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Operational characteristics of residential and light-commercial air-conditioning systems in a hot and humid climate zone

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ABSTRACT

Forced-air space-conditioning systems are ubiquitous in U.S. residential and light-commercial buildings, yet gaps exist in our knowledge of how they operate in real environments. This investigation strengthens the knowledge base of smaller air-conditioning systems by characterizing a variety of operational characteristics measured in 17 existing residential and light-commercial air-conditioning systems operating in the cooling mode in Austin, Texas. Some key findings include: measured airflow rates were outside of the range recommended by most manufacturers for almost every system; actual measured cooling capacities were less than two-thirds of rated cooling capacities on average; hourly fractional operation times increased approximately 6% for every °C increase in indoor–outdoor temperature difference; and lower mean indoor surrogate thermostat settings and higher supply duct leakage fractions were most associated with longer operation times. The operational characteristics and parameters detailed herein provide insight into the magnitude of the effects of HVAC systems on both energy consumption and indoor air quality (IAQ) in residential and light-commercial buildings.

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1. Introduction

Buildings account for approximately 40% of the total amount of energy consumed in the United States, with nearly equal contributions from both residential and commercial buildings [1]. Over 70% of residential buildings in the U.S. are single-family dwellings [2] and over 50% of commercial buildings are light-commercial buildings (defined as having less than 465 m^2 of floor area) [3]. Centralized space conditioning has become ubiquitous in U.S. buildings. Over 60% of existing residential buildings and approximately 90% of newly constructed residences in the U.S. use central forced air distribution systems for air-conditioning purposes [4] and approximately 20-25% of all light-commercial buildings in the U.S. use the same style of central air-conditioning systems found in residences [5]. The characteristics of the U.S. building stock and their heating and cooling systems are important not only for energy consumption, but from an air quality perspective as well. On average, Americans spend nearly 90% of their time indoors and nearly 75% of their time at home or in an office [6], and human exposure to airborne pollutants is often greater indoors than outdoors [7,8].

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Despite their importance, gaps exist in our knowledge about how residential and light-commercial HVAC systems actually operate in real environments, particularly in the peer-reviewed archival literature. Several studies have found that the actual field performance of HVAC systems is different from laboratory performance or design conditions, in terms of system capacity, airflow, and refrigerant charge, which can have major implications for energy consumption [9–14]. Because filters in central air-conditioning systems are often the major mechanisms of indoor pollutant removal and are often relied upon to deliver clean air to occupied spaces, short operation times and low airflow rates can also have implications for indoor air quality (IAQ). In addition, typical input parameters to IAQ models and experiments that evaluate exposures and pollutant removal technologies, such as airflow rates, temperatures, and operation times, often come from ideal or design conditions (or are simply assumed) [15–20] and may not accurately describe real systems.

This work attempts to strengthen the knowledge base of smaller air-conditioning systems in the U.S. by characterizing a variety of operational characteristics measured in 17 existing residential and light-commercial air-conditioning systems in the hot and humid climate of Austin, Texas, collected from a previous dataset [21]. Relevant characteristics and parameters, including indoor and outdoor unit operation, ductwork characteristics, pressure measurements, fractional operation times, and a surrogate for thermostat settings are reported and compared to values measured or assumed in the literature. The magnitude and direction of the impact that some key





parameters have on energy consumption are also explored. The results herein provide insight into operational characteristics and parameters that influence both energy consumption and IAQ in residential and light-commercial buildings and provide a reference for modelers and experimenters investigating energy and IAQ to use in their work.

2. Background

A typical residential or light-commercial central air-conditioning system in the U.S. consists of an air-handling unit (AHU) with a blower fan, heating coil, and cooling coil, connected to supply (and usually return) ductwork (Fig. 1). The cooling coil in the AHU is connected with refrigerant lines to a condenser-compressor unit located outdoors and systems cycle on and off to meet thermostat demands for space conditioning. There is generally no intentional outdoor air intake or mechanical ventilation. Ductwork is often located outside of conditioned space and unintentional duct leaks can increase energy consumption, peak electricity demand [22-25], and air infiltration rates [26]. Several standards exist for testing the energy performance of systems at standardized laboratory conditions (AHRI Standard 210/240), as well as actual field performance of duct systems (ASHRAE Standard 152, ASTM E1554). Fig. 1 shows a typical system arrangement and key parameters that influence both energy consumption and IAQ.

