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Analysis of energy savings in a supermarket refrigeration/HVAC system

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ABSTRACT

The paper analyzes refrigeration/HVAC system energy consumption in a typical food retail store to study the effects of indoor space conditions. Refrigerated display cases are normally rated at a store environment of 24 °C (75 °F) and a relative humidity of 55%. If the store can be maintained at lower relative humidity, significant quantities of refrigeration energy, defrost energy, and anti-sweat heater energy can be saved. Calculations were done for a typical day in a standard store for each month of the year using the climate data for Tampa, Florida. This results in a 24 h variation in the store relative humidity. Using these hourly values of relative humidity for a typical 24 h day, the store relative humidity distribution was calculated for a full year. The annual average supermarket relative humidity was found to be 51.1%. It is shown that for a 5% reduction in store relative humidity, the display case refrigeration load is reduced by 9.25%, and that results in total store energy load reduction of 4.84%. The results were compared to available experimental data and found to have a good agreement.

1. Introduction

Supermarkets are high-volume food sales outlet with large storage turnover. Most food products need to be kept under certain ambient temperature and relative humidity. These foods are displayed in highly particular and flexible refrigerated display cabinets. Most of the retail food can be spoiled unless refrigerated. These foods include meat, dairy products, frozen food, ice-cream and frozen desserts, and different individual items such as bakery and deli products and cooked meals. When a refrigerated display case operates in the supermarket environment, it exchanges heat and moisture with its environment. The moisture exchange between the display case and the store environment is the most troublesome part of this event due to an increase in energy requirement to maintain a satisfactory temperature within the display case. Nevertheless, maintaining a low relative humidity in the store environment requires an air-conditioning system with larger capacity. This maybe more expensive and have higher operating cost. On the other hand, the operating cost of the display cases will be lower due to less latent load on the refrigeration coil, fewer defrosts to be required and less anti-sweat heater operation. Higher store relative humidity will result in lower operating cost of the air-conditioning equipment but in higher condensation on the display case walls, products and further frost on the evaporator coils.

In the literature, a reasonable number of research studies on refrigerated display cases have been reported. Howell and Adams [1] studied the effects of indoor space conditions on refrigerated display case performance. Howell [2,3] developed a procedure that evaluates the effects of relative humidity on the energy performance of refrigerated display case energy requirement, anti-sweat heater energy, and defrost energy requirements. Howell [2] showed that the savings in energy of the display cases ranged from 5% for closed door reach-in cases to 29% for multi-shelf display cases when operated at store relative humidity of 35% rather than at 55%. The increment of AC energy requirement when the store is operated at 35% relative humidity rather than 55%, ranged from 4% to 8% depending on the energy efficiency ratio (EER) value of the air-conditioning unit.

Tassou and Datta [4] investigated the effects of in-store environmental conditions on frost accumulation at the evaporator coils of open multi-deck refrigerated display cabinets. Their field and environmental chamber-based tests have shown that ambient relative humidity and temperature of a store have a significant effect on the rate of frost formation on the evaporator coils, with the effect of relative humidity being more evident than the effect of temperature. They concluded that a considerable opportunity exists to implement sophisticated defrost control strategies to save energy and improve temperature control. Orphelin et al. [5] discussed a new approach to estimate impacts of temperature and humidity set points on the total energy balance of typical French supermarkets. Their model took into account the cold aisle effect and the occurrence of thermal coupling between the supermarket display cases and the air-conditioning system. Their results showed that it is not cost effective to maintain a lower relative humidity level under 40% within the store during summer time. In addition, their results showed that the performance of air-con-

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Nomenclature

EER	energy efficiency ratio (Btu/W h)
h	enthalpy of moist air (kJ/kg)
$\dot{m}_a$	air curtain mass flow rate (kg/s)
NP	number of people
P	pressure (kPa)
q	volume flow rate (m^3/s)
Q	heat transfer (kW)
QL	latent heat (kW)
RH	relative humidity (%)
T	temperature (°C)
w	humidity ratio (kg_{water}/kg_{air})

ditioning and refrigeration systems for operating the display cases, have to be well known in order to define an acceptable set point in terms of energy consumption and customer comfort.

Rosario and Howell [6] experimentally evaluated the energy savings produced by reducing the relative humidity of the store. Eight supermarkets in the Tampa, Florida area were monitored for 12 h and 7 day periods between November 1997 and October 1998 in order to know the typical store relative humidity prior to its potential reduction. Five different areas of the eight supermarkets were monitored. The relative humidity within the store differed up to 20% and the average annual relative humidity between different stores varied up to 12%. Their results show that the average relative humidity of all stores have a minimum value of 37% during the month of March and a maximum value of 56% during the month of September. The annual average value for all stores is 45%. An algebraic expression based on experimental results was used to correlate indoor humidity ratio as a function of outdoor humidity ratio. Their results showed that the theoretical moisture balance model's prediction was within ±10% in comparison with the experimental data. They concluded that the total store energy bill (i.e. display cases, air-conditioning and lights) could be reduced up to 5% by lowering the store relative humidity by 5%. The store relative humidity reduction of 1% represented the savings of 18,000–20,000 kW h annually.

Kosar mad Dumitrescu [7] provided an updated review of currently available databases that address the effect of supermarket humidity on refrigerated case energy performance from computer simulations, laboratory tests, and field evaluations. Their database reviewed findings and tabulated those by case type, humidity range, and energy performance impact which were separated by compressor energy, defrost energy, and anti-sweat heater energy. Their findings revealed that the reduction in anti-sweat heater energy operation, compressor energy reductions, and electrical defrost reductions represent the 55%, 44%, and 1% of the store energy savings potential respectively. Although these conclusions differ with the store mix of case types and controls for anti-sweat and defrost operation, it is clear that anti-sweat heater requirements deserve as much attention as compressor or refrigeration loads of display cases at low humidity levels. Chen and Yuan [8] experimentally investigated the effects of some important factors on performance of a multi-shelf refrigerated display case. The factors include the ambient temperature and humidity, discharge air velocity, night covers and air flow from perforated back panels. The results showed that ambient temperature and relative humidity increase cause the temperature and heat gain of the display case to increase.

Due to the importance of numerical modeling to have effective and efficient refrigerated systems, Smale et al. [9] reviewed all numerical modeling techniques and the application of CFD during the period of 1974–2005 for the prediction of airflow in refrigerated food applications including cool stores, transport equipments, and retail display cabinets. Getu and Bansal [10] numerically and experimentally analyzed evaporator in frozen food display cases at low temperature in a supermarket in Auckland, New Zealand. Extensive experiments were conducted to measure store and display case relative humidities and temperatures, and pressures, temperatures and mass flow rates of refrigerants. The mathematical model used different empirical correlations of heat transfer coefficient and frost properties for the heat exchanger in order to investigate the influence of indoor conditions on the performance of the display cases. Experimental data were used to validate the model so that the model would be a tool for designers to evaluate the performance of supermarket display case heat exchangers under different retail store conditions. Ge et al. [11] integrated CFD with cooling coil model to simulate and analyze the performance of a multi-deck medium temperature display case. The 2D CFD model can predict the dynamics of air flow; heat and mass transfer among the airflow, food products and ambient space air. The model simulated different pipe and fin structures and circuit arrangements, with the outputs from the cooling coil model used as the inputs to the CFD model and vice versa. The validated model was used to examine the cabinet performance and explore the optimal control strategies at various operating conditions.

