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Dehumidification performance of a variable speed heat pump and a single speed heat pump with and without dehumidification capabilities in a warm and humid climate

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Research paper

Dehumidification performance of a variable speed heat pump and a single speed heat pump with and without dehumidification capabilities in a warm and humid climate

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ABSTRACT

Conventional air conditioning systems in houses respond to thermal loads by means of controlling dry-bulb temperature through the thermostat. As part of the process to control temperature, dehumidification is also provided. However, as houses are becoming more efficient, supplemental dehumidification is often necessary for homes located in hot and humid climates to control relative humidity intentionally. This study compared the dehumidification performance of a residential air conditioning system working in three operations modes to emulate three different systems: a system with a variable speed mode, a single-speed system with an enhanced dehumidification mode, and a single-speed system operating in a traditional or normal cooling mode. With operation mode changes achieved through software, this study constituted a novelty in the topic of humidity control by using a single machine, with the same exact physical set-up to directly compare the dehumidification performance of three types of systems.

Two types of days were of interest in the study, hot and humid days (summer season) and mild and humid days (Fall shoulder season). After assessment of the dehumidification performance, the variable speed mode was able to maintain relative humidity between 50% to 52% on summer days. In the single-speed with enhanced dehumidification, a slightly less effective humidity control was achieved on summer days with the mode keeping the relative humidity between 53% to 55%. In the normal cooling mode, which resembles a conventional system, the humidity levels were controlled between 55% to 60%. In the shoulder season, the variable speed and enhanced dehumidification modes maintained the relative humidity between 55% to 58% and 53% to 56% respectively. In the shoulder season, the normal cooling mode kept the indoor relative humidity near or above 60%. In terms of dehumidification efficiency expressed as a function of the amount of water condensate per unit of energy, the variable speed was determined to be more efficient than the other modes.

1. Introduction

Humidity control has become a greater issue in warm and humid climates with the improvements in home energy efficiency. Field testing of traditional homes in hot, humid climates by Shirey et al. (2006) has proven that conventional cooling equipment could be considered adequate to meet dehumidification loads. However, more recent studies have demonstrated that the same conclusion could not be drawn in newer and energy-efficient homes (Rudd and Henderson). As homes become more energy-efficient, an indirect approach to humidity control is less effective especially during the spring and fall season (mild temperature, high humidity). In fact, energy-efficient homes have low sensible heat gain which translates into less moisture removal while the latent load in those homes tends to prevail due to occupants' internal moisture generation (Ruud, 2013) and ventilation requirements (Rashkin, 2015). Furthermore, a more direct approach to humidity control in high-performance houses is desired because the percentage of dehumidification energy consumption from the total energy consumption can rise from 1.5–2.7% to as much as 12.6–22.4% if the relative humidity is outside of the desirable range of 50%–60% (Fang et al., 2011). Leakage in the return air duct of residential AC systems in warm and humid climates has proven to be detrimental to humidity control. In fact, high outdoor humidity conditions with return air leakage from an attic space as low as 10% resulted in sensible heat ratio (SHR) values greater than one (meaning the unit was unable to remove moisture from the air) in a study by O'Neal et al. (2002). Another

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Nomenclature

AC	Air conditioning
AHU	Air handler unit
ODU	Outdoor unit
CFM	Cubic feet per minute
CDD	Cooling degree days
DAQ	Data acquisition system
ERV	Energy recovery ventilation
SSPD	Single-speed mode with normal cooling mode
SSPD-ED	Single-speed mode with enhanced dehumidification mode
VSPD	Variable speed mode

reason for humidity control becoming a greater issue is that newly built homes use tighter constructions and tighter homes trap moisture loads from daily activities and have less natural ventilation (Turpin, 2010). ASHRAE standard 62.2 recommends controlling a home relative humidity below 60% to be within an acceptable range (ASHRAE, 2016). The primary advantage of humidity control is that it improves occupants' comfort and helps protect homes and belongings (Schwartz, 2011). Too much moisture inside a house leads to mold growth and bacteria which can result in health issues while moisture level below 40 percent can cause dry throats and noses to occupants (Turpin, 2010).

Traditionally, humidity control has been achieved either by dehumidification as part of an air conditioning system process of conditioning the humid air by means of controlling the dry-bulb temperature in a house or by direct dehumidification using dehumidifiers (Bathia, 2018). The market for supplemental dehumidification in the residential space conditioning industry is still in its early stages and technology continues to improve as homes become more efficient. In fact, the United States of America's demand for heating, ventilation, and air conditioning (HVAC) equipment has increased from nearly 11 billion US dollars in 2004 to 19 billion US dollars in 2014 (Rafique et al., 2015).