Table 1 describes the types of primary effects that individual system and operational parameters in Fig. 1 have on energy consumption and IAQ, if treated independently. However, many of the individual parameters combine to affect energy and IAQ in complex ways. For example, airflow rates through the AHU influence fractional operation times (i.e., duty cycle) and recirculation rates through filters, but also influence AHU fan power draws, cooling capacity, and temperature and humidity differences within ducts and AHUs. Conversely, airflow rates and plenum operating pressures are directly related, and airflow rates are influenced by pressure drops across filters and heating and cooling coils in the AHU. Operating pressures also influence duct leakage rates, which influence both energy and IAQ as duct leaks waste energy and can

be sources or losses of indoor pollutants. Finally, occupant thermostat settings affect many parameters too, including fractional operation times, recirculation rates, cooling capacity, and temperature and relative humidity.

Although some of these parameters have been well described in the literature, there are still gaps in our knowledge of how interactions of many of these system operational parameters affect energy use and IAQ in real buildings. Much of the current state of knowledge of individual parameters is explored below in the context of four main system components: (1) AHUs, (2) outdoor units, (3) ducts, and (4) occupant influences and overall performance.

2.1. AHU operation

2.1.1. Airflow and recirculation rates

The performance of an air-conditioning system is in part dependent on the airflow rate through the system. Manufacturers typically recommend airflow rates for smaller systems between 169 and 193 m³ h⁻¹ per kW of capacity, although a wide range of airflow rates have been measured in field installations [10,11]. The recirculation rate (the HVAC volumetric airflow rate divided by the volume of space that a system serves) is an important parameter in IAQ models, particularly those that assess pollutant removal technologies, because the product of in-duct air cleaner efficiency and recirculation rate can be directly compared to other loss mechanisms including air exchange and deposition loss. Recirculation rates are a function of system airflow rates, house volume, and fractional operation times (i.e., duty cycles) and typical values used in models and experiments in the literature range from 0.67 to 24 h⁻¹ [15,17,19,20,29].

2.1.2. Fan power draws

Studies have shown that AHU power draws often exceed standard assumptions for air-conditioner rating test procedures and that residential AHU fans regularly consume more energy annually than a typical refrigerator [30]. Proctor and Parker (2000) compiled results from 9 field tests and reported that AHU fan power draws ranged from 0.29 to 0.34 W per m³ h⁻¹ of airflow (compared to the



Fig. 1. Typical residential or light-commercial building with central air-conditioning system. Filters are also often installed at the AHU, downstream of return ductwork.

Table 1

Primary effects of individual system parameters on energy and IAQ.

System category	Change in parameter ^a	Primary effects on energy and IAQ ^b
AHU	Airflow rate	Increased airflow rates can increase cooling efficiency [10]. Increased face velocities can increase filtration
		efficiency of larger particles and decrease filtration efficiency of smaller particles [27]. Decreased airflow
	Recirculation rate ^c	Increased recirculation rates imply longer system runtimes but provide more opportunity for removal by
	Recirculation face	in-duct air cleaners. Increased recirculation rates can also increase deposition of
		particles and ozone to ducts [19].
	Fan power draw	Increased fan power draw both directly and indirectly increases energy consumption by drawing more
		electrical power and by adding heat to the air stream.
Outdoor Unit	Cooling capacity ^d	Increased cooling capacity reduces system runtimes.
	Power draw	Increased outdoor unit power draw directly increases energy consumption and decreases cooling efficiency.
Ducts	Supply duct leakage	Increased supply duct leakage to an exterior zone wastes energy [22,34] but may remove more
		contaminants by exfiltration.
	Return duct leakage	Increased return duct leakage reduces cooling capacity [22,34] and may introduce
		new pollutants from outdoors.
	Temperature differences ^d	Increased conduction through duct surfaces (between the unconditioned exterior and the interior of ducts)
		can decrease cooling capacity by elevating supply air temperatures.
	RH differences ^d	Increased water vapor transfer from humid exteriors into return duct leaks can increase latent loads.
Occupant Influences	Fractional operation	Increased operation time increases energy consumption directly but allows for more contact time of indoor
and Overall Performance		air with in-duct air cleaners [41].
	Thermostat settings	Increased thermostat settings decrease energy consumption by lowering system runtimes.

