# HEAT VENTILATION AND AIR CONDITIONING MODELLING FOR MODEL BASED FAULT DETECTION AND DIAGNOSIS

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## ABSTRACT

This paper presents a library of simplified, yet accurate, physical models of the different components that can be found in a typical air-handling unit. Models development was focused on high accuracy with low computational cost aiming at the use of the library for real time applications like fault detection and diagnosis.

Model library was developed to reduce to the minimum the initial data needed for setting up a simulation model. The data needed is commonly found in the datasheets provided by the manufacturer. A real data case study is also presented, compared, and thoroughly discussed.

## INTRODUCTION

Energy efficiency and greenhouse gases emission reduction in buildings (40% of total energy consumes in EU and US) are significant challenges presently faced by the scientific community. Great effort is concentrated on the reduction of energy consumption in buildings both for economic and environmental factors included in the EU2020 objectives (EU 2010). Since heating ventilation and air conditioning (HVAC) systems represent the biggest share (around 50%) of the energy consumption in the buildings (Pérez-Lombard, Ortiz, and Pout 2008), they are among the primary targets for implementing energy efficiency measures in buildings. With this in mind, one of the main goals is to improve the efficiency of the existing HVAC systems while maintaining adequate comfort levels in the conditioned environment. Energy inefficiencies in HVAC systems are common; this is due to the presence of undetected failures in one or more of its components. However, due to a compensation made by the control algorithms of other elements belonging to the same AHU comfort conditions are well maintained even if energy is wasted and the faulty behaviour remains unknown until detailed analysis are carried out. In many cases, faults may be very difficult to identify and localise.

Different fault detection and diagnosis (FDD) methodologies have been developed, mostly based on expert knowledge to help identify the faulty condition and its source (Katipamula, Michael, and Brambley 2005). They define a set of modes under which the system can be considered to be faulty operation and then, develop and tune a set of rules that make possible to detect the existence of the fault and possibly help diagnosing its origin (House, Vaezi-Nejad, and Whitcomb 2001). A new trend in FDD is using accurate models of the HVAC systems that provide a base line for optimal operation supporting the detection of deviation from this optimum (Isermann 2005). However, models are often impractical for real time applications due to elevated computational resources needed for development and simulation.

This paper proposes a library of simplified, first principle models of the HVAC components typically found in a AHUs. Models were developed used the modelica language (Elmqvist 1978) using Dymola as software development tool.

## PROPOSED METHODOLOGY

The proposed methodology comprises three basic steps as shown on Figure 1. First, the development of the first principle models based on energy and mass balance equations representing the energy interaction between different mediums (mainly water and air) in the components of the AHU. The second step is to calibrate the models based on a novel procedure. Finally, the developed and calibrated models will be used for model-based real-time fault detection and diagnosis. The first two steps are presented in this paper while the last is part of the future work

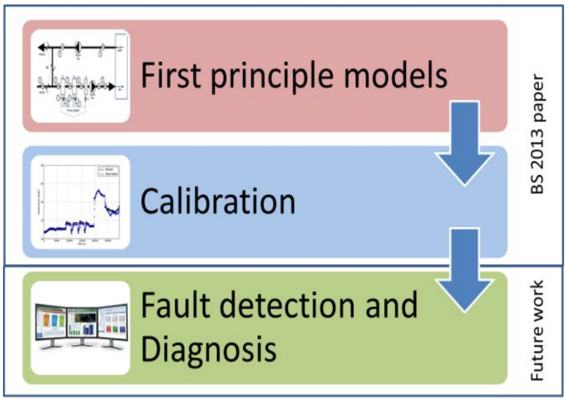


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*Figure 1Proposed Methodology*

