust the simulation time step for each part of the model separately: model components that have large time constants may be updated at larger time steps, whereas fast-responding components are updated more frequently, providing a balance between computational efficiency and accuracy.

Because the control range of the valves is 0–100%, the lower and upper saturation limits of the PI-controllers were set accordingly. Anti-windup was used to prevent overflow of the integrator using ‘back-calculation’.

Although PI-controllers are readily available in Simulink, an important functionality was added: The release module. This release module decides if a controller is released to control its component. This prevents unwanted simultaneous controlling, for example, simultaneously controlling the heating and the cooling coils. This is particularly important to many museums controlling $T_i$ and RH, very strictly. The release module comprises the following rules to release a component: (i) the ‘watched component’ is not released AND the error (deviation from setpoint) should be larger than a preset threshold value, (ii) rule number ($i$) should be TRUE longer than a preset delay.

To unrelease: (i) the error should be smaller than a preset threshold value AND the controller’s action is decreased to 0%; (ii) rule number ($i$) should be TRUE longer than a preset delay. The module outputs ‘1’ if the component is released and ‘0’ if the component is not released.

The PI-controller blocks are preceded by a zero-order-hold block to model the AD-conversion, and a dead-zone block preventing excessively actuation of valves.

As illustrated in Figure 10, some valves have a large control region that has no influence on the outcome, for example, the water mass flow remains zero when controlling the heating coil valve from 0% to 20%. Therefore, a start value may be set in the BMS for the PI-controllers. For example, a start value of 20 ensures that the valve starts at 20% if released. In the model, this is implemented as the initial value of the integrator: ‘start value/$K_r$’. A falling signal of the release module, that is, from released ‘1’ to unreleased ‘0’, resets the integrator.

Figure 11 shows the validation of the PI-controller model of the steam humidifier ($K_r = 2$, $t_i = 180$ s, dead
Figure 10. Mass flow as a function of valve position: (a) HT-CC, (b) LT-CC, (c) LT-HC, (d) fresh air supply.

Figure 11. Validation of the PI-controller model of the steam humidifier. Controller’s inputs were zone’s RH (RH<sub>z</sub>) and setpoint (RH<sub>sp</sub>). Measured controller output (C<sub>meas</sub>) is compared to the simulated controller output (C<sub>sim</sub>). Zone = 0, start value = 0). The results show good agreement, with minor deviations due to the following. The controllers operate at a sampling time of 10 s. However, only measurement data with a sampling time of 16 min were available from the BMS. Interpolation was used to calculate intermediate values, predominantly affecting the integrator value. Nevertheless, the result is satisfying.

The same procedure has been applied to model the PI-controllers for heating, cooling, dehumidification, fresh-air-supply and supply air temperature. Figure A3 in the Appendix shows an overview of the control system in Simulink.

5. Results

5.1. IBES: dynamics of T<sub>i</sub>, RH<sub>i</sub>, and power demands

The IBES-model’s ability to reproduce the dynamic behaviour of the building and AHU-components is important for accurate energy predictions. This is particularly true in the case of large hourly variations of internal heat and moisture loads as in this case study. Figure 12 compares measurements to simulation results of the following signals: indoor air temperature, relative air humidity, thermal powers for heating (LT-HC), cooling (HT-CC) and dehumidification (LT-CC), and electrical power consumption for humidification. Fluctuations of T<sub>i</sub> and RH<sub>i</sub> are rather small because the indoor climate is conditioned very strictly to maintain 21°C and 50% RH. Therefore, it is hard to compare simulated and measured T<sub>i</sub> and RH<sub>i</sub>. Nevertheless, the results show that the fluctuations’ frequency and magnitude agree well. Heating, cooling, humidification and dehumidification powers include much more dynamics. The heating (LT-HC) and cooling (HT-CC) powers show a strong correlation with daily patterns of internal heat gains, that is, occupants and lighting. Notice that the rate of change varies among the simulated powers and agree well with measurements.

Figure 13 provides a detailed overview of the hygrothermal processes in the exhibition hall’s AHU in the case of dehumidification (LT-CC) and post-heating (LT-HC). Figure 13(a) shows measurement results and Figure 13(b) shows simulation results obtained with the IBES-model for a typical week in summer.
5.2. Measured compared to simulated energy demand

The measured energy demand was compared to the simulated energy demand by the IBES-model (see Figure 14(a)). Heating (LT-HC), humidification, and ventilation energy demands agree well. In case of dehumidification demand, the HT-CC precools the air under certain conditions and the LT-CC continues the dehumidification process by deep-cooling: the sensible part of the dehumidification process is partly accounted for by the HT-CC (Figure 14(a), blue hatched area). This has not been implemented in the simulation model, in which the HT-CC is only used for controlling $T_i$ and the LT-CC is used for controlling RH$_i$. The reason to do so is to illustrate clearly what share of the total energy consumption may be attributed to heating, cooling, humidification and dehumidification in such a museum building as the Hermitage Amsterdam.

