An Investigation of the Impact of Building Entrance Vestibule on Indoor Humidity Using A Calibrated Multi-Zone Model for A Small Supermarket in Hot and Humid Climate

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Abstract

The automatically operating entrance doors in many retail buildings have been found to be main paths of air infiltration and this uncontrolled airflow through the automatic entrance doors is a significant factor of indoor moisture problems. This study proposes the calibration method to numerically investigate realistic transient indoor humidity ratios via a calibrated multi-zone model. The calibrated results is used to further calculate the amount of moisture contents transported through the automatic entrance doors with and without a vestibule in a retail building located in a hot and humid climate. The results from the calibrated model coupled with an updating adsorption mechanism agreed well with the measured humidity ratios. The indoor humidity ratio was decreased by approximately 9 percent when a door vestibule was added. The door vestibule can control indoor humidity levels within thermal comfort condition. Future studies should acquire more accurate number of occupants to reduce the uncertainty from the calculation of indoor moisture source.

Keywords: Humidity control, Multi-zone model, Automatic entrance door, Calibration, Entrance vestibule

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1. Introduction

High indoor humidity level causes unsatisfied thermal comfort and indoor mold growth problems. High indoor humidity significantly affects the evaporation rate on body skin and negatively impacts on human thermal sensation (Arundel, Sterling, Biggin & Sterling, 1986; Jing, Tan & Liu, 2012). For thermal comfort conditions, ASHRAE standard 55 (2004) required the limit humidity level for thermal comfort should not be above 0.012 $g_{water}/Kg_{dry\ air}$. About an influence of indoor humidity on indoor mold growth, Abe (1993) and Clarke et al. (1999) studied an impact of indoor environmental conditions, specifically temperature and relative humidity, on ability of indoor mold growth. Abe (1993) presented that mold growth ability increased significantly as indoor humidity level increased. At temperature 25°C with 70% relative humidity, mold germination time ranged from 8 to 30 days and the mold germination time was within 24 hours when relative humidity exceeded 85%. The recommended range of indoor humidity level for control indoor mold growth should be lower than 65-70 percent (American Society of Heating, Refrigeration and Air Conditioning Engineering [ASHRAE], 2008; Abe, 1993; Clarke et al., 1999). Based on Sterling et al.'s study (1985), ASHRAE (2008, 2010) recommended the maximum limit of indoor relative humidity of 65% at normal room temperatures for indoor mold growth control whereas Abe (1993) and Clarke et al. (1999) suggested that the indoor relative humidity should be below 70% ($g_{water}/Kg_{dry\ air}$) at room temperature of 25°C. Retail buildings have been reported as having high air infiltration rates when compared to those in other public buildings. The automatic entrance door of retail buildings is considered as the main path of air infiltration. Existing studies shows that airflow rate through operating automatic entrance doors ranges from 20 - 1,250 L/s.m^2 (Simpson, 1936; Kohri, 2001; Cho et al., 2010; Bennett et al., 2012; Jareemit, 2014; Srisuwan & Varodompun, 2013, pp. 5-20). Due to the high infiltration rate through the operating automatic entrance doors, retail building might have a risk of indoor mold growth and unsatisfied thermal comfort relative to high indoor humidity level.

A multi-zone model, CONTAMW, was developed from conservation of mass, energy, and contaminant equations (Walton & Dols, 2005). Such a model is specially used to analyze transient airflow rates and contaminant concentrations between zones driven by mechanical ventilation, local wind, and buoyancy effect. However, the accuracy of the simulation results relies on the accuracy of the model input (Persily & Ivy, 2001). One previous research analyzed the potential of using a multi-zone model with mass balance equations of water vapor sources and sinks (Emmerich, Persity & Nabinger, 2002). The results show that the simulation did not agree well with the measured data due to the inaccurate data of moisture storage and generation. In addition, the model was performed with a limited number of houses. Therefore, to understand the impact of operating automatic entrance doors on indoor humidity level, the objective of this study is to develop and validate a calibrated multi-zone model to calculate time-dependent indoor moisture content based on Emmerrich's procedure (2002). The base case model used for the calibration indoor humidity in this paper was developed from the calibrated model for ventilation rate including air infiltration through the automatic entrance doors in an earlier project (Jareemit, 2014).