In the wake of HVAC technological improvements, variable speed technology has emerged in recent years and presents a new alternative for controlling humidity using an air conditioning system. Variable speed systems can often operate at 30%–40% of their rated cooling capacity and adjust its fan speed to lower cycling losses and improve indoor humidity control respectively (Munk et al.). Variable speed compressor technology relies on a compressor and static inverter. The static inverter converts the incoming alternating current to direct current. The variable-frequency alternating current is used to drive the compressor motor which is then capable of varying its speed and its amount of heating or cooling (Mix, 2014). The benefits of a variable speed air conditioning (AC) system include consistent indoor comfort and dehumidification in the sense that the extended system runs translates into more moisture removal (Goodman: Air Conditioning and HVAC Systems, 2019). More specifically, the long-extended runs of variable speed systems combined with the lower than standard cooling airflow will result in supply ducts operating at colder temperatures than cycling systems. These colder ducts will in turn lead to a lower delivered sensible heat ratio which is good for humidity control and dehumidification. These variable systems would operate at a low compressor speed (low cooling demand) in the mornings when outdoor temperatures are relatively cool. In the afternoons and as outdoor temperatures climb, the cooling load would increase, and the system would respond by increasing its capacity as well. Furthermore, variable speed systems are energy efficient. According to the Office of Energy Efficiency and Renewable Energy, a variable speed motor running continuously at half speed use 25% of the power that a single-stage motor would use to move the same amount of air (Office of Energy Efficiency and Renewable Energy, 2019). This study compares the dehumidification performance of three residential AC systems: a system with a variable speed mode (VSPD), a single-speed system with an enhanced dehumidification mode (SSPD-ED), and a single-speed system with a normal cooling mode (SSPD). Previous studies (Chan et al., 2009; Chen et al., 2018) have shown that variable speed and enhanced dehumidification AC systems were capable of achieving year-round improved indoor humidity control when compared to conventional units. Another study by Douglas (2006) has proven conventional systems with enhanced dehumidification components such as a wraparound heat pipe exchanger or wraparound desiccant dehumidifier can improve an integrated system's moisture removal capacity, thus resulting in a lower SHR that can better match higher latent loading applications. As a more accurate comparison, this study innovates by directly comparing three types of systems but using the same machine since the change of operation modes between the three types of system is done through software. With this approach, a real comparison is achieved for better evaluation of humidity control and determine if one system is more advantageous than another. In this study, data were recorded in I-P units. However, data in tables are presented in both SI and I-P units, while data in figures are presented in SI units.

2. Research facility

The facility used for this study was the "Patriot House" on the campus of the University of Texas at Tyler. The research house is a Net-Zero Energy house with a Home Energy Rating System (HERS) of minus 11 with a 7.4 kW of solar photovoltaic array. The two greatest advantages of using the "Patriot House" for comparison of VSPD, SSPD-ED, and SSPD were that: (1) the AC system was the same regardless of the operation mode, with changes from one mode to the other achieved by software change; (2) the house was a high-efficiency house and newly-built houses are heading towards that direction; thus allowing for the investigation of a current issue in humidity control.

In addition, the house had fresh air continuously provided by an energy recovery ventilation (ERV) system. Fig. 1 presents a layout of the research facility with the supply registers of the conditioned space and ERV identified with their airflows in SI and I-P units in Table 1. The airflow rates of the supply registers and ERV registers are based on the volume of air needed to satisfy the load of each room. These airflows were measured at the start and end of the study to ensure that no airflow distribution issues were encountered.

To conduct the research, the house was equipped with a data acquisition (DAQ) system to record the indoor and outdoor conditions. For the study, data were collected at 15-s intervals and Table 2 presents the monitored parameter sensor characteristics.

In terms of system description, the facility is equipped with a new 7-kW heat pump capable of switching operation modes by software changes. The variable speed mode (VSPD) single-speed relies on a variable speed compressor that can adjust its compressor speed and airflow based on demand. On the other hand, the single-speed modes, SSPD-ED and SSPD, use a compressor and supply fan which operates on On–Off cycling of the compressor.

The supply air temperature (or dewpoint) of the coil and the air volumetric flow rate supplied by the Air Handler Unit (AHU) were another difference between the operation modes. Table 3 recapitulates the system description and characteristics of the operation modes.