The importance of an air curtain in refrigerated display cases modeling motivated many researches and a number of studies have been published on the development of an air curtain. Howell et al. [12] theoretically and experimentally investigated the heat and moisture transfer through turbulent plane air curtains. They investigated the performance of air curtain by the variation of the number of jets, thickness, width, height, velocity, turbulence level of the air curtain, and pressure and temperature difference across the air curtain. An eddy viscosity model was used with finite difference technique to calculate the sensible, latent, and total heat transfer through air curtains. Howell and Shibata [13] experimentally investigated the relationship between the heat transfer through a recirculated air curtain and its deflection modulus. The deflection modulus was defined as the ratio of the initial momentum of the air curtain jet and the transverse forces magnitude in which the air curtain attempts to seal against. The authors demonstrated that there is an optimum flow velocity for the air curtain to seal the doorway and minimize the heat transfer rate and moisture effect.

Ge and Tassou [14] developed a comprehensive model, based on the finite difference technique, which can be used to predict and optimize the performance of air curtains. Based on the results obtained from their model, correlations for the heat transfer across refrigerated display case air curtains have been developed to enable quick calculations and parametric analyses for refrigeration equipment design and sizing purposes. Cui and Wang [15] used a computational fluid dynamics (CFD) method to evaluate the energy performance of an air curtain for horizontal refrigerated display cases and optimize their design. The authors studied the key factors that influence the air curtain cooling load such as: air curtain velocity, the height and shape of products inside the display case, temperature difference between the inlet and ambient air, and the relative humidity of the ambient. Their results showed that there is an optimum value for the inlet velocity of the air curtain, while other design parameters remain unchanged. They also found that the air curtain is heavily affected by both the inlet air temperature and the relative humidity of ambient air. Therefore, properly controlled indoor conditions, i.e. dry-bulb temperature and relative humidity, could well balance the cooling load of the store against that of the display cases and help achieve overall energy efficiency.

Navaz et al. [16] presented a comprehensive discussion on past, present, and future research focused on display case air curtain performance characterization and optimization. Ge and Cropper [17] developed a display case model by integrating three main component sub-models; an air-cooling finned-tube evaporator, an air curtain and a display case body at steady state. They described the analysis and performance comparison of a display cabinet system using R404A and R22 as the refrigerants. They concluded that the total cooling load of display case and refrigerant mass flow rate increased at higher ambient air humidity. A model of a typical supermarket was presented by Howell et al. [18,19]. This model was developed for a typical supermarket and an hourly moisture balance was performed for a typical 24 h day. The model stated that the net moisture loss due to building envelope and the operation of the air-conditioning equipment is balanced by the net production of moisture within the supermarket.

You and Lee [20] reported an analysis of air conditioning and energy consumption in a library building located in Kuala Lumpur, Malaysia. The performance of the HVAC system was improved by incorporating an ice slurry cooling coil. Chua et al. [21] presented a review of current technology on energy recovery using heat pumps in industrial, commercial, and residential applications. Manzela et al. [22] performed an experimental investigation of an ammonia–water absorption refrigeration system using the exhaust of an internal combustion engine as the heat source.

The objective of this work is to model a supermarket refrigeration/HVAC system, and to perform a numerical simulation for this model using MATLAB [23]. The model integrates the air curtain model developed in [14] for display cases within the main supermarket model. The simulation is performed for a typical day under standard store conditions for each month of the year using climate data for Tampa, Florida. A parametric study of this system and a prediction of energy consumption are done to study the effect of indoor space conditions on supermarket energy consumption. A sensitivity analysis is performed for the proposed model and validated with available experimental data. The main contributions are to validate the air curtain model developed by Ge and Tassou [14] within the supermarket model for Tampa, Florida weather conditions and calculate the energy consumption and energy savings when the store relative humidity is reduced.

2. Supermarket model

A model was developed for a typical supermarket based on data prepared by the Food Marketing Institute Energy Committee and the information presented in Refs. [18,19]. The layout for this typical supermarket is shown in Fig. 1. The store description is as follows:

Store floor area	3716 m^2 (40,000 ft^2)
Conditioned space	2787 m^2 (30,000 ft^2)
Air supply rate	14.16 m^3/s (30,000 cfm)
Outside ventilation air	1.84 m^3/s (3900 cfm)
Hours of operation	24 h/day
People in store	180 maximum. 92 W/person (315 Btuh/person) sensible and 75 W/person (255 Btuh/person) latent. People occupancy schedule is shown in Fig. 2
Indoor conditions	24 °C (75 °F), variable relative humidity
Supply air conditions	13 °C (55 °F), 95% relative humidity

The capacity of installed refrigerated display cases were set as the following: medium temperature horizontal single shelf display at [73 m (240 ft)], medium temperature vertical multi-shelf display at [73 m (240 ft)] and low temperature closed door reach-in at [91 m (300 ft)].

The hourly outdoor weather condition for Tampa, Florida is averaged for the years 2000–2010 [24] and illustrated in Fig. 3. The hourly moisture balance was performed on the supermarket for a typical 24 h day and averaged over the years 2000–2010. The annual effect can be obtained from the averaged weather data. The moisture balance, in terms of the latent energy balance is given by the following equation:

$$QL_{space} + QL_{infil} = QL_{people} + QL_{produce} + QL_{meat} + QL_{bakery} - QL_{display\ case} \quad (1)$$

The moisture balance states that the net moisture loss due to the building envelope and the operation of the air-conditioning equipment is balanced by the net production of moisture within the supermarket. The terms of the moisture equation are calculated from the following equations and thermal conditions [19]:

$$QL_{space} = 3010\, q_{space}(w_{space} - w_{supply}) \quad (2)$$

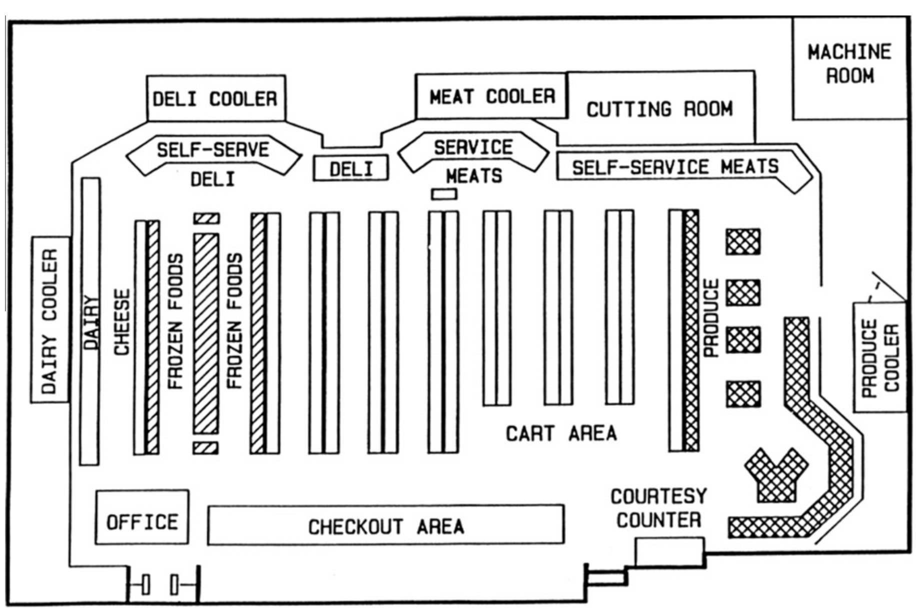

Fig. 1. Layout of a typical supermarket.