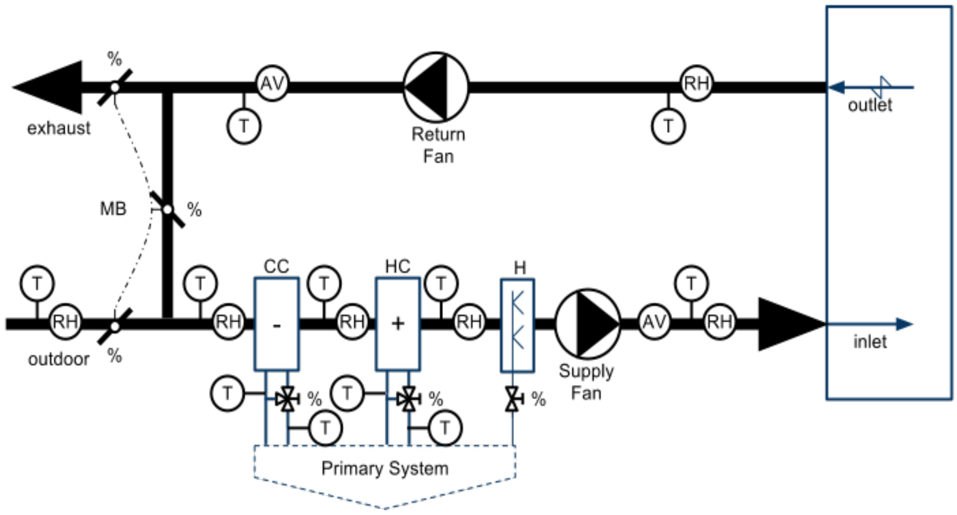


*Figure 2 Air handling unit schematic*

## FIRST PRINCIPLE MODELS

An AHU is comprised of a combination of the following components: mixing box, cooling and dehumidification coil, heating coil, humidifier, ducts, filters, and fans. In this work, the focus is the so-called active elements that are used for changing air temperature and humidity to match those required by the space being served. To this end, it is assumed that ducts and filters have negligible effects on air temperature and the fans just causes an air temperature increase and has no effect on the air humidity ratio.

Additionally, it assumed steady and adiabatic conditions and no frictional losses exist.

On Figure 2, a diagram showing the AHU configuration and its elements is presented. In the following sections, the components of the AHU will be described.

### Mixing box model

The model is composed of equations describing the energy and mass balance between mixing air streams and is based on the model presented by (Tashtoush, Molhim, and Alrousan 2005). The model comprises two inputs and one output. To calculate the mass flow rate of the air outlet it is used a simple mass balance equation:

$$mflow_O = mflow_1 + mflow_2 \quad (1)$$

In equation (1), $mflow_1$ and $mflow_2$ represent outdoor air and return air respectively, and are calculated according to the control signal called *damp_position* following the relations shown in equations (2) and (3):

$$mflow_1 = mflow_{I1}*damp\_position \quad (2)$$

$$mflow_2 = mflow_{I2}*(1-damp\_position) \quad (3)$$

where $mflow_{I1}$ and $mflow_{I2}$ represent the mass flow rates of the air streams coming from the dampers.

Finally, the output temperature and humidity ratio are calculated using the following energy balance equations:

$$mflow_1*T_1 + mflow_2*T_2 = mflow_O*T_O \quad (4)$$

$$mflow_1*W_1 + mflow_2*W_2 = mflow_O*W_O \quad (5)$$

### Heating coil model

The heating coil model calculates the outlet steady-state conditions in both, water and air sides, using equations derived from the conservation of energy principles and the definition of effectiveness in the classical eff-NTU method given by equations (6), (7) and (8) (ASHRAE 2009):

$$Q = C_a*(T_{aO}-T_{aI}) \quad (6)$$

$$Q = C_w*(T_{wI}-T_{wO}) \quad (7)$$

$$Q = eff*\min(C_a, C_w)*(T_{wI}-T_{aI}) \quad (8)$$

The effectiveness *eff* depends on the coil configuration (parallel flow, counter flow, or cross flow with both streams unmixed).