2. Building Description

A building case study is a one-story grocery store with retail area of 780 m^2 and total building volume of 5,390 m^3. The store is located in Austin, TX which is in hot and humid climates, (zone 2a). This building contains retail and storage area. There are two types of openings in this retail store: swinging door, and loading door. The main entrance is a swinging door without a vestibule. Loading doors are found at service area. Most of interior doors are double swinging door. During the field experiment, the ventilation system was operated at constant air volume with supply airflow rate of 3,400 L/s (±7% error), return air ratio of 90 percent and outdoor air ratio of 32 percent of supply airflow. Figure 1. presents the building layout, opening location, and number of rooftop units.

3. Background of the Simulation Model

Jareemit (2014) developed a calibrated multi-zone model to investigate transient airflow rates through automatic entrance doors in two retail stores. The calibrated differential pressure potentially provided realistic calculated infiltration rate through the automatic entrance doors. In order to correct airflow rate through opening based on customer use, the previous study updated flow coefficient based on Yuill's investigation (2000). The leakage airflow element in the model setting was calculated by Equation 1.

$$Q=C\Delta P^{0.5} \qquad (1)$$

where Q is the airflow through the opening, L/s. ΔP is the pressure difference across the automatic entrance doors, Pa. *C* is the flow coefficient, L/s.m^2.Pa$^{0.5}$. The calculation of *C* is performed by multiplying the airflow coefficient, C_a, by

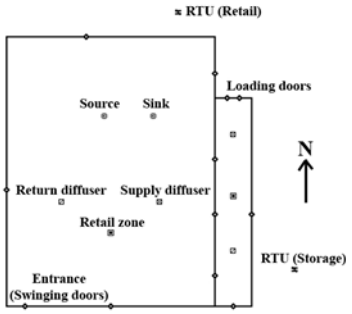


Figure 1. Building layout presents zones, location of openings, opening types, and number of ventilation units.

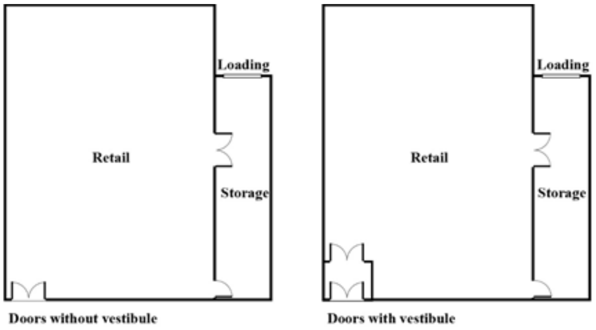


Figure 2. Layouts of entrance doors with and without vestibule used in the simulation model.

the total door area of 4 m^2. The airflow coefficient, C_a, is a function of the number of people using the automatic entrance doors in each hour. The airflow coefficient, C_a, for the swinging doors with a vestibule and the door without a vestibule was calculated from the chart shown in Figure 3.

The calibrated model developed by the previous study (Jareemit, 2014) provides realistic results pertaining to air infiltration through the automatic entrance doors and total building ventilation rate. However, this model cannot provide realistic results of indoor moisture concentration since the model lacks of moisture source and sink operations. Consequently, this present study further develops a calibrated multi-zone model to investigate reliable transient indoor humidity levels.