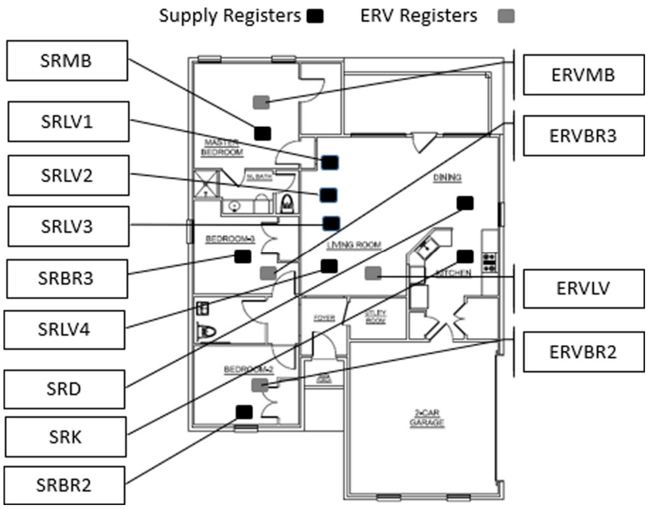

Fig. 1. Research facility layout with supply registers and ERV supply registers.

Table 1
Description of supply registers and ERV supply registers.

Space identification of supply register	Register identification	Airflow (m^3/min)	Airflow (CFM)
Master Bedroom	SRMB	1.70	60
Living Room 1	SRLV1	2.12	75
Living Room 2	SRLV2	0.79	28
Living Room 3	SRLV3	1.70	60
Living Room 4	SRLV4	0.85	30
Bedroom 3	SRBR3	1.73	61
Dining	SRD	3.40	120
Kitchen	SRK	2.69	95
Bedroom 2	SRBR2	2.27	80
Space identification of ERV supply registers	**Register identification**	**Airflow (m^3/min)**	**Airflow (CFM)**
Master Bedroom	ERVMB	0.99	35
Bedroom 3	ERVBR3	0.51	18
Living Room	ERVLV	0.51	18
Bedroom 2	ERVBR2	0.48	17

3. Approach for performance comparison

Knowing that finding days to evaluate both operation modes under similar conditions would be challenging, a dehumidification performance plot was used to compare the actual daily dehumidification (Y-axis of Fig. 4) with respect to a reference humidity load (X-axis of Fig. 4). The daily dehumidification was obtained from the sum of the differences in humidity ratio between the outdoor and indoor conditions multiplied by the time interval of data collection in 24 h. On the other hand, the daily humidity load was obtained as the sum of differences in humidity ratio between the outdoor and ideal indoor conditions (23.9 °C, 50% RH) multiplied by the time interval of data collection. Eqs. (1) and (2) illustrate how the daily dehumidification and daily humidity load, expressed in grams of water per kilograms of dry air, were determined.

$$\textit{Daily dehumidification} = \sum_{i=0}^{24} (W_o - W_i) \cdot \Delta t \tag{1}$$

$$\textit{Daily humidity load} = \sum_{i=0}^{24} \left(W_o - W_{23.9\ ^\circ\mathrm{C},50\%RH}\right) \cdot \Delta t \tag{2}$$

This approach worked because the house was unoccupied, and the humidity load introduced was due to mechanical ventilation of the ERV as well as infiltration. This approach allowed the comparison of performance for each day independently of weather conditions.

Furthermore, because results showed that the operation modes performed differently in the summer and fall shoulder season, it was thought that a methodology to estimate an approximate cutoff day between summer and fall seasons was important to have more general conclusions. The rationale behind the methodology was that the cutoff day between summer and fall season should account for the effects of the weather as well as the response of the system. Therefore, the daily cooling degree days (CDD), based on the actual balance point temperature of the house (21.1 °C), was selected to account for weather and daily amount of time the system was off (compressor off time) was chosen to account for system response to the weather. The Thompson Tau technique was applied to the data of CDD and compressor off time to remove outliers. Next, the CDD and compressor off time were normalized and plotted for the days during the test period. As illustrated in Fig. 2, the trend lines or curve fittings for the data show how September 3rd was obtained as a reference day between the summer and fall seasons.

Table 3
AC system description and characteristics.

Parameter	Value
AC system capacity	7 kW (2 tons)
Refrigerant	R410A
Airflow in VSPD	0 to 22.1 m^3/min (0 to 780 CFM)
Airflow in SSPD-ED	15.6 m^3/min (550 CFM)
Airflow in SSPD	22.1 m^3/min (780 CFM)
VSPD supply Temp	7.2 °C (45°F)
SSPD-ED supply Temp	10.0 °C(50°F)
SSPD supply Temp	12.8 °C (55°F)

Table 2
Monitored parameter sensor characteristics.