### Cooling and dehumidification coil model

This model is based on a model previously proposed and validated by (Lemort 2008). It calculates two operation regimes simultaneously, fully dry and fully wet. In both, the cooling capacity is calculated and the one with the higher cooling capacity is used as shown in equation (9).

$$Q = \max(Q_{dry}, Q_{wet}) \quad (9)$$

In dry regime, the outlet steady-state conditions in both sides (water and air) are calculated using a procedure similar to the one previously shown for the heating coil. Equations (10), (11) and (12) show energy balance for the dry regime.

$$Q_{dry} = C_a*(T_{aI}-T_{aOdry}) \quad (10)$$

$$Q_{dry} = C_w*(T_{wOdry}-T_{wI}) \quad (11)$$

$$Q_{dry} = eff_{dry}*\min(C_a, C_w)*(T_{aI} - T_{wI}) \quad (12)$$

In wet regime, the equations used are similar as dry regime but wet-bulb temperature is used instead of dry-bulb temperature. Also, it is assumed that the air is a perfect gas thus its enthalpy is fully defined by the wet bulb temperature. Next, it is shown the energy balance equations for the wet regime:

$$Q_{wet} = C_{fa}*(Twb_{aI}-Twb_{aOwet}) \quad (13)$$

$$Q_{wet} = C_w*(T_{wOwet}-T_{wI}) \quad (14)$$

$$Q_{wet} = eff_{wet}*\min(C_{fa}, C_w)*(Twb_{aI} - T_{wI}) \quad (15)$$

The parameter $C_{fa}$ form equation (15) is an auxiliary specific heat used in the wet regime given by equation (16) and that is used to determine the conditions of the output air by abstracting the coil as a semi-isothermal heat exchanger that is explained in detail by (Lemort 2008).

$$C_{fa} = mflow_a*(h_{aI}-h_{aO})/(Twb_{aI}-Twb_{aOwet}) \quad (16)$$

**Humidifier model**

In the humidifier model, the outlet air temperature is calculated using the weighted average of the incoming air and steam temperatures (Clark and May 1985).

$$T_{aO} = (c_s*mflow_s*T_s+c_{aI}*mflow_a*T_{aI})/(c_s*mflow_s+c_{aI}*mflow_a) \quad (17)$$

where $c_{aI}$ is the specific heat capacity of the incoming air. The $c_{aI}$ parameter depends on the humidity ratio given by:

$$c_{ai} = (c_a + c_s*W_{aI})/(1 + W_{aI}) \quad (18)$$

In order to determinate the outlet humidity ratio, it is assumed that all the steam is absorbed by the air which leads to the following equation:

$$mflow_s = mflow_a *(W_{aO}-W_{aI}) \quad (19)$$

However, the humidity ratio $W_{aO}$ could be higher than the humidity ratio at saturation $Ws$ times the saturation efficiency $n$ then

$$W_{aO}^{final} = \min(W_{aO}, n*Ws) \quad (20)$$

The $Ws$ is calculated using the psychrometric properties of wet air.

## CALIBRATION METHOLOGY

As mentioned before, the models were developed under the premise that a minimum set of data should be needed in order to run any simulation. A common set of the physical parameters should be established for every component model: specific heat capacities of water and air, atmospheric pressure and the saturation efficiency. The element-specific parameters are commonly provided by the manufactures. Table 1 shows the manufacturer data used for each component.

*Table 1 Parameters provided by the manufacturer*

| COMPONENT | PARAMETER |
|---|---|
| Mixing Box | No data required |
| Heating Coil | Nominal air input temperature<br>Nominal air output temperature<br>Nominal air mass flow rate<br>Nominal water input temperature<br>Nominal water output temperature<br>Nominal water mass flow rate |
| Cooling Coil | Nominal air input temperature<br>Nominal air input relative humidity<br>Nominal air output temperature<br>Nominal air output relative humidity<br>Nominal air mass flow rate<br>Nominal water input temperature<br>Nominal water output temperature<br>Nominal water mass flow rate |
| Humidifier | Maximum steam mass flow rate<br>Steam temperature |

The proposed calibration methodology starts with a component-by-component calibration using real operation data obtained from the facility's building management system (BMS) and ends with a final tuning of the assembled AHU. For the calibration procedure, instead of trying to adjust each of the component's parameters, the approach used is by assuming all the calibration can be done with the valve model explained below in this section.