^a Holding all other parameters constant.

^b Primary IAQ impacts concern only indoor pollutants and ignore secondary effects such as moisture.

^c A recirculation rate is the volumetric airflow rate through an air-handling unit divided by the volume of space that the system serves. It is comparable to an air exchange rate and has dimensions of inverse time.

^d These parameters can also affect indoor moisture levels in many ways, from localized moisture accumulation to overall moisture removal in the conditioned space.

standard assumption of 0.21 W per m³ h⁻¹) [31]. The inverse of those measured values (3.0–3.5 m³ h⁻¹ W⁻¹ measured vs. 4.8 m³ h⁻¹ W⁻¹ assumed) provides a measure of fan efficacy, or the amount of air moved per unit of power drawn by the AHU fan.

2.2. Outdoor condenser-compressor unit operation

The outdoor condenser-compressor unit typically draws the greatest amount of power in an air-conditioning system (e.g., 80–85% of total power, with AHU fans drawing the remaining 15–20%) [21,32], which impacts both energy consumption at the building level and the peak demand of electric utilities when aggregated. Equipment size, refrigerant charge levels, and climate conditions all affect the power draw of outdoor units. Actual measured cooling capacities are often lower than rated capacities because of differences between rating test and operational conditions, inadequate refrigerant charge, duct leakage, and low airflow rates. Proctor and Downey (1999) reported that the average performance of residential air-conditioners is at least 17% below rated performance [33]. In an overview of almost 9000 residential air-conditioners and over 4000 light-commercial air-conditioners in California, Downey and Proctor (2002) reported that the majority of residential and commercial systems had rated capacities of 8.8–10.6 kW and 15.8–17.6 kW, respectively [14]. Over half of the systems had either too much or too little refrigerant charge, defined as more than 5% from correct charge as recommended by the manufacturer.

2.3. Ducts

2.3.1. Duct leakage

Parker et al. (1993) simulated residential duct systems and estimated that the combination of air leakage and heat transfer in ductwork located in unconditioned attics could increase summertime peak electricity consumption more than 30% [34]. In one field study, Jump et al. (1996) reported an average decrease in HVAC energy use of 18% in houses that were tested before and after duct retrofitting (ranging from 5% to 57%) [22]. The IAQ impacts of duct leakage and environmental conditions within duct systems are currently not well characterized, although some knowledge exists

infiltration rates [26,36]. For example, Modera (1993) reported that the operation of HVAC fans in residences with an average of approximately 0.5 cm² m⁻² of return and supply duct leakage area increased average infiltration rates from 0.24 h⁻¹ to 0.69 h⁻¹ [36].

on the contribution of return duct leakage to filter bypass [35] and

2.3.2. Operating pressures and pressure drops

System pressures are important for both energy and IAQ because they drive the magnitudes and directions of many other influential parameters. For example, the airflow rate through an AHU is governed by the response of the fan to the airflow resistance of the distribution system (i.e., the total system pressure). Establishing the duct system resistance prior to system installation is difficult since most systems are site built and duct resistance is often affected by installation issues such as return grilles that are smaller than planned, inadequate duct design, or collapsed ducts [10]. Excessive system pressures associated with distribution systems have been shown to severely restrict system airflow rates [10]. In addition, duct leakage is strongly related to pressure differences between the distribution systems and surrounding space, as well as the position of leakage areas in the distribution system. We are aware of only a few studies that have reported actual operating pressures in supply and return duct systems [10,31,36,37].

Other important pressure drops within typical HVAC systems are the pressure drops across the filter and coils, and how those relate to total system pressure. We previously reported that median filter pressure drops across three types of filtration efficiencies as measured in the 17 systems in occupied buildings discussed in this paper ranged from 34 Pa with low-efficiency (MERV <5) filters to 55 Pa with high-efficiency (MERV 11-12) filters [21]. Ranges of those individual filter and coil pressure drops were 1-162 Pa and 1–269 Pa, respectively [38], albeit with a high level of uncertainty because of difficulties locating pressure taps in appropriate locations in some systems. In two unoccupied test house systems, we measured mean pressure drops ranging from 16 to 86 Pa across three types of filters and from 48 to 75 Pa across cooling coils, decreasing slightly as filter pressure drop increased [39]. We are not aware of much work in the literature on the relative importance of filter and coil pressure drops in the total pressure drop of systems in occupied buildings, specifically in hot and humid climate zones.