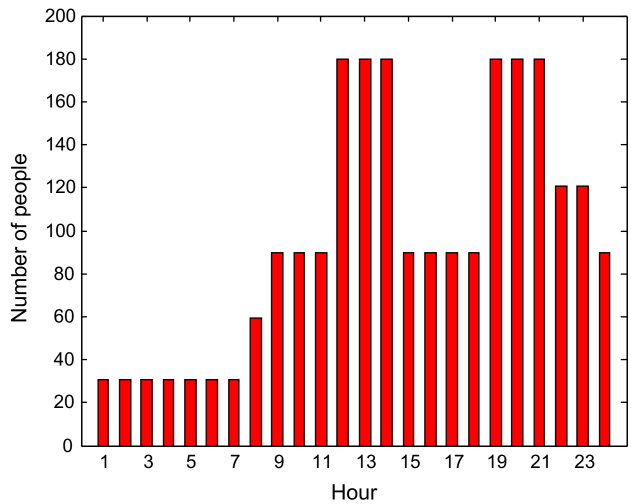

Fig. 2. Schedule of people occupancy in the supermarket model.

$$QL_{\text{infil}} = 3010\, q_{\text{infil}}(w_{\text{space}} - w_{\text{outside}}) \quad (3)$$

$$QL_{\text{people}} = 0.075\ \text{NP} \quad (4)$$

$$QL_{\text{produce}} = 0.4103\ \text{kW} = 1400\ \text{Btu/h} \quad (\text{constant for 24 h}) \quad (5)$$

$$QL_{\text{meat}} = 0.4103\ \text{kW} = 1400\ \text{Btu/h} \quad (\text{from 5 am to 10 am}) \quad (6)$$

$$QL_{\text{bakery}} = 3.517\ \text{kW} = 12{,}000\ \text{Btu/h} \quad (\text{from 5 am to 10 pm}) \quad (7)$$

where

$q_{\text{infil}} = (44.5\ \text{NP} - 0.095\ \text{NP}^2 + 10^{-4}\ \text{NP}^3)\Delta P_{\text{build}}$
$q_{\text{space}} = 14.16\ \text{m}^3/\text{s} = 30{,}000\ \text{cfm}$
$\Delta P_{\text{build}} = 4.02\ \text{mm}\ H_2O = 0.16\ \text{in}\ H_2O$
NP = Number of people in the store

The major component of the display case model ($QL_{\text{display case}}$), is given by the air curtain. A strong heat and mass transfer exist within the air curtain as it separates the internal and external environment as shown in Fig. 4. Fig. 4 illustrates a vertical multi-shelf refrigerated display case, a typical horizontal single shelf refrigerated display case and standard closed door reach-in refrigerated case. The correlation of Ge and Tassou [14] was used for the air curtain of the refrigerated display cases. The four main parameters that affect the heat transfer of air curtain are the store air enthalpy, the dry-bulb temperature of the air curtain supply, display case air

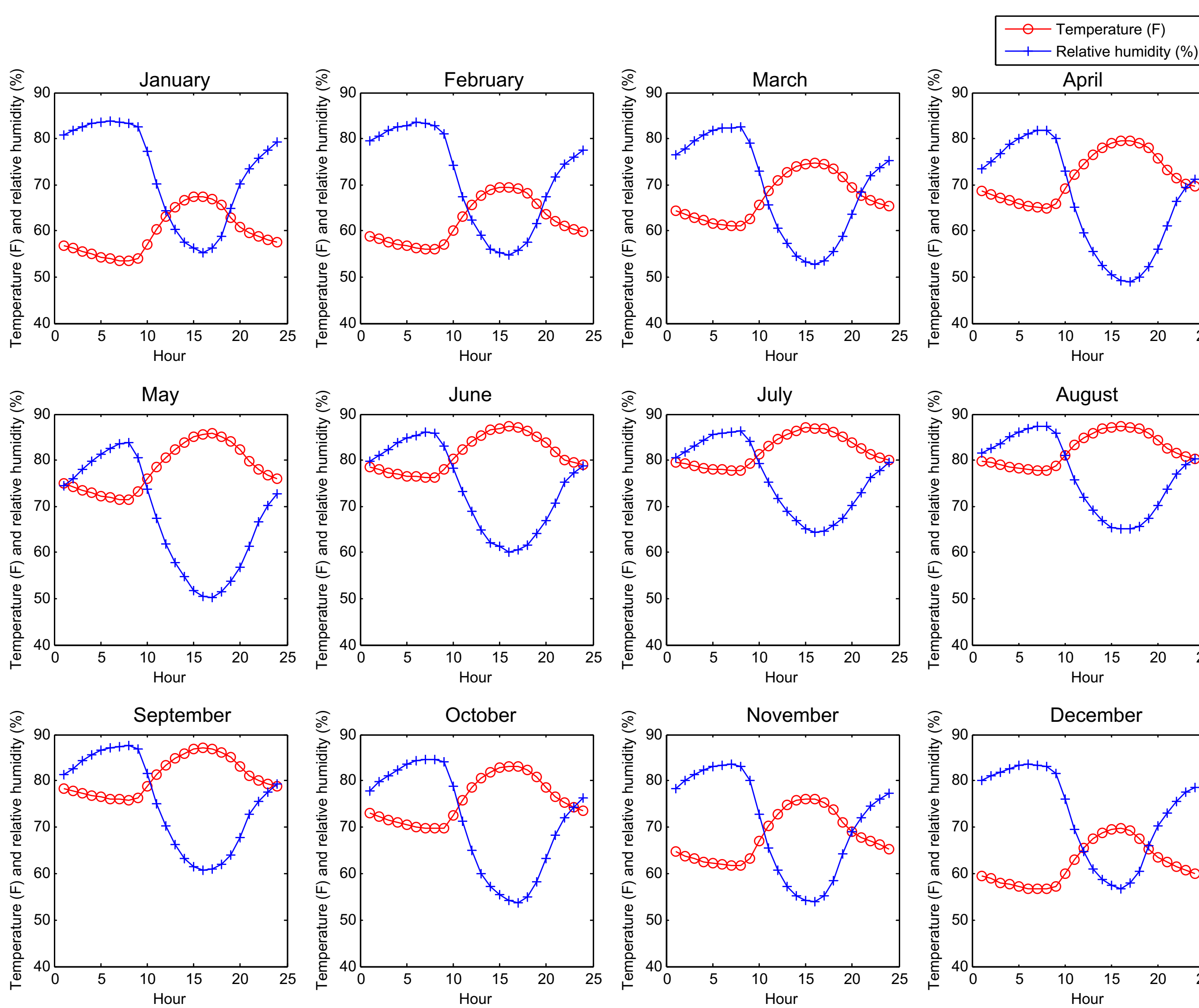

Fig. 3. Annual average hourly outdoor temperature and relative humidity variation in Tampa, Florida (2000–2010).