There are two types of components according to the main heat transmission medium: those controlled with water mass flow rate and those controlled with air mass flow rate.

In the heating coil, cooling coil and humidifier, the air outlet temperature is controlled by water mass flow rate using valves. A control signal determines the valve's position. The mixing box's air outlet temperature is controlled by air mass flow rate.

Real valves have no lineal behaviour but they present hysteresis. To model the valve's hysteresis, several options can be followed, e.g. using on-off hysteresis, linear hysteresis and non-linear hysteresis. For the purposes of this research work, a linear hysteresis (Figure 3) was chosen since it produced a good trade-off between accuracy and simplicity. The chosen hysteresis model will still be a good representation of the real operation of the valve while it does not add much calculation burden to the model.

There are three parameters to calibrate. *mflowMAX* is the water mass flow rate when the control signal is equal to 1 (maximum opening position), *centHys* and *delta* characterise the hysteresis' curve.

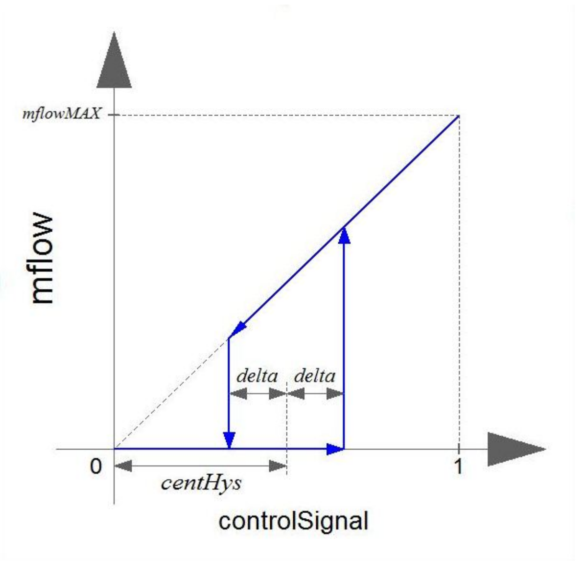


*Figure 3 Valve hysteresis*

The real data has to be carefully observed to find the maximum opening points and then the *mflowMAX* value is fixed in order to decrease the difference between real data and model results of the controlled variable in those points (temperature and/or humidity ratio).

To determine *centHys* and *delta*, the employed strategy was to find sharp changes in controlled variable. When the controlled variable has a sharp raise, the control signal coincides with a value equal to *centHys+delta*; controlled variable has a sharp decrease, the control signal coincides with the value equal to *centHys-delta*.

The mixing box presents a particular case since there are two considerations to keep in mind: there is not *mflowMAX* parameter to be found (nor any equivalent parameter) and the controlled variable is the air outlet temperature. With the above in mind, the calibration for the mixing box only follows the tuning of the *centHys* and *delta* parameters.

## CASE STUDY

The case study comprises the AHU already shown in Figure 2 and two real data sets; namely calibration data and validation data. Both data sets correspond to one week data. The calibration dataset was taken during summer operation while the validation data set was taken during winter operation (Figure 9). The data sets were extracted directly from the BMS and correspond to one-minute resolution data from each of the sensors shown in Figure 2. The calibration dataset was used to follow the calibration procedure previously mentioned. All the results presented in figures 4 to 9 and table 2 correspond with test performed using the validation data set.

The AHU serves a facility consistent of an audio laboratory of around 50 m$^2$, where strict conditions of temperature and humidity should be met. The building is located in the Republic of Ireland.