Figure 14(a) (red hatched area) shows the heating energy used to post-heat the air after dehumidification. So, the vast majority of the total annual energy consumption pertains to dehumidification. Figure 14(b) shows the total energy consumption for heating (LT-HC), cooling (HT-CC and LT-CC), humidification, and fan. The relative simulation errors are shown in Figure 14(c): heating energy was underestimated by 3%, cooling energy was overestimated by 7%, and energy for humidification was underestimated by 6.5%. Fan energy was accurately reproduced because the fan is operated at a fairly constant rotation speed and the power consumption, known from measurements, was directly implemented in the model (overestimated by 2%).
Figure 13. The hygrothermal processes in the exhibition hall’s AHU in the case of dehumidification (LT-CC) and post-heating (LT-HC): (a) measurements, (b) simulation results using the IBES-model.

Figure 14. (a) In the case study building, the HT-CC was also used to precool the air in case of dehumidification demand (blue hatched area), but in the simulation model only the LT-CC was used for the dehumidification process. Heating demand was predominantly needed for post-heating after dehumidification (red hatched area). (b) The total energy consumptions for heating, cooling, humidification, and fan. (c) Simulation errors relative to measurements for energies specified in (b).

Figure 15. (a) Predicted annual energy consumption by ZABES and IBES-models. The red hatched area indicates post-heating after dehumidification. (b) Relative prediction errors (ZABES-model compared to IBES-model).

5.3. ZABES vs. IBES

To show the necessity of a simulation model including dynamic models of AHU-components and accurate models of control algorithms, the predicted energy consumptions of the IBES-model and ZABES-model are compared (see Figure 15(a)). Figure 15(b) shows relative errors of the ZABES-model compared to the IBES-model.

A small part of the shown deficiencies may be attributed to the control of the outdoor air supply. Both the ZABES-model and the IBES-model include CO₂-controlled outdoor air supply, but the ZABES-model relies
on an ideal control, that is, providing the required airflow instantaneously to keep the CO₂-level at the desired maximum level of 1000 ppm, whereas the IBES-model includes the dynamics of the control algorithm and control valve, resulting in slightly higher CO₂-levels than the setpoint of 1000 ppm (see Figure 5).

Comments to dehumidification: Dehumidification plays an important role during a large part of the year because the museum requires a steady indoor RH in combination with significantly high internal moisture sources from occupants. This high dehumidification demand is accomplished by an AHU that dehumidifies via deep-cooling, resulting in excessive annual cooling energy consumption. Moreover, the case study museum employed a high ACR of 7.5 h⁻¹, resulting in the fact that the humidity of recirculation air deviates marginally from the desired level, for example, by 0.3 g/kg. Nevertheless, this recirculation air first needs to be cooled down to the dew point temperature before the actual dehumidification commences. This leads to an incongruous ratio between the sensible cooling part and latent cooling part of dehumidification. The result is that the ZABES-model underestimates dehumidification energy demand by 90% compared to the IBES-model.

A comment to heating: The IBES-model predicts much more heating than the ZABES-model, caused by post-heating after dehumidification (Figure 15(a) red hatched area). A comment to cooling: The IBES-model predicts less cooling than the ZABES-model because dehumidification already provided a large part of the cooling. A comment to humidification: The ZABES-model underestimates the energy consumption for humidification by approximately 40%. This is partly caused by the steam humidifier’s efficiency (energy is partly wasted by heating the air), but also because the ZABES-model is simply based on energy balance equations, whereas the IBES-model includes the effect of system dynamics and settings of PI-controllers (aggressive or robust).

5.4. Energy impact of occupants

The museums’ occupants include visitors and staff. To demonstrate the influence of occupants’ presence, the ZABES-model and IBES-model have been simulated by including occupants and by excluding occupants. Excluding occupants results in less internal heat gains, only heat gains of lighting systems remain, and this eliminates internal moisture gains.

Figure 16 shows results of the ZABES-model. Heating and cooling energy demands are influenced by occupants’ sensible heat loads, whereas humidification and dehumidification energy demands are influenced by occupants’ latent heat loads, that is, moisture gains. The results are intuitive (see Figure 16(b)): excluding occupants results in higher heating and humidification demands, and lower cooling and dehumidification demands.