temperature or the air temperature differential between the display case and the curtain supply, and air curtain properties such as: air curtain velocity and length. The air curtain thickness effect is included as part of the curtain velocity, which is generally presented by the mass flow rate. In the current work, the effect of air curtain length is ignored by the assumption of a unit length. In addition, Ge and Tassou [14] correlation was used to predict the heat transfer of horizontal and closed door reach-in refrigerated display cases and vertical multi-shelf refrigerated display cases. This was validated as part of the sensitivity analysis. The design specifications of Howell and Adams [1] for different display cases are shown in Table 1. In general, the following correlation was used to predict the heat transfer of air curtain for any display case [14]:

$$Q_{\text{air curtain}} = [c_1 h_{\text{space}}^2 + c_2 h_{\text{space}} + c_3 (T_{\text{case}} + \Delta T)^2 + c_4 (T_{\text{case}} + \Delta T) + c_5 h_{\text{space}} (T_{\text{case}} + \Delta T) + c_6] \dot{m}_a \tag{8}$$

where

$$h_{\text{space}} = 1.0 T_{\text{space}} + w_{\text{space}} (2501.3 + 1.86 T_{\text{space}}) \tag{9}$$

and c_1 through c_6 are constants, which can be correlated from the simulation results of Ge and Tassou [14]. The correlated results of these constants are shown in Table 2.

ASHRAE [25] gave the percentage of latent load for each type of refrigerated display case; 12% for single shelf, 19% for multi-shelf and reach-ins respectively. These values take into account the performance of display cases with a store relative humidity maintained at 55%. The latent load percentage values for each type of refrigerated display case decreases at lower relative humidity and affects the simulated store relative humidity. However, the latent load percentage values for each type of refrigerated display case are taken as the maximum to prevent any frost formation and maintain the desired temperature of products. In this work, MATLAB [23] software was used to simulate the latent heat balance inside the supermarket. Steady state simulations were carried out on an hourly basis for the typical day in each month using the averaged annual data obtained from [24] using the weather conditions of Tampa, Florida. The average of these data are illustrated in Fig. 3. The store temperature was maintained at 24 °C (75 °F). The hourly moisture balance, Eq. (8), was used along with the air curtain heat equation, Eq. (9), and resulted in a relative humidity profile for the typical day inside the store. Fig. 5 shows a flow-diagram of how the simulation works using input data. The output includes an hourly store relative humidity for a typical day of each month of the year.

3. Simulation results

The results from the supermarket model simulations were run for the typical 24-h day for a year. The results present the store relative humidity each hour for typical day each month. Fig. 6 illustrates an hourly plot of store relative humidity for typical year in Tampa, Florida. The store relative humidity remains in the range of 40–60% from January till May and from October till December. During the summer season from June till September, the store relative humidity increases above 60% during noon times. This is obvious because of the hot and humid weather in Tampa, Florida during noon times.

The hourly values for the all months simulated have been averaged separately and are presented in Table 3. The monthly store relative humidity in Table 3 remains in the range of 40–60%. These results are dependent on the assumptions made for the supermarket model. However, these results appear to be typical for a supermarket with air-conditioning located in a weather condition similar to Tampa, Florida. The variation expected in the store relative humidity would be in the range of 40–60% for hot and humid climates. The results in Table 3 should be considered of what is anticipated in a supermarket rather than using a design store relative humidity of 55%. Thus, changes in the refrigerated display case energy can be estimated for increases or decreases in store relative humidity, and that can result in changes in the operation of the supermarket air-conditioning system. This will be analyzed later on in the energy consumption analysis section.

4. Sensitivity analysis

The performance of the model representing the supermarket refrigeration system needs to be evaluated for different outdoor conditions each month for the whole year. The incorporation of

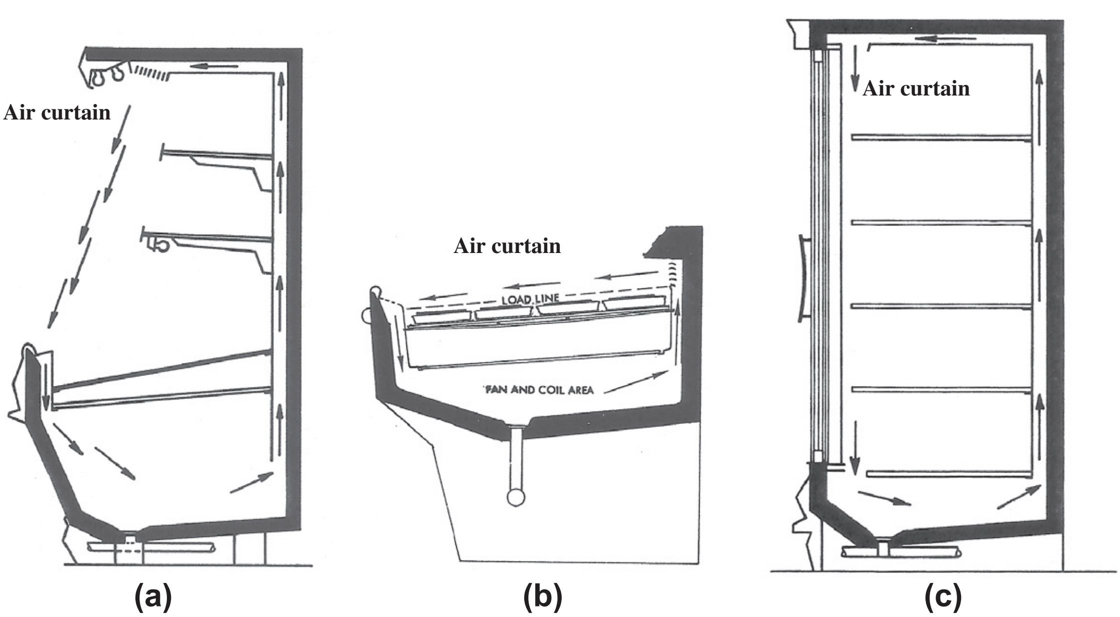

Fig. 4. Typical refrigerated display cases: (a) vertical multi shelf, (b) horizontal single shelf and (c) closed door reach-in.

Table 1
Design specifications for different types of refrigerated display cases [1].