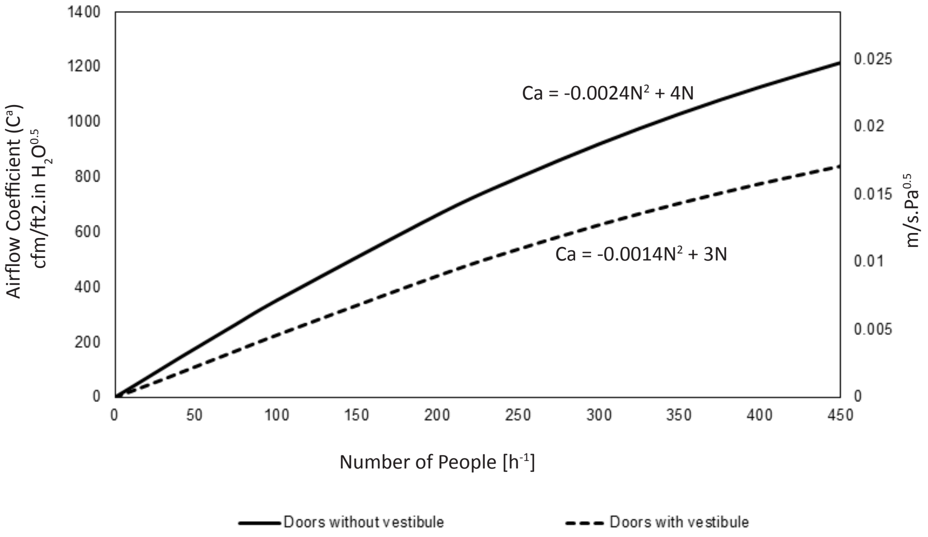


Figure 3. Experimental correlations for the airflow coefficient and people passing through the door per hour for automatic swinging doors with and without a vestibule (the data source is Yuill et al., 2000).

4. Research Methodology

In this study, the model was calibrated by matching the simulated indoor humidity ratio with the measured data performed during the store's operation hours (about 17 hours) on August 3rd, 2012 provided in ASHRAE RP-1596 (Siegel et al., 2013). The simulation was modeled using AMY (Actual Meteorological Year), which measured data, including ambient temperature, humidity ratio, wind direction, and wind speed, for Austin, TX, (zone 2a). The calibration procedure for indoor humidity ratio composed of the following four steps:

1. Without sink model with adjusted occupancy rate (Base case model)
2. Sink model with adjusted occupancy rate
3. Sink model with measured occupancy rate
4. Actual cooling coil efficiency with measured occupancy rate

In the base case model (step 1), occupants were considered only an indoor moisture source. An average rate of moisture geneated by an adult was 8×10^{-5} kg/s (Krama & Karagiozis, 2004). This number was multiplied with the occupant schedule. In ASHRAE RP-1596 project (Siegel et al., 2013), the field data collection for this building was performed from August 1st to 4th, 2012 (4 days). However, the ventilation rate and most contaminant concentration measurements was performed only one day on August 3rd, 2012. The measured occupant rate is the hourly transaction rate recorded at the check-out counter recorded by the store's computer system. The relative difference of the occupancy rate recorded in different days ranged from 1-12%. However, it was considered that this transaction rate recorded at check-out counter might not be accurate because some people might not purchase goods and just walk past the counter. Therefore, this hourly transaction rate was multiplied by an adjustment factor of 1.3. The factor is a ratio of hourly traffic passage through the automatic entrance doors to hourly transaction rate at check-out counter. The hourly traffic passage through the automatic entrance doors was measured by a thermal image people counter installed at ceiling above the exit doors. The ratio of 1.3 was calculated based on the field investigation in Site MiP in the ASHRAE RP-1596 project where these two parameters were measured. Figure 4. compares the measured hourly transaction rate and calculated traffic passage on August 3rd, 2012.

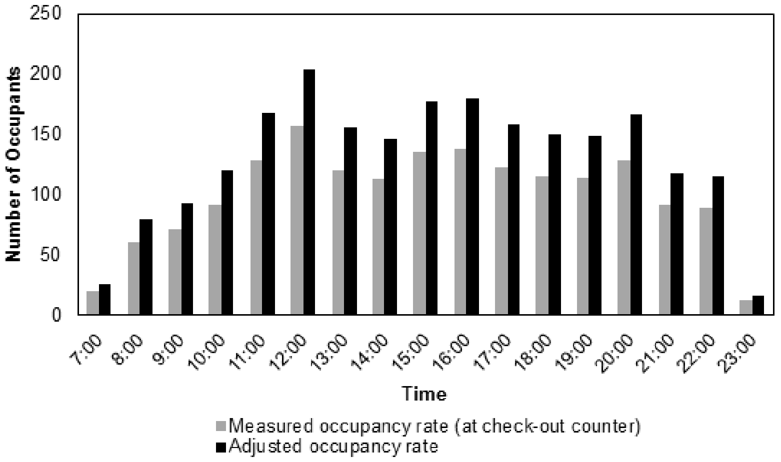


Figure 4. Occupant schedule on August 3rd, 2012.