Monitored parameter	Sensor description	Sensor accuracy	Reference
House energy consumption AHU energy consumption ODU energy consumption	Electric meter	± 0.5%	EKM-Omnimeter I v.3
Indoor RH	RH sensor	± 2% RH	Dwyer RHP-2W10
Indoor temperatures	Type T thermocouple	± 1 °C	Thermocouple wire
Solar radiation	Solar radiation sensor	± 5%	Davis Vantage Pro2
Data logger	Temperature input module Current input module	±0.02 °C ± 0.04%	NI-9213 NI-9208
Outdoor temperature and RH	Temperature and RH sensor	±0.1 °C ±1.5% RH	Sensirion SHT3x-DIS

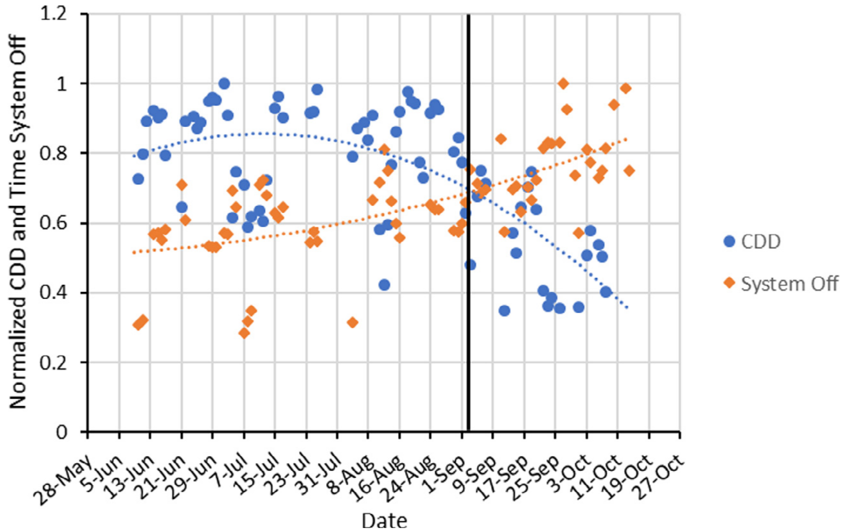

Fig. 2. Illustration of the methodology to separate summer and fall season.

4. Results and discussion

Analysis of the data collected allowed for the presentation of the results of the study with trends identified. For the study, it is also important to highlight the role of the ERV. Because the house was unoccupied, the ERV was the primary source of humidity load in the facility. Figs. 3 and 4 present the daily dehumidification versus humidity load for all the days collected in the summer and fall shoulder seasons. The relative humidity lines correspond to a plot of humidity load at indoor conditions of 23.9 °C and 45%, 50%, 55% RH (Y-axis) versus the reference humidity load at 23.9 °C and 50% RH (X-axis). These lines help to evaluate the mode performances in terms of relative humidity.

4.1. Summer season days

Fig. 3 shows the daily dehumidification vs humidity load for the data collected in the summer season. From this figure, VSPD dehumidified more effectively than SSPD-ED and SSPD respectively. In fact, VSPD maintained the relative humidity between 50% to 52% while SSPD-ED kept it between 53% and 55%. SSPD was able to control the relative humidity between 55% to 60%. In those summer days, VSPD had runtimes between 17 to 22 h while SSPD-ED and SSPD had total system runtimes between 8 to 9 h and 8 to 10 h respectively. As expected, the longer runtimes of VSPD on summer days created the opportunity to dehumidify more indoor air and provide a more comfortable environment. In contrast, the lesser humidity control achieved by SSPD-ED and SSPD can be explained on one hand by the On–Off cycling of the unit and on the other hand by the fact that it takes a while for the system to build latent capacity during the run cycles.