Case type	Orientation	Case length m (ft)	Case temp. °C (°F)	Air curtain supply temp. °C (°F)	Air curtain velocity m/s (fpm)	Air curtain thickness m (in.)
Medium temp. single shelf	Horizontal	73 (240)	4.4 (24)	2 (35)	0.56 (110)	0.102 (4.0)
Medium temp. multi-shelf	Vertical	73 (240)	3 (37)	0 (32)	1.32 (260)	0.114 (4.5)
Low temp reach-in	Vertical	91 (300)	−2 (29)	−4.4 (24)	0.68 (133)	0.076 (3.0)

Table 2
The correlated constants $c_1 - c_6$ [14].

c_1	c_2	c_3	c_4	c_5	c_6
−0.180	303.180	−0.781	216.309	−0.448	509.975

air curtain correlation for latent heat calculation of the refrigerated display cases has been evaluated by a comparison with pervious simulation model [19] and existing experimental data [6] for the weather conditions of Tampa, Florida. The supermarket model developed by [19] has the same description of the current model, however, the current model includes the effect of store relative humidity on refrigerated display cases assigned in the moisture balance. Also, it can simulate the store relative humidity on an hourly basis. Figs. 7 and 8 show comparison of the hourly store relative humidity of [19] and the current model for the months of January and August, respectively. In January, the relative humidity inside the store exhibits stable behavior for the current model. This is because the current model has a precise representation of the display cases, and they are affected by the store relative humidity. The results of the current model are comparable with model in [19]. The discrepancy is because the input weather data used for the model in [19] is interpolated or extrapolated, while it is taken hour by hour in the current model. In the month of August, Fig. 8, there is an increase in the store relative humidity during noon times. This is due to the high temperature and high relative humidity conditions during summer season. However, the results of the current model are comparable with model in [19].

For another comparison, the average monthly relative humidity inside the store for [19] and the current model are calculated and plotted in Fig. 9. The current model shows matching trend with [19] where maximum store relative humidity is approximately 60% in August and minimum store relative humidity is approximately 45% in January. Comparison of the current model with available experimental data of Getu and Bansal [10] is shown in Fig. 10. For the same design condition in supermarket in Auckland, New Zealand, the current model simulates the store relative humidity for a typical day in December 2004. It is shown that the results are reasonably comparable with experimental data. Differences may be attributed to measurement errors (the paper reported an error bound of 2% for humidity sensors, 0.6% for thermocouples, and 2–5% for flow meters) and simplified assumptions in theoretical modeling.

Fig. 11 shows a comparison of the current model with the experimental data of Rosario and Howell [6]. It shows the maximum, minimum and average store relative humidity for a typical store located in Tampa, Florida. There is an increase in the store relative humidity during the summer season beginning in May and ending in September. The current model has a trend in the range of experimental data. Also, the percentage values assumed for latent heat calculation for the refrigerated display cases are maximum, so for more precise results, they need to be varied with store relative humidity. Therefore, the representation of display

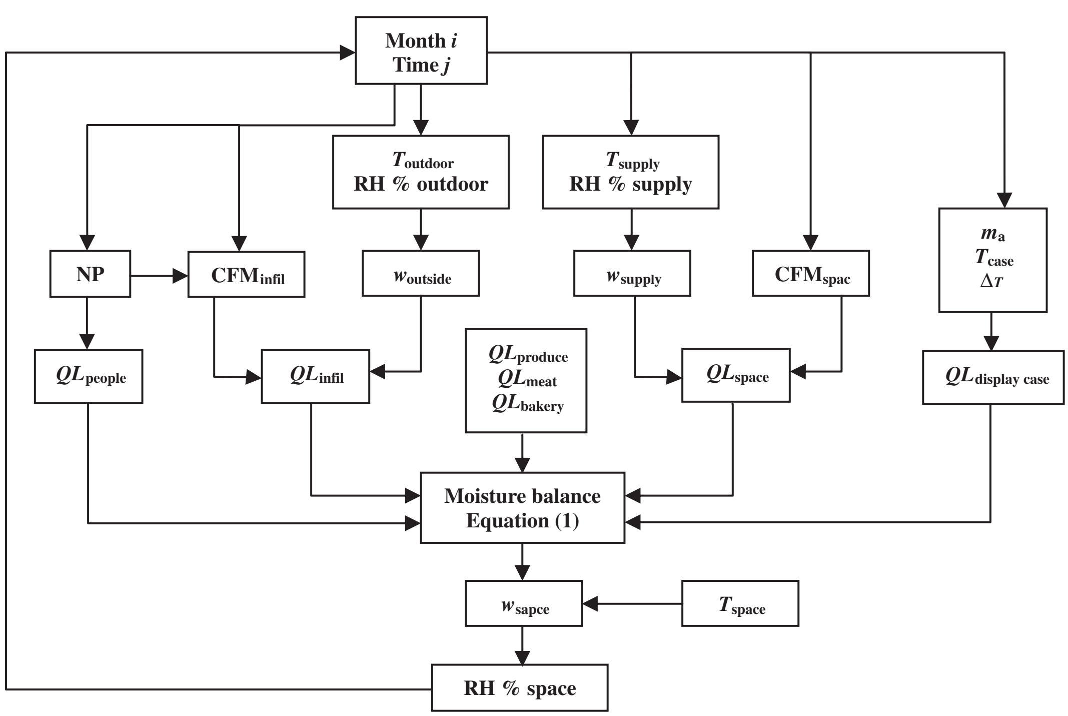

Fig. 5. Information-flow diagram for store relative humidity simulation.