The parameters characterising each component are presented in Table 3 while the resulting calibration parameters for the valves are shown in Table 4. It is important to state that the real AHU did not have sensors on the outlet of the humidifier but only on the outlet of the fan. For this reason, during the calibration process, a simplified fan model was included in the humidifier model in order to be able to compare the simulated and measured air temperature and humidity according to a similar configuration.

Figures from Figure 5 to Figure 8 show simulation results pre and post calibration for each AHU component.

To qualify the goodness of the models, two metrics were calculated for each simulation result (ASHRAE 2002): coefficient of variation of the root-mean-square error (CV-RMSE) and the mean bias error (MBE). The results for the afore mentioned metrics are presented in Table 2.

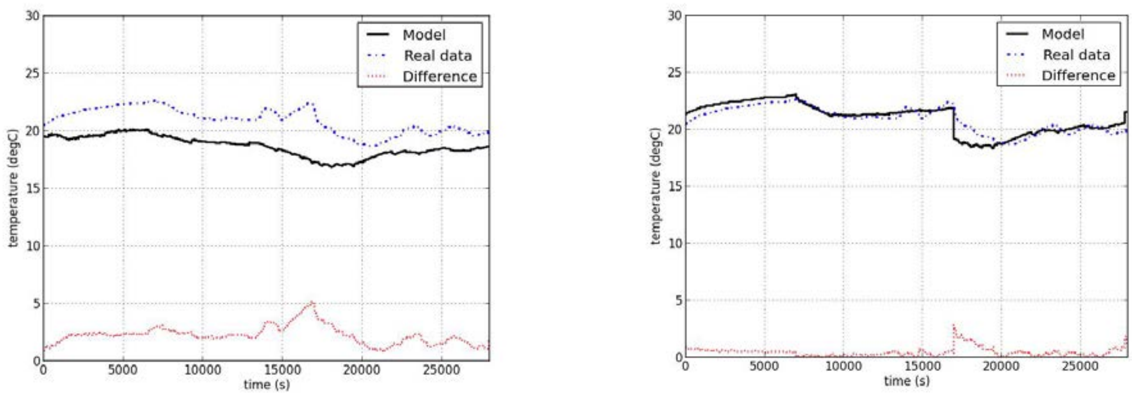


*Figure 4 Simulated supply air temperature vs. measured supply air temperature* for the ***whole AHU*** *model. Left: non-calibrated. Right: calibrated*

*Table 2 CV-RMSE and MBE pre and post calibration for each AHU component*

| ERROR (%) | MIXING BOX | HEATING COIL | COOLING COIL | HUMIDIFIER | AHU |
|---|---|---|---|---|---|
| Pre-calibration CV-RMSE | 9.32 | 5.80 | 9.13 | 26.67 | 11.37 |
| Post-calibration CV-RMSE | 2.60 | 4.33 | 5.04 | 4.10 | 2.86 |
| Pre-calibration MBE | 8.83 | 0.11 | 7.21 | 26.11 | 10.71 |
| Post-calibration MBE | 1.34 | 0.90 | 1.22 | 1.34 | 0.50 |

### Air Handling Unit Calibration

Once each component model was calibrated, the whole AHU model was assembled and simulated. Simulation results for the output air temperature for the whole AHU are presented in Figure 4 right. On the left hand side of Figure 4 it is shown how the AHU model behaves when none its components is calibrated and on the right hand side, the fully calibrated model can be seen. Figure 4 shows also the difference between measured temperature and temperature simulations results.

As per the individual components, the metrics CV-RMSE and MBE where calculated for the full AHU and the resulting values can be seen on Table 2.

On one hand, pre and post calibration figures of the whole AHU (Figure 4), show the significant improvement brought by the calibration process. In the figure, a substantial reduction in the absolute error can be perceived in almost all the points. However, at about 16000 seconds of simulation the error value peaks shortly at just over 3 ºC. This peak is due to the simple linearization of the valve model. To obtain a smoother curve, the valve model could have been approximated to a higher order, but this would have to be done at the expense of increasing the computational effort. On the other hand, by looking at Table 2, it can be seen how calibrated model is far better than no calibrated one regarding to CV-RMSE, which means a substantial decrease in the deviation of the results.