The multi-zone model does not provide a function for the dehumidification process at a cooling coil. Consequently, an equation of gas removal in filter was used to calculate a cooling coil efficiency to remove moisture content in supply air. The calculation of the coil efficiency used Equation 2.

$$\eta = 1 - \frac{C_{out}}{C_{in}} \quad (2)$$

where C_{out} is humidity ratio of the air leaving the cooling coil, g_{water}/kg_{dryair}, η is the cooling coil efficiency, and C_{in} is humidity ratio of the air entering the cooling coil, g_{water}/kg_{dryair}.

In this step, the cooling coil efficiency used in the calibration model was firstly assumed an ideal coil with constant water flow 'filter' based on the assumption used in Emmerich et al.'s study (2002). The calculation of the ideal coil efficiency, η, used a psychrometric chart when a setting of the cooling coil temperature was assumed 12°C. Humidity ratio of air leaving the coil (C_{out}) was 0.0105 g_{water}/kg_{dryair}. The AMY (Actual Meteorological Year) measured data was used in the model calibration. Average outdoor air temperature calculated from the AMY measured data was 24°C with relative humidity of 61 percent. The indoor temperature was controlled constant temperature of 25°C. The building brought outdoor air ratio of 32 percent of supply airflow. The moisture content of air entering the coil at mixing air plenum was 0.012 g_{water}/kg_{dryair}. Consequently, the design cooling coil efficiency used in the simulation model was approximately 19 percent.

In step 2, in addition to the dehumidification in the cooling coil, indoor moisture vapor dissolved on interior surfaces was accounted in the calibration process. For this process, the present study used the boundary layer diffusion controlled (BLDC) sink model shown in Equation 3:

$$S(t) = h\rho A\left(\frac{C_i - C_s}{K}\right) \qquad (3)$$

In the equation, h is the surface-average mass transfer coefficient, which is 0.72 m/h. ρ is the air density at boundary layer near adsorbent surface of 1.2 kg/m^3. The film density is an air layer between adsorbent surface and room The interior surface of walls and ceiling was 13 mm. thick gypsum board and the weight of gypsum board was approximately 10 kg/m^2 (Axley, 1991). *A* is the surface area of the adsorbent approximately of 8,000 m^2. Consequently, a mass of walls and ceiling is approximately 80,000 kg. C_i is the concentration of water vapor in air and C_s is the concentration of water vapor in the adsorbent. *K* is the Henry adsorption constant or partition coefficient, which is assumed approximately 5 (Emmerich, Persily & Nabinger, 2002).

In step 3, the model setting was the same as those in step 2, except that the measured occupancy rate (transaction number at check-out counter) was used instead of the adjusted occupancy rate.

The issue of using the ideal cooling coil efficiency of 19% in step 2 and 3 was that the humidity ratio of the air leaving the cooling coil is varied depending on the humidity ratios of the air entering the coil. Practically, the air condition leaving the coil should be fixed at constant value even though the air condition entering the coil varies. To solve this issue, the cooling coil efficiency was recalculated and varied depending on the air condition entering the cooling coil. In step 4, the model updated the calculated cooling coil efficiencies during the store's operation hours from 7:00 am. to 10:00 pm. on August 3rd, 2012 as shown in Figure 5. The relative difference of outdoor air condition used to calculate the coil efficiency in different days was within 8% for temperature and 12% for relative humidity.

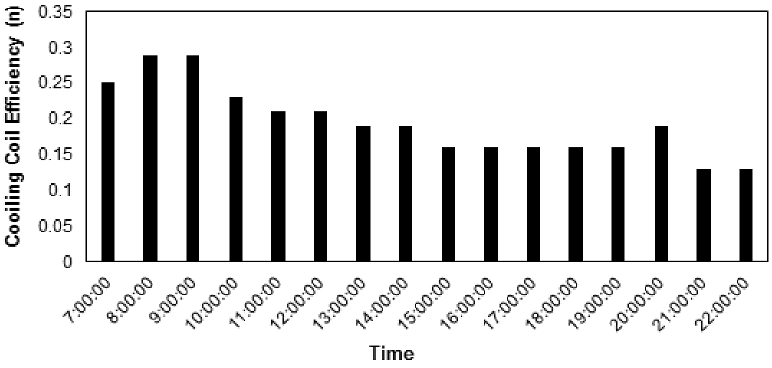


Figure 5. Calculated hourly cooling coil efficiency used in the model calibration in scenario 3.