Figure 17 shows the results of the IBES-model. The results in Figure 17(b) appear to be counter-intuitive: excluding occupants (i.e. less internal heat and moisture loads) results in lower heating demand, higher cooling demand, lower dehumidification demand and a negligible effect on humidification demand. The explanation for the odd differences in heating and cooling demands lies in the fact that dehumidification is realized by deep-cooling and that most heat is needed for post-heating after dehumidification (see Figure 17(a)). The results are actually realistic: excluding occupants eliminates internal moisture gains and consequently decreases dehumidification demand, that is, less deep-cooling. In turn, this results in significantly less post-heating pertained to dehumidification. Additionally, less dehumidification demand results in more sensible cooling demand, since most cooling demand was covered by deep-cooling in case of dehumidification.

The explanation for the almost unchanged humidification demand pertains to the CO₂-controlled fresh air supply. If occupants are included, their CO₂-production eventually results in fresh air supply via the automatically controlled outdoor air valve. Occupants’ moisture gains
contribute to maintaining indoor RH in winter. However, this is counteracted by cold dry air via fresh air supply. If occupants are excluded, the fresh air supply valve remains closed due to the absence of CO₂-production. So, there is no benefit of occupants’ moisture gains in winter, but there is also no counter effect of dry fresh air supply. In the end, humidification demand is almost equal to the case in which occupants are excluded. In summer, both occupants’ moisture gains and fresh outdoor air supply result in higher dehumidification demand. So, dehumidification demand is significantly affected by occupants as shown in Figure 17(b).

6. Discussion and conclusions

The hygro-part of buildings is still underrepresented in BES-programs illustrated by the limited occurrences of the words ‘humidity’ and ‘moisture’ in overview papers, for example, in Crawley et al. (2008). Examples of software packages capable of creating ZABES-models including thermal and moisture models are EnergyPlus’ Purchased Air Mode, IES VE’s combination of ModellIT and ApacheSim, and TRNSYS’ Building Model ‘Type 56’. A few packages have been dedicated to detailed combined hygrothermal simulations, for example, PowerDomus, BSim, HAMBASE, and WUFI. IBES-models may be developed for example using EnergyPlus’ packages for Integrated Simultaneous Simulation, IES VE’s ApacheHVAC, TRNSYS’ comprehensive Type-library, and Modelica.

The high level of transparency and flexibility of the MATLAB Simulink environment proofed to be important throughout this study and even indispensable for the exact modelling of the controllers: Small errors in the controller model resulted in large deviations from measurements. Only a few dedicated simulation programs offer this flexibility. An important step forward in the BES-field may be the development of Modelica, an object-oriented approach offering maximum flexibility and transparency (Wetter 2009). Moreover, Modelica proofed to be very suitable for the integral simulation of the building, HVAC systems and control loops (Kim, Braun, and Wetter 2013).

This study has demonstrated that the simulated energy consumption by ZABES-models may deviate tremendously (underestimated by 84% in this case study) from the realized energy consumption of buildings including humidity control. Moreover, the results show that the detailed IBES-model was able to simulate the realized energy consumption much more accurately: within 10% for heating, cooling and dehumidification, and humidification. However, a comprehensive measurement campaign was needed to provide a myriad of data for model validation and calibration to reach the simulation accuracy obtained by the IBES-model. CO₂-data have been used intensively to obtain an occupant’s profile and helped to model infiltration, ventilation, and interzonal airflow. In general, developing the IBES-model was very labour-intensive, paying much attention to ventilation, infiltration, interzonal airflows, heat and moisture gains by occupants, validation of AHU-components, and dynamic behaviour of controllers.

The impact of occupants was taken into account by including an hourly occupants’ presence profile covering a period of one week. The mean hourly presence was determined via CO₂-level analysis. Both heat and moisture gains were included as a function of indoor air temperature. The resulting occupant model may be characterized as deterministic, whereas recent advances in research include occupant behaviour via stochastic models, taking into account uncertainties and varying presence over time (Stoppel and Leite 2014; Yan et al. 2015). However, a standard deterministic presence profile will suffice for most museums because visitors do not interact with the building and presence may be predicted well (Hoes et al. 2009).