## CONCLUSION

In this paper, a set of steady-state simple models to simulate HVAC systems is presented. It consists of a mixing box model, cooling coil model, heating coil model, humidifier model and simplified fan model. These simple models can be used to study any typical AHU, by simulating the main energy and mass transfer interactions between components at a reduced computational cost. These models are suitable for real-time applications of controls and fault detection and diagnosis as well as hardware in the loop simulations.

A limited number of parameters are needed to set up models but they are commonly found in the manufacturer data sheet and ASHRAE fundamentals. This characteristic improves the usability of the models.

On the other hand, a simple novel procedure of calibration is shown. Using data collected from the building management system it makes possible reduce simulation errors beyond the limits suggested by international guidelines like (ASHRAE 2002).

## FUTURE WORK

Next step in this research is to improve the calibration algorithm by developing an automatic tool that could be implemented underpinned by machine learning.

In addition, a comparison with other libraries already validated will do, for example with modelica buildings library from the Lawrence Berkley National Laboratory at the United States (Wetter and Berkeley 2009) or the Matlab/Simulink HAMBase library (Van Schijndel 2007).

An exhaustive study of different real cases will be done in order to determine the feasibility and accuracy of the models for FDD. In particular, a set of experiments under faulty conditions will be carried out on the facility in order to compile comparative data for running FDD algorithms.

## NOMENCLATURE

| | | |
|---|---|---|
| *eff* | effectiveness | [1] |
| *Q* | heat transfer | [W] |
| *c* | specific heat capacity | [J/kg·K] |
| *C* | capacity flow | [W/K] |
| *n* | saturation efficiency | [1] |
| *mflow* | mass flow rate | [kg/s] |
| *T* | temperature | [°C] |
| *Twb* | wet buld temperature | [°C] |
| *W* | humidity ratio | [1] |
| *Ws* | humidity ratio at saturation | [1] |

**Subscripts**

| | |
|---|---|
| *a* | air |
| *I* | input |
| *O* | output |
| *w* | water |
| *dry* | dry regime |
| *wet* | wet regime |

**Functions**

| | |
|---|---|
| max(·, ·) | largest value between arguments |
| min(·, ·) | smallest value between arguments |

## ACKNOWLEDGEMENTS

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*Table 3 Model Parameters and physical quantities*

| MODEL PARAMETER | COOLING COIL | HEATING COIL | HUMIDIFIER |
|---|---|---|---|
| Nominal air input temperature (°C) | 25.0 | 6.3 | - |
| Nominal air input relative humidity (%) | 50.0 | - | - |
| Nominal air output temperature (°C) | 13.8 | 18.8 | - |
| Nominal air output relative humidity (%) | 88.0 | - | - |
| Nominal air mass flow rate ($m^3/s$) | 1.35 | 1.35 | - |
| Nominal water input temperature (°C) | 6.0 | 82 | - |
| Nominal water output temperature (°C) | 12.0 | 71 | - |
| Nominal water mass flow rate (kg/s) | 0.97 | 0.47 | - |
| Maximum steam mass flow rate (kg/s) | - | - | 0.0014 |
| Steam temperature (°C) | - | - | 80.0 |
| **PHYSICAL QUANTITY** | | | |
| Air specific heat capacity (J/(kg·K)) | 1006 | | |
| Water specific heat capacity (J/(kg·K)) | 4186 | | |
| Atmospheric pressure (Pa) | 101325 | | |
| Saturation efficiency | 1 | | |

*Table 4 Calibration Parameters*

| CALIBRATION PARAMETER | MIXING BOX | HEATING COIL | COOLING COIL | HUMIDIFIER |
|---|---|---|---|---|
| *mflowMAX (kg)* | - | 0.47 | 1.31 | 0.0077 |
| *centHys* | 0.55 | 0.25 | 0.13 | 0.0 |
| *delta* | 0.00 | 0.025 | 0.07 | 0.0 |
| Air temperature change by fan (ºC) | - | - | - | 1.5 |