4.2. Fall shoulder season days

The days in the shoulder season were characterized by milder temperature and high humidity and thus a less effective humidity control. Another characteristic of the shoulder season was the shorter equipment run times because of the lower sensible heat gain on the facility. Fig. 4 shows that both VSPD and SSPD-ED followed trends different from the summer days. As a matter of fact, SSPD-ED controlled the relative humidity between 53% to 56% while VSPD controlled it between 55% to 58%. Overall, the VSPD and SSPD-ED modes maintained the relative humidity below the recommended level of 60%. However, in the SSPD mode, the humidity levels recorded were near or above 60% in the fall shoulder season. With the average total runtime of VSPD reduced in half from summer season to fall shoulder season (17 h to 8.5 h), the dehumidification performance of VSPD was reduced although its overall dehumidification performance was still acceptable. Comparatively, the shorter equipment runtimes in SSPD-ED and SSPD from summer to shoulder season (nearly 2 less hours of daily runtimes on average) led to less air being dehumidified.

4.3. Energy consumption and water removal

A comparison of daily water condensate as a function of the daily energy consumption was performed between the operation modes to determine whether one operation mode was more efficient than the others. Fig. 5 presents the liters of water condensate per total energy consumption (ODU + AHU). The daily water removal was obtained after performing a mass balance using the supply and return conditions of the AHU as well as the AHU airflow. From this Figure, VSPD was more efficient than SSPD-ED and SSPD since it removed more liters of water condensate per kWh of energy on average.

In the summer, the average gallons of condensed water per energy were 2.4 L/kWh, 2.2 L/kWh, and 2.0 L/kWh for VSPD, SSPD-ED, and SSPD respectively. In other words, VSPD removed roughly 8% more water condensate per kWh than SSPD-ED and 15% more water condensate per kWh than SSPD.

In the fall shoulder season, the averages for VSPD, SSPD-ED, and SSPD were 2.3 L/kWh, 1.9 L/kWh, and 2.0 L/kWh respectively. It means the VSPD removed about 21% more water condensate per kWh than SSPD-ED and 18% more than SSPD in the shoulder season.

5. Conclusions

The comparison of a residential heat pump operating as a variable speed system (VSPD), as a single-speed system with an enhanced dehumidification mode (SSPD-ED), and as a single-speed system (SSPD) was done in this study. The comparison using a single heat pump equipment was possible through an innovative approach achieved through software. This approach of using the same equipment eliminates the uncertainty associated with the hardware/equipment when comparing operation modes. Data collection in the three operation modes took place

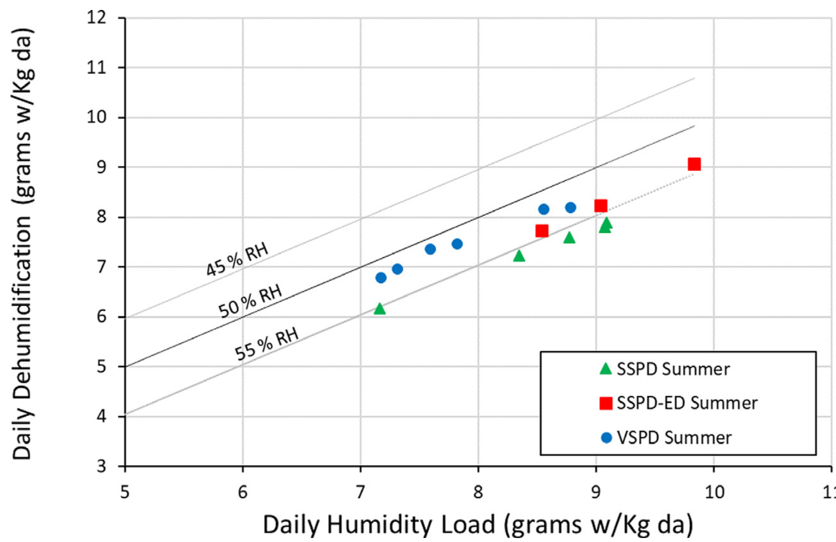

Fig. 3. Daily dehumidification vs. humidity load on summer days.