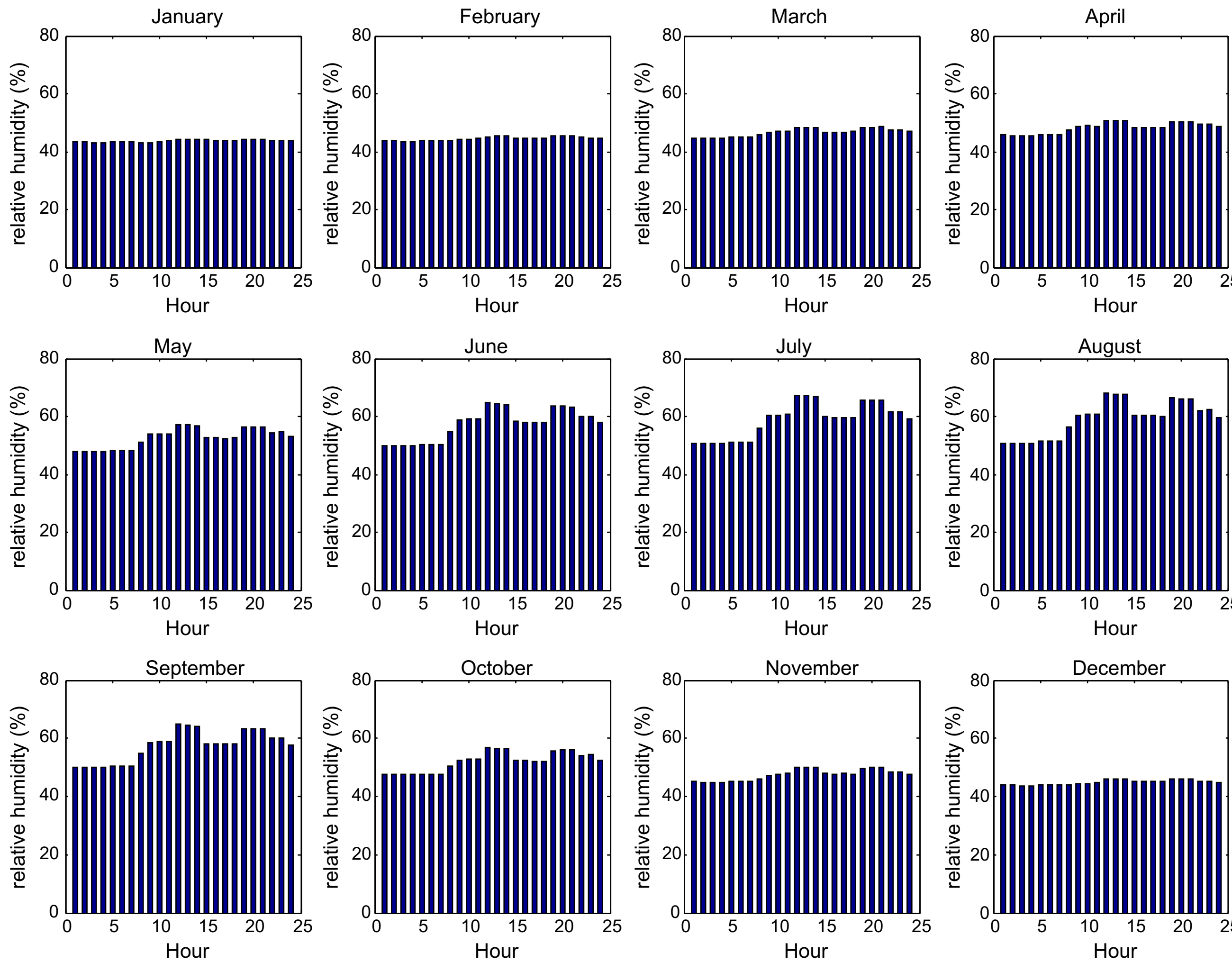

Fig. 6. Hourly relative humidity for model store for a typical year in Tampa, Florida.

Table 3
Average store relative humidity for supermarket model simulated at 24 °C (75 °F) for each month for Tampa, Florida.

Month	Average relative humidity inside store (%)
January	43.72
February	44.51
March	46.66
April	48.24
May	52.57
June	57.40
July	58.87
August	59.28
September	57.41
October	52.08
November	47.48
December	44.92

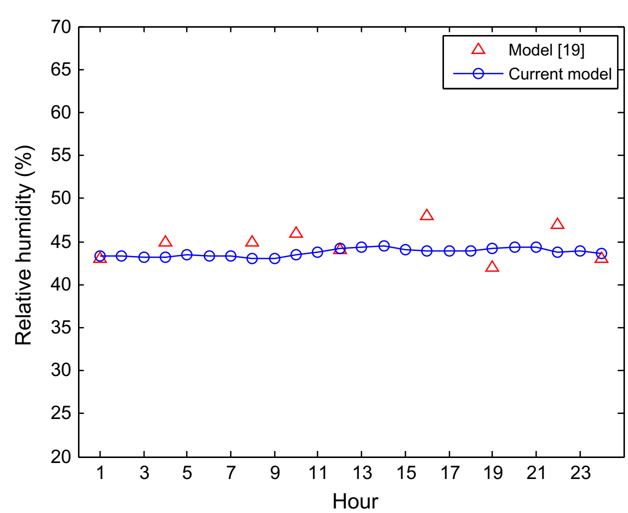

Fig. 7. Comparison of hourly relative humidity for the month of January.

cases brought by the air curtain energy Eq. (8) is feasible to be used in the supermarket model for latent heat calculations.

5. Energy consumption analysis

The modern supermarket is the greatest consumer for refrigeration energy within the commercial sector. In the United States, the electrical consumption in supermarkets presents 2.3% of the national electric use, and 50% of the total retail store energy is consumed by the refrigerated display cases and air-conditioning systems [25]. The relationship between the store HVAC and the refrigeration is very important in terms of the overall energy

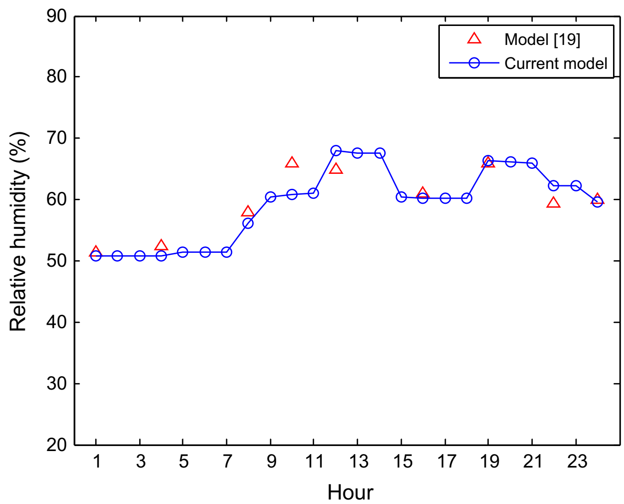

Fig. 8. Comparison of hourly relative humidity for the month of August.

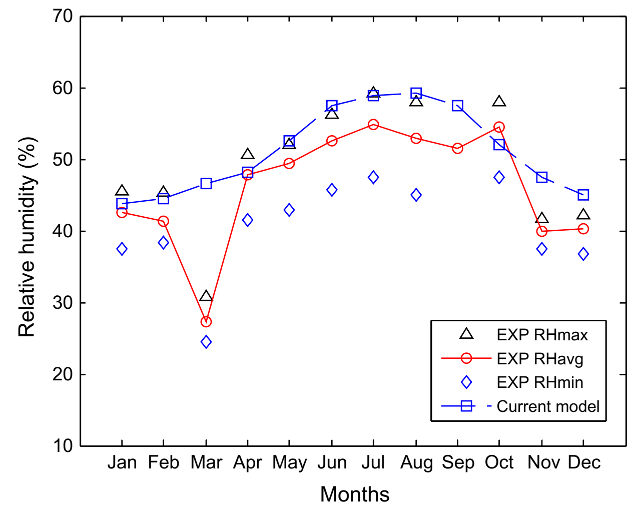

Fig. 11. Comparison of predicted relative humidity with experimental data [6].