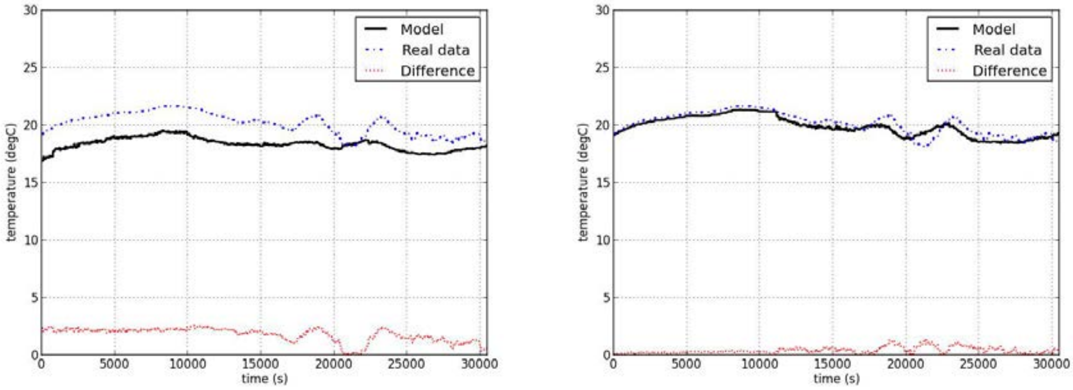


*Figure 5 Simulated air output temperature vs. measured temperature for the* ***mixing box*** *model. Left: non-calibrated. Right: calibrated.*

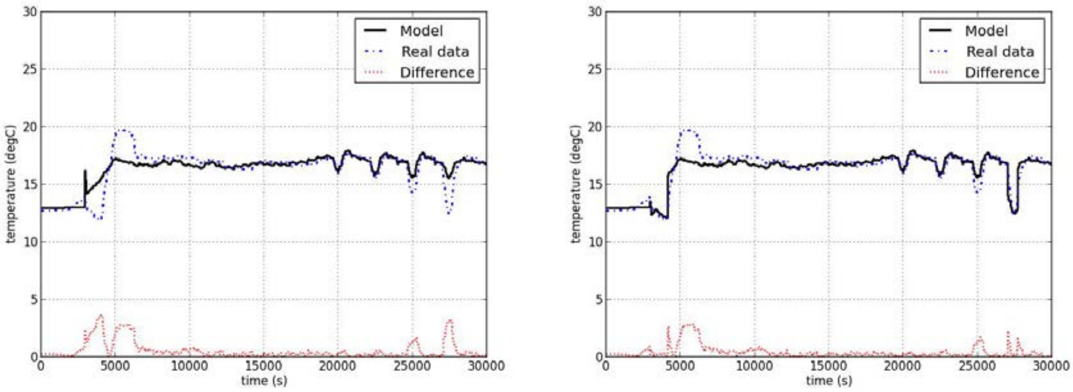


*Figure 6 Simulated air output temperature vs. measured temperature for the* ***heating coil*** *model. Left: non-calibrated. Right: calibrated.*