2.4. Occupant influence and overall performance

2.4.1. Fractional operation times

Residential and light-commercial air-conditioning systems typically cycle on and off to meet the cooling load of the building and the frequency of system operation times affects both energy and IAO. However, we are not aware of much information in the literature about how often systems operate to meet cooling loads in real environments. Previous IAQ modeling investigations have traditionally either assumed values for fractional operation times [15,17,20] or estimated them from energy models [40]. James et al. (1997) reported average fractional operation times of 8-14% for correctly sized systems in Florida homes in the summer [9]. Thornburg et al. (2004) measured the duty cycles of residential HVAC systems during 182 days of heating and cooling operation in 26 homes in North Carolina and 33 days of cooling operation in 9 homes in Florida [41]. Mean air-conditioner duty cycles were 6% (std. dev. 5%) and 21% (std. dev. 11%) in the NC and FL homes, respectively. It was not clear whether duty cycles were typically high enough to effectively decrease indoor pollutant levels and that additional data are needed to characterize ranges of fractional operation times, which is one of the primary goals of this paper.

3. Methodology

Seventeen air-conditioning systems were previously monitored during 2007–2008 for another project investigating the energy implications of higher-efficiency air filtration in occupied buildings [21]. That investigation generally concluded that the energy impacts of filters were minimal and that a wide variety of climate conditions and occupant thermostat settings heavily influenced the results. This paper reports previously unpublished operational parameters of the test systems made during the cooling season. A shorter duration of the measurements was made during heating season visits, but are not included in this paper due to the small number of those visits. Full details on measurement methods are available in Stephens et al. (2010) [21].

The 17 systems were located in buildings in the hot and humid climate of Austin, Texas (climate zone 2A according to ASHRAE Standard 169 [42]). Eight of the 17 systems were located in singlefamily residences and nine were located in light-commercial buildings. The test sites were visited once a month for one year, during which time three categories of filtration efficiency typically used in residential and light-commercial systems were installed: low-efficiency (MERV <5), mid-efficiency (MERV 6-8), and highefficiency (MERV 11-12) filters, as defined by ASHRAE Standard 52.2. Pressure measurements were made across the filter(s) and cooling coil and between the occupied space and the supply and return plenums. Two custom-built data-loggers containing power meters and pressure transducers were launched to log for approximately 24 h with the thermostat operated normally by the building occupants. One data-logger was connected to pressure taps, voltage taps, and current transducers at the AHU and logged the pressure drop directly across the filter(s) and cooling coil and the true power draw of the AHU fan. The pressure and voltage taps and current transducers remained installed for the duration of the one-year test period. The second data-logger was connected to transducers installed in a similar fashion at the outdoor condensercompressor unit, logging the true power draw of the unit. Temperature and relative humidity was logged outdoors, in the zone that contained the majority of the ductwork (usually the attic), inside the return plenum, inside the supply plenum, and at a single supply register. Airflow rates were measured once with a flow plate device and subsequently estimated during each monthly visit by correcting for system operating pressures. Duct leakage was measured with a calibrated fan and also corrected for operating static pressure. Manufacturer-reported uncertainty for each measured variable is reported in full detail in Stephens et al. (2010), but uncertainty values for measurements of pressure drop, power draw, temperature, relatively humidity, airflow rates, and duct leakage flows were 1%, 1.5%, 0.4 °C, 2.5% RH, 7%, and 3%, respectively.

Many of the values subsequently reported are measured at periods of "steady-state" operation. Steady-state cooling operation is effectively achieved in our analysis when the supply plenum temperature did not vary for a period of at least 2 min by more than 0.5 °C from the lowest temperature recorded during a cycle. Steadystate cycles also had to be at least 6 min in length due to the response time of the temperature and relative humidity instrumentation. All data analysis was performed using the statistical software package Stata, Version 11 [43]. A Shapiro-Wilk test was performed on many of the parameters identified in the subsequent section in order to test for normality or lognormality of the distributions. The null hypothesis that the variables were from either distribution was rejected when the *p*-value was less than 0.05 and was accepted when greater than 0.05. Medians and ranges are reported for all variables, as well as arithmetic means and standard deviations if the variables were consistent with this definition of a normal distribution, and geometric means (GM) and standard deviations (GSD) if the variables were consistent with this definition of a lognormal distribution.