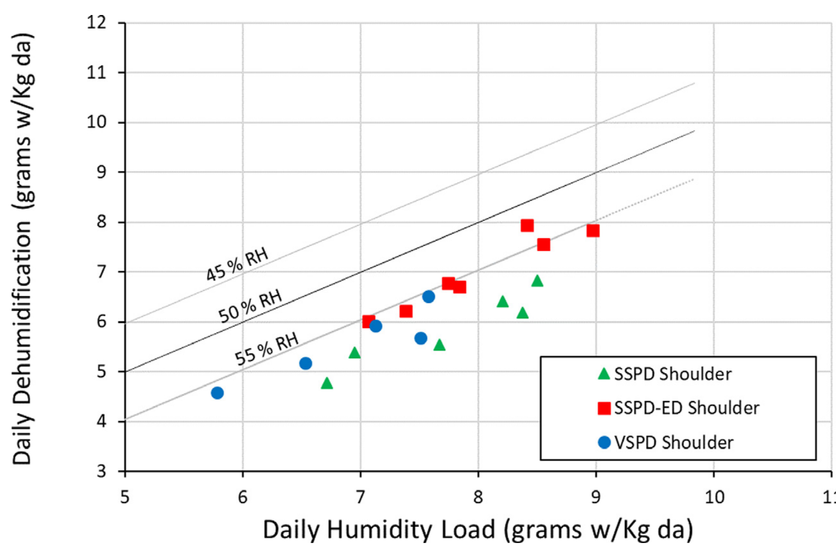

Fig. 4. Daily dehumidification vs. humidity load in the fall shoulder season days.

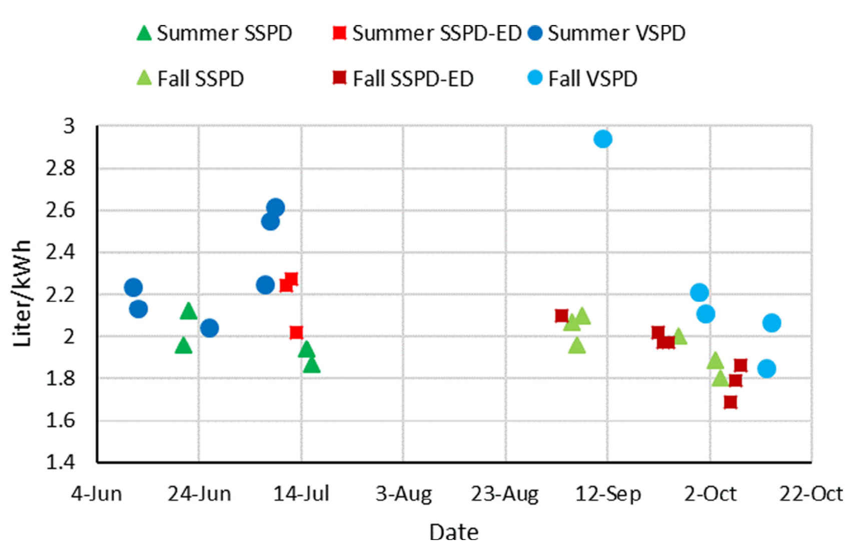

Fig. 5. Liters of water condensate per kWh in both seasons.

during the cooling season in 2018. Trends on dehumidification performance showed a significant difference between summer days and shoulder fall season. This motivated the development of a methodology to approximate a reference day to separate these two seasons in the data analysis. As expected, the dehumidification performance of both modes decreased in the shoulder season, which was characterized by part load conditions and shorter equipment run times. This result confirms that supplemental dehumidification is necessary in warm and humid climates since the modes were less effective in the shoulder season.

For the analysis of the data, a performance plot of daily dehumidification versus daily humidity load is proposed and used to allow the comparison of daily performances independently of the weather conditions. In the summer type days, VSPD proved to be more effective than SSPD-ED and SSPD. In fact, VSPD was able to maintain the indoor relative humidity levels within 50% to 52%, as opposed to SSPD-ED and SSPD that kept the levels between 53% to 55% and between 55% to 60% respectively. In the shoulder season, VSPD, SSPD-ED, and SSPD controlled the relative humidity between 55% to 58%, 53% to 56% and near or above 60% respectively. Although VSPD in the fall shoulder season did not dehumidify as well as it did in the summer, the operation mode still maintained the humidity below the 60% recommended level. Furthermore, VSPD proved to be more energy-efficient than the two other modes. VSPD removed more liters of water condensate per energy than SSPD-ED and SSPD in the summer and fall shoulder season. Total equipment runtimes turned out to be a determining factor in the dehumidification performance of the VSPD mode. Long extended runs in that mode improved humidity control and thermal comfort in the summer. Despite VSPD runtimes reduced in half in the shoulder season, there was a positive trade-off between relinquishing a few percent of relative humidity in favor of improved efficiency when comparing VSPD to SSPD-ED and SSPD.

In warm and humid climates, the increased use of variable speed AC systems offers the potential for improved year-round humidity control and energy savings.

Declaration of competing interest

The authors declare the following financial interests/personal relationships which may be considered as potential competing interests: As Nelson Fumo, a co-author on this paper, is the Editor-in-Chief of Energy Reports, he was blinded to this paper during review, and the paper was independently handled by Yaodong Wang as Associate Editor.

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