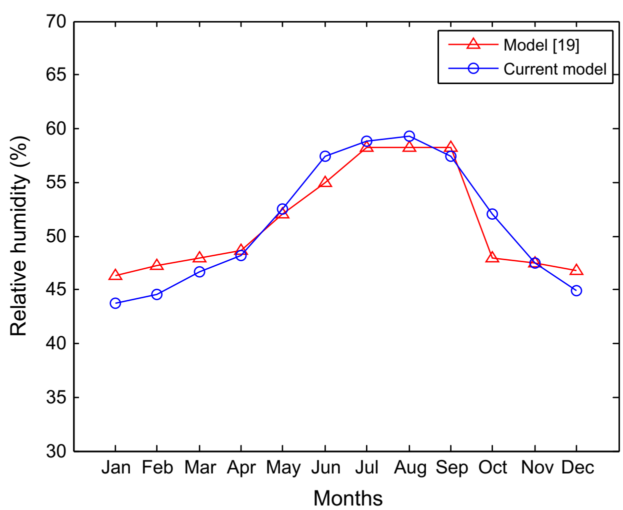

Fig. 9. Comparison of average monthly relative humidity for Tampa, Florida.

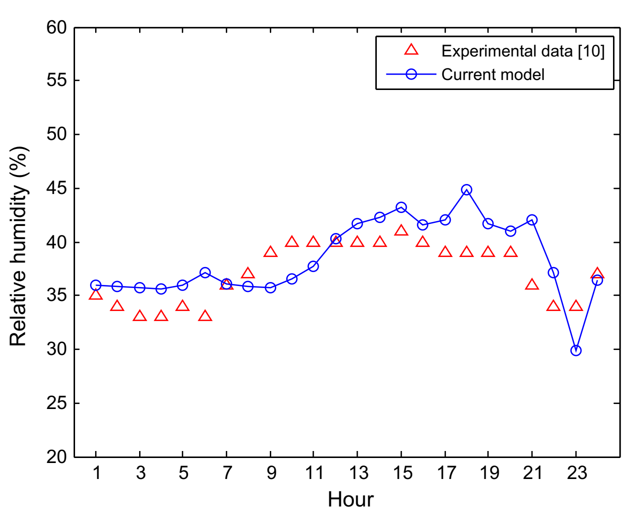

Fig. 10. Comparison of predicted relative humidity with experimental data [10] for a typical day in Auckland, New Zealand in December, 2004.

consumption of the supermarket. Howell and Adams [1] and Howell [2–3] developed several procedures to calculate the savings in store energy requirement by the knowledge of the indoor store relative humidity distribution during the year. They showed the influence of the indoor relative humidity on the refrigerated display case energy consumption. The method developed will be used in this work to evaluate the effect of store relative humidity on display case energy requirements. These energy requirements were divided into three components: energy required by the case refrigeration, energy required by anti-sweat heaters, and energy required for defrost. The store relative humidity affects all three of these components. Each of these refrigerated display case loads were evaluated on a percent change basis, compared to operation at relative humidity of 55%, and are given by the following equations [1]:

$$TP = QRH/QR \tag{10}$$

$$DP = DFRH/DF \tag{11}$$

$$AP = ASWRH/ASW \tag{12}$$

where TP is the ratio of display case refrigeration energy requirement when operated at a relative humidity other that 55%; QRH is the display case refrigeration energy requirement at a given relative humidity; QR is the display case refrigeration energy requirement at the design value of relative humidity of 55%; DP is the ratio of display case defrost energy requirements when operated at a relative humidity other than 55%; DFRH is the defrost energy requirement for the display case at a given relative humidity; DF is the defrost energy requirement for the display case at the design value of relative humidity of 55%; AP is the ratio of display case anti-sweat heater load when operated at a relative humidity other than 55%; ASWRH is the anti-sweat heater energy requirement for the display case at a given relative humidity and ASW is the anti-sweat heater energy requirement for the display case at the design value of relative humidity of 55%.

Howell [2] evaluated the values for TP, DP and AP when the store temperature was kept at 24 °C (75 °F). These values can be used for a wide variation in types of display cases as well as a full variation of case sizes and operating conditions. These values will be used later to estimate the energy saving when the store relative humidity is reduced.

In order to calculate the savings in energy in the operation of the display cases, it is necessary to establish its standard energy

Table 4
Display case refrigeration energy for simulated store at 24 °C (75 °F) and 55% relative humidity.

Case Type	Orientation	Case length m (ft)	Case temperature °C (°F)	Cooling rate W/m (Btu/h ft)	Cooling capacity kW (Btu/h)	Dimensionless EER (EER in Btu/W h)	kW demand	kW h/month
Medium temp. single shelf	Horizontal	73 (240)	4.4 (24)	604 (628)	44 (150,906)	2.344 (8)	18.86	13,580
Medium temp. multi-shelf	Vertical	73 (240)	3 (37)	1668 (1735)	122 (416,600)	2.051 (7)	59.50	42,840
Low temp. reach-in	Vertical	91 (300)	−2 (29)	515 (536)	47 (160,738)	1.758 (6)	26.79	19,289
Total		237 (780)	–	–	–	–	105.15	75,709
Defrost	2–4 per day							16,667
Anti-sweat heaters	23.4 kW							16,850

Table 5
Display case energy modifiers for various average annual store relative humidities.

Average annual store relative humidity	51.1% RH			45% RH			40% RH			35% RH		
	TP	DP	AP	TP	DP	AP	TP	DP	AP	TP	DP	AP
Medium temp. single shelf	0.966	0.904	0.882	0.905	0.738	0.688	0.855	0.594	0.510	0.811	0.470	0.312
Medium temp. multi-shelf	0.945	0.884	0.899	0.858	0.703	0.730	0.788	0.553	0.580	0.717	0.406	0.410
Low temp. reach-in	0.959	0.908	0.930	0.893	0.766	0.810	0.839	0.648	0.700	0.786	0.532	0.570

Table 6
Display cases annual energy requirements at various store relative humidities.

	55% RH	51.1% RH	45% RH	40% RH	35% RH
Refrigeration (kW h)	908,508	864,908	795,260	738,682	682,649
Defrost (kW h)	200,004	179,750	147,136	119,691	93,852
Anti sweat (kW h)	202,200	182,696	150,167	120,646	87,081
Total (kW h)	1310,712	1227,354	1092,564	979,019	863,582

Table 7
Percentage changes in energy for various store relative humidities (percent change compared to base case at 51.1% RH).

	55% RH	51.1% RH	45% RH	40% RH	35% RH
Total (kW h)	1310,712	1227,354	1092,564	979,019	863,582
Change (%)	+6.36	0.00	−10.98	−20.23	−29.64
Defrost (kW h)	200,004	179,750	147,136	119,691	93,852
Change (%)	+10.13	0.00	−18.14	−33.41	−47.79
Anti-sweat (kW h)	202,200	182,696	150,167	120,646	87,081
Change (%)	+9.65	0.00	−17.80	−33.96	−52.34
Refrigeration (kW h)	908,508	864,908	795,260	738,682	682,649
Change (%)	+4.80	0.00	−8.05	−14.59	−21.07