The simulation results were compared with the measured data of indoor humidity ratio investigated in the ASHRAE RP-1596 project (Siegel et al., 2012). In the field measurement, the humidity sensors with accuracy of ±2.5% were used collect the data in two locations in the retail space. An evaluation of indoor air quality (IAQ) modeling can be found in ASTM D5157 (Standard Guide for Statistical Evaluation of Indoor Air Quality Models). The objectives of this evaluation method are (1) to compare the performance of two or more models for a specific situation or set of situations and (2) to assess the performance of specific models for different situations. The standard requires that quality of the simulation results should be evaluated with six statistical models as presented in Table 1.

Statistical variable	Satisfactory Range
Correlation coefficient	0.9 or greater
Regression slope	0.75–1.25
Regression intercept	25% or less average measured concentration
NMSE	0.25 or lower
FB	0.25 or lower
FS	0.5 or lower

Table 1. List of six statistical variables and satisfactory ranges required in ASTM D5157.

5. Results

Figure 6. presents the calibration results of indoor humidity ratios in each calibration step. In step 1, the model used adjusted occupancy rate (hourly transaction number) and the sink model was not accounted for the simulation. The results had a range of indoor humidity ratio from 0.011 - 0.016 $g_{water}/kg_{dry\ air}$. After the sink model was added in the model (step 2), the simulated humidity ratio had a good agreement with the measured data. The range of humidity ratio was from 0.011 - 0.013 $g_{water}/kg_{dry\ air}$. In step 3, the model used measured occupancy rate instead of the adjusted rate. The amplitude of simulated humidity ratio were matched with the measured data; however, the humidity ratios during the morning from 7 am. - 12 pm. were quite below when compared to the measured data. In step 4, the cooling coil efficiencies were updated in the model. The uncertinty of humidity ratio during the morning found in step 3 was decreased.

The quality of the simulation results of the each calibration step was evaluated with six statistical models provided in the ASTM D5157 Standard, as presented in Table 2.

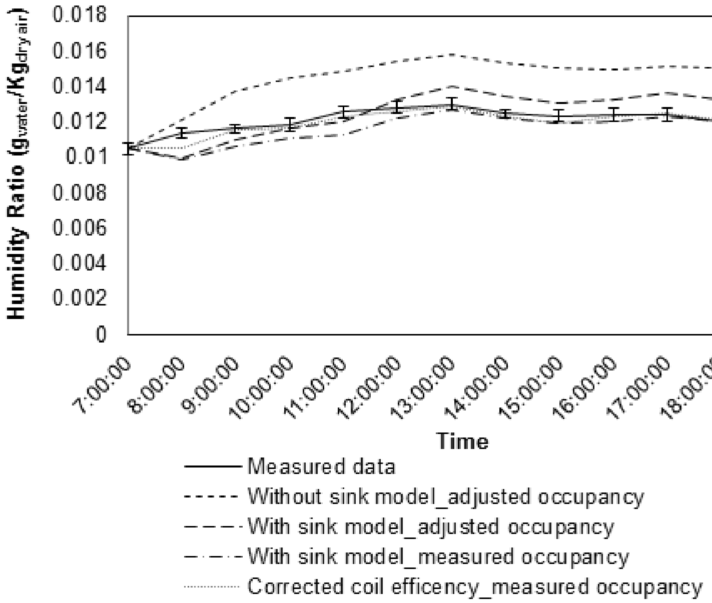


Figure 6. Calibration results of indoor humidity ratios.