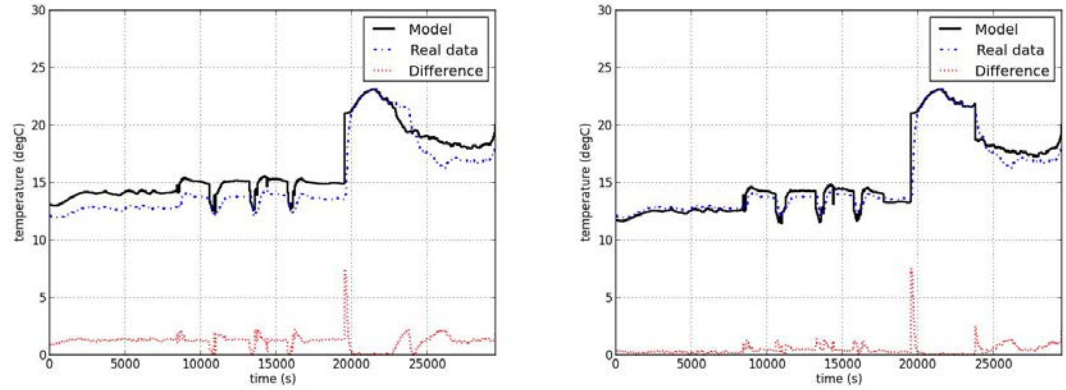


*Figure 7 Simulated air output temperature vs. measured temperature for the* ***cooling coil*** *model. Left: non-calibrated. Right: calibrated.*

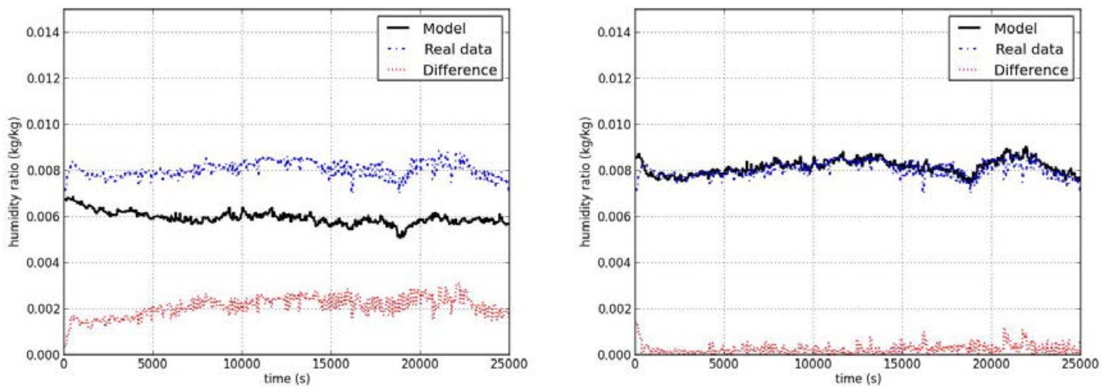


*Figure 8 Simulated air output humidity ratio vs. measured humidity ratio for the* ***humidifier*** *model. Left: non-calibrated. Right: calibrated.*

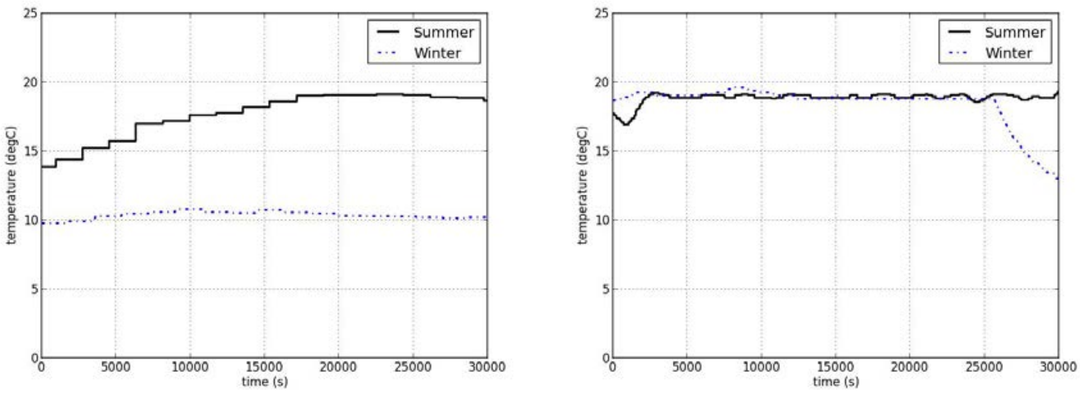


*Figure 9 Summer vs. Winter external temperature conditions. Left: Outdoor Temperature. Right :Zone Return Temperature*