The energy impact of occupants in the case study building has been demonstrated using the ZABES-model and the IBES-model. The resulting occupants’ energy impact as predicted by the ZABES-model intuitively seems to be
correct but is unrealistic for buildings including humidity control. The IBES-model showed that excluding occupants resulted in significantly less heating demand and significantly more cooling demand due to less dehumidification and less post-heating after dehumidification. This holds for well-insulated buildings with high internal gains and humidity control.

The unrealistic predicted energy demands of the ZABES-model, especially in the case of significant occupants’ impact, create the awareness that only a building model will not suffice for many cases, including temperature and humidity control, even in the design phase. For example, balancing calculations for combined (cooling and heating) ATES systems (Nordell, Snijders, and Stiles 2015) in which building simulations are used to assess the feasibility of an ATES system by calculating the ratio between annual heating and cooling demand. The results show that relying on ZABES-models may yield erroneous results and may have far-reaching consequences. Energy Service COUntracts (ESCOs) (Sorrell 2007) provide another example: the ability to predict annual energy consumptions close to realized annual energy consumptions is highly important to make ESCOs feasible and work.

The following may be concluded from this study and holds for buildings including both temperature and humidity control:

- ZABES-models (energy calculations based on the zone air’s energy balance) may provide unrealistic results. The study enabled to quantify the energy simulation error for a case study museum. The total annual energy consumption was underestimated by 84% in this case study.
- The IBES-model has accurately predicted annual energy consumptions, with prediction errors less than 10%, and has captured the building’s and system’s dynamics well.
- Excluding occupants’ heat and moisture gains in the ZABES-model yielded intuitive, but unrealistic results. In contrast, excluding occupants’ heat and moisture gains in the IBES-model yielded realistic results: no occupants (i.e. less heat loads and no moisture loads) resulted in less heating and more cooling, related to the processes of dehumidification by deep-cooling and post-heating after dehumidification.
- IBES-models are essential for reliable energy predictions of buildings, including humidity control, taking the HVAC systems, control systems, and occupants’ impact into account.

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7. Nomenclature

- $A$ surface area [m$^2$]
- ZABES Zone Air Building Energy Simulation
- AHU air handling unit
- ACR air change rate [h$^{-1}$]
- ATES aquifer thermal energy storage
- BES building energy simulation
- BMS building management system
- $C$ heat capacitance [J/K], concentration [kg/m$^3$] or [ppm], controller output [%]
- $\dot{C}$ heat exchange rate [W/K]
- $C_p$ specific heat [kJ/kg K]
- $G$ production rate [kg/s]
- $h$ surface heat transfer coefficient [W/m$^2$ K], specific enthalpy [kJ/kg]
- HT-CC high temperature cooling coil
- HVAC heating ventilation and air conditioning
- IBES Integral Building Energy Simulation
- $K_v$ flow resistance coefficient [kg/m$^3$]
- $K_i$ controller’s proportional gain
- $L$ thermal coupling coefficient [W/K]
- LT-CC low temperature cooling coil
- LT-HC low temperature heating coil
- $\dot{m}$ mass flow rate [kg/s]
- $N$ count [-]
- NTU number of transfer units [-]
- ODE ordinary differential equation
- $P$ power [kW], pressure [kPa]
- PI proportional and integral gain
- $r$ ratio [-]
- $R$ total thermal resistance [K/W]
- RH relative humidity [%]
- $t$ time [s]
- $T$ temperature [°C]
- $w$ humidity ratio [kg/kg]
- $V$ volume [m$^3$]
- $\dot{V}$ volume flow rate [m$^3$/s]
- $\varepsilon$ heat transfer effectiveness [-], simulation error
- $\rho$ density [kg/m$^3$]
- $\eta$ thermal fin efficiency [-]
- $\Phi$ heat loss [W]

Subscripts

- a air, air side
- c coil
- cross cross sectional
- cv convection
- evap evaporation
- fin individual fin
- i indoor, inlet, integration
- inf infiltration
- intz interzonal
- meas measured
- mech mechanical ventilation
- o outdoor, outlet
- occ occupants
- r radiation
- s surface, sampling
- sat saturation
- sim simulated
- sp setpoint
- st steam
References
Appendix
A detailed overview of the IBES-model in Simulink is provided in Figures A1–A4.

Figure A1. Overview of the IBES-model in Simulink, including the S-function for the ZABES-model (HAMBASE).
Figure A2. Overview of the AHU in Simulink. See Section 4.3.1 for the underlying equations of the coils.
Figure A3. Overview of the Control System in Simulink. See Section 4.4 for a description of the composition of PI-controllers.
Figure A4. Hygrothermal coupling of the AHU and the ZABES-model in Simulink.