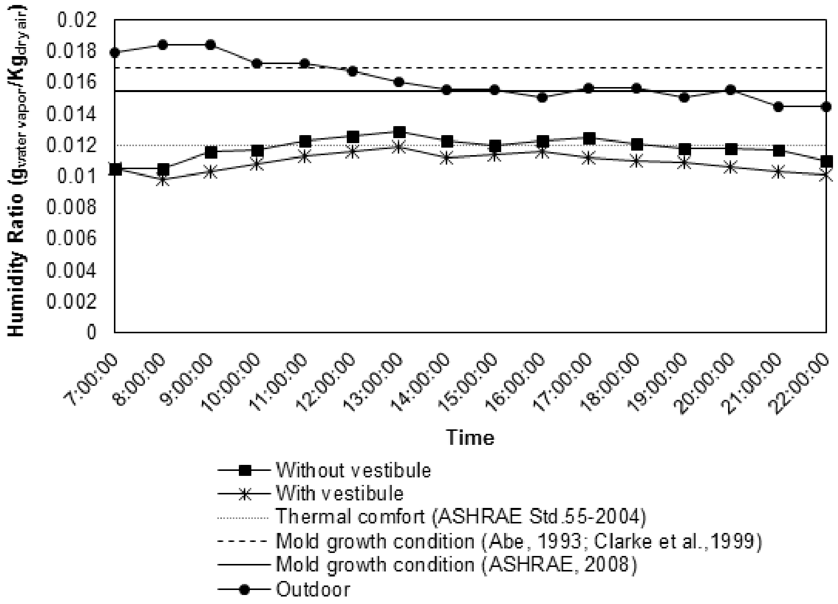
Table 2. Assessment of model performance required in ASTM D5157.

Statistical variable	No sink model	With sink model	Corrected coil efficiency
Correlation coefficient (>0.9)	*0.84*	*0.74*	0.94
Regression slope (0.75 to1.25)	*1.57*	0.78	0.99
Regression intercept (25% or less average measured concentration)	0.005	0.002	0.008
NMSE (<0.25)	0.03	0.003	0.0004
FB (<0.25)	0.18	-0.04	-0.012
FS (<0.5)	*1.69*	-1.62	0.23

When the multi-zone simulation was conducted without the sink model, the calculation of the correlation coefficient, the regression slope, and the index of bias (FS) were not within the satisfactory ranges required the ASTM D5157 Standard. The model quality was increased when the sink model was integrated in the model. All, except the correlation coefficient, met the standard requirement. The updated correction of cooling coil efficiencies improved the model quality when all statistical variables met the standard requirement.

After calibrating indoor humidity ratios, the study further used the model to investigate the effect of a door vestibule on control indoor humidity levels for thermal comfort and mold growth conditions. It was found that installing the door vestibule can control moisture contents from outside transport through the automatic entrance doors. In Figure 7, the calculated indoor humidity ratios for the case without the door vestibule ranged from 0.011-0.013 $g_{water}/kg_{dry\ air}$. For the case with installing the door vestibule, the indoor humidity ratio ranged from 0.0105-0.012 $g_{water}/kg_{dry\ air}$. Installing the door vestibule reduced the indoor humidity ratios by approximately 9 percent. Mold growth did not occur in both cases since the indoor humidity levels were lower than 0.016 $g_{water}/kg_{dry\ air}$ defined by ASHRAE (2008, 2010) and 0.018 $g_{water}/kg_{dry\ air}$ defined by Abe (1993) and Clarke et al. (1999). However, the case without the door vestibule had the indoor humidity levels above thermal comfort during 11 am. – 6 pm. Installing the door vestibule can help to control indoor humidity levels within thermal comfort condition.

Figure 7. Calibration results of indoor humidity ratios.



6. Discussions

The calibrated results show a good agreement with the measured data. However, there is large uncertainty found when store opened at 7- 8 am. This error might be from the delay of measured hourly transaction rate. Number of customer was counted when people paid the services at check-out counter. This number might not represent the customer number at the time period that they enter the store and check-out. Future studies should investigate more accurate customer numbers to reduce the uncertainty for the calculation of indoor moisture source.

In this study, the model calibration was performed only one day during the store's operation hours from 7 am. to 10 pm. Consequently, the application of this calibrated model is limited only this building case study and one day measurement. To examine the robust of the calibrated model for intended uses, the model needs to check repeatability under different days with various outdoor and indoor conditions or test in a few more buildings.

7. Conclusions

This study investigates the impact of a door vestibule on hourly indoor moisture contents by using a calibrated multi-zone model. The calibrated model provides realistic transient indoor humidity ratios. It is found that a door vestibule reduces indoor humidity ratio by approximately 9 percent when compared the humidity ratios in the case without the door vestibule. Installing the door vestibule can help to control indoor humidity levels within thermal comfort condition.

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