consumption for the refrigeration energy, defrost energy and anti-sweat heater energy. Eq. (8) is used to calculate the refrigeration energy for the display cases at 24 °C store temperature and relative humidity of 55% as shown in Table 4. The medium temperature single shelf horizontal units of length 73 m (240 ft) has calculated refrigeration energy of 604 W/m (628 Btu/h ft) and assumed to have a dimensionless energy efficiency ratio of 2.34 (EER of 8 Btu/W h). The medium temperature multi-shelf vertical units of length 73 m (240 ft) has calculated refrigeration energy of 1668 W/m (1735 Btu/h ft) and assumed to have a dimensionless energy efficiency ratio of 2.05 (EER of 7 Btu/W h). The low temperature closed door reach-in units of length 91 m (300 ft) has calculated refrigeration energy of 515 W/m (536 Btu/h ft) and assumed to have a dimensionless energy efficiency ratio of 1.76 (EER of 6 Btu/W h). Thus, kW demand and the kW h per month can be calculated as shown in Table 4 for the three display cases. Howell and Adams [1] gave approximate values for defrosts energy and anti-sweat heaters energy as shown in Table 4. They are taken at the rated store relative humidity of 55%. The number of defrosts varied from 2 to 4 per day and consumed 16,667 kW h per month and the total anti-sweat heater load was 23.4 kW which consumed 16,850 kW h per month. The annual energy load for the refrigeration, defrost and anti-sweat heaters is about 1311,000 kW h. Normally, this load is about 70% of the supermarkets total annual energy consumption.

In order to evaluate savings in display case energy with reductions in ambient store relative humidity it is necessary to determine TP, DP and AP at different store relative humidities. These three factors or modifiers can then be used with the energy loads given in Table 4 to estimate energy requirements at the different store relative humidity. The average monthly relative humidity for supermarket model is determined and listed in Table 3. Assuming each month has the same number of days, the 12 months are

Table 8
Changes in total store energy requirements at various relative humidities.

	55% RH	51.1% RH	45% RH	40% RH
Total display case annual energy (kW h)	1310,712	1227,354	1092,564	979,019
AC annual energy (kW h)	478,600	486,790	499,600	508,100
Lights and appliances, annual energy (kW h)	300,000	300,000	300,000	300,000
Total store annual energy (kW h)	2089,312	2014,144	1892,164	1787,119
Saving realized by changing from 55% RH (kW h)	–	75,168	197,148	302,193
Saving in kW h for each 1% reduction in RH (kW h)	–	19,274	19,715	20,146
Percentage savings in total store energy by changing from 55% RH (%)	–	3.60	9.44	14.46
Percent savings in total store energy for each 1% change in RH (%)	–	0.92	0.94	0.96

averaged resulting in an annual average store relative humidity of 51.1%. This seems to be a feasible value for the Tampa, Florida climate. Since display cases are designed for 55% ambient relative humidity, the actual annual energy requirement for the display cases for this supermarket model would be less than 1.31 million kW h as previously calculated.

To calculate the energy savings for these display cases, the three energy factors or modifiers are determined for the new store relative humidity of 51.1%. In addition, values for TP, DP and AP are determined for average annual store relative humidities of 45%, 40%, and 35%. They are listed in Table 5 for the three display cases. The values for TP, DP and AP are 1.0 for 55% RH. This is the reference point for upcoming results. Using the display case energy requirements at 55% relative humidity in Table 4, and energy modifiers listed in Table 5, annual energy requirements at various store relative humidities are estimated in Table 6. The total display cases energy load is separated into refrigeration energy, defrost energy and anti-sweat energy so that it can be compared with the actual situations. It can be noticed from Table 6 that reducing the store relative humidity results in considerable reduction in the total display cases energy requirements.

The percent savings in energy for the various components as well as the total display case energy savings for the various store relative humidities are given in Table 7. The base case for comparison is at store relative humidity of 51.1%. It may be noticed from Table 7 that for range of store relative humidity of 35–55%, the changes in energy requirements are approximately linear. These results show that for a 5% reduction in store relative humidity; the refrigeration load is reduced by 6.5%, the defrost load is reduced by 15%, the anti-sweat heater load is reduced by 16%, and the total display case load is reduced by 9.25%.

In order to justify the reduction in store relative humidity, the percent increase in air-conditioning energy required to reduce the store relative humidity by 5% should be determined. Howell [3] estimated the annual air-conditioning energy requirement needed to maintain the store at 24 °C (75 °F) and relative humidity of 55%. They simulated a retail store, and found that for AC unit with an energy efficiency ratio of 9.5 Btu/W h or 2.812 W of cooling per Watt of power, the annual energy was estimated to be 478,600 kW h. When reducing the store relative humidity to 45%, the same AC unit would require 499,600 kW h, and for store relative humidity of 35%, 516,600 kW h was required. For our designed store relative humidity of 51.1%, the AC energy required can be estimated to be 486,790 kW h. Howell [3] also showed, in order to evaluate reasonable percent changes in energy for the total supermarket, lights and appliances annual energy are required and estimated as 300,000 kW h. These data are shown in Table 8 to compare changes in energy requirements at different relative humidities for each component of the store electric bill. From Table 8, it can be determined that for a 5% reduction in store relative humidity, there is about 4.82% reduction in the total store annual energy. Also, it can be determined that for each 1% reduction in store relative humidity, there is an approximate savings in annual store energy of 19,000 to 20,000 kW h.

Howell et al. [19] estimated the annual air-conditioning energy requirement needed to maintain the store at 24 °C (75 °F) and relative humidity of 51.2%. They found that for a 5% reduction in store relative humidity, the display case refrigeration load is reduced by 10%, and that results in total store energy load reduction of 4.7%. Because of the integration of store relative humidity within the air curtain correlation in the moisture balance, the current model shows a reduction in the display case refrigeration load by 9.25% for a 5% reduction in store relative humidity, while Howell et al. [19] model had a reduction in refrigeration load of 10%. However, the recommended relative humidity by Howell et al. [19], and the one determined in this work are comparable. Estimated store relative humidities by Howell et al. [19] and in the current model are 51.2% and 51.1%, respectively. The current model shows a reduction in the total store energy load by 4.84%, while Howell et al. [19], the load was reduced by 4.7%. This explains the sensitivity of the current model. In addition, the current model incorporate the effect of store relative humidity with the refrigerated display cases incorporated in the supermarket model.

6. Conclusions

The integration of air curtain with moisture balance for supermarket model is necessary in order to assess the effect of reduced store relative humidity on display case energy requirements. So, thermodynamic analysis was used to simulate supermarket refrigeration/HVAC system using MATLAB software. For the simulated supermarket model described in this work with different types of refrigerated display cases, and located in a hot and humid weather such as Tampa, Florida, the annual average supermarket relative humidity was found to be 51.1%. The simulated store relative humidities were found to be in the range between 40% and 60% during the model year. The results show good agreement with previous model, and the experimental data validates the proposed model. The effect of indoor space conditions on supermarket energy consumption is studied. It is shown that for a 5% reduction in store relative humidity, the display case refrigeration load is reduced by 9.25% and that results in total store energy load reduction of 4.84%. These results evaluated the integration of air curtain correlation for quick design calculation and for the simulation of different types of display cases within a supermarket model. These results, which are not generally known for typical supermarkets in hot and humid climate will now allow the designer of the supermarket to simply and quickly determine typical store relative humidity so that savings in display case operation and total store energy load are correctly estimated.

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