4. Results and discussion

The following section details a variety of system characteristics and operational parameters measured in the test systems, organized by the parameters listed in Fig. 1.

4.1. Site and measurement summary

Some building and individual HVAC system characteristics are described in Table 2 (systems are referred to as "Sites" in the remainder of this work). Seventeen systems were located in 14 buildings, two of which contained multiple HVAC systems serving different floors or areas (Sites 3 and 4 and 16 and 17) and one of which was two offices in each half of a duplex with separate HVAC systems (Sites 9 and 10). Sites 1-8 were in residential buildings and sites 9-17 were in light-commercial buildings. All but four sites had supply ducts located in the attic. Sites 3, 7, and 17 had supply ductwork installed in conditioned space and Site 16 had ductwork installed in an outdoor closet. Sites 2 and 8 had return ducts located partially in a garage. The median system had a rated cooling capacity of 10.6 kW and 15 out of the 17 AHU fans had permanent split capacitor (PSC) motors, which is approximately the same 90% market share as the U.S. average [30]. In total, 114 useful monthly visits were made during the cooling season, providing 3132 h (most visits were longer than 24 h) of data collection measured at a median outdoor temperature of 27.9 °C.

The last column of Table 2 shows that the median hourly fractional operation time across all systems was approximately 20.6%, or 12.4 min per hour (the data were lognormally distributed with a GM of 22.8%, or 13.7 min per hour, and a GSD of 1.64). Even in the warm climate of Austin, TX, these cooling systems did not operate very often on average, but large standard deviations from individual mean operational fractions reveal a wide spread in hourly operation fractions in the test systems. Median cycle lengths across all systems and all cycles were 8.0 min (N = 3736, with an interquartile range of 5.7–11.7 min). The longest cycle length was almost 20 h. Outdoor temperature, indoor loads, cooling capacities, and occupant

Table 2
Building and individual HVAC system characteristics

Site	Year Built	Floor Area, m ²	Volume, m ³	Rated Capacity, kW _{cap}	Number of Cooling Visits	Total Monitored Cooling Hours	Mean Outdoor Temperature (std. dev.), °C	Mean Hourly Fractional Operation (std. dev.), %
1	1975	170	442	14.1	6	177	27.5 (1.9)	16.5% (17.5%)
2	1973	133	323	10.6	6	183	28.7 (2.3)	10.7% (21.6%)
3	1999	100	346	8.8	8	219	27.9 (2.9)	11.1% (13.0%)
4		30	108	5.3	8	232	27.9 (2.8)	24.8% (25.0%)
5	1949	106	292	8.8	8	225	27.7 (3.8)	39.4% (31.4%)
6	1941	139	340	10.6	6	165	28.6 (2.6)	32.6% (28.2%)
7	1970s ^a	111	272	10.6	7	206	29.7 (3.2)	20.6% (16.9%)
8	1984	125	323	10.6	6	168	28.6 (3.4)	32.9% (39.9%)
9	1995	121	439	17.6	6	177	27.3 (2.9)	18.4% (22.5%)
10		121	439	12.3	6	174	26.2 (2.6)	15.5% (20.4%)
11	1940	123	351	12.3	8	214	26.6 (3.5)	55.3% (33.8%)
12	1935	173	422	17.6	4	100	29.0 (4.0)	20.1% (23.6%)
13	1920	133	346	12.3	9	221	26.2 (3.3)	41.0% (38.5%)
14	1941	91	221	10.6	7	176	28.3 (3.1)	34.7% (35.4%)
15	1970s ^a	93	232	8.8	7	186	28.4 (3.0)	33.3% (43.2%)
16	2000s ^a	71	266	5.3	6	155	27.8 (3.7)	13.8% (28.5%)
17		26	59	5.3	6	154	27.8 (3.7)	13.7% (23.1%)
				Total	114	3132	Median: 27.9	20.6% (25.0%)

^a Estimated year built.

thermostat settings are some typical drivers of cycle lengths and fractional operation times, some of which are explored in later sections.

The following sections explore measurements of operational parameters most closely related to the following HVAC system components: AHUs, outdoor units, ducts, and occupant effects and overall performance.

4.2. AHU operation

4.2.1. Airflow rates

Fig. 2 shows the range of system airflow rates measured in each test system, normalized by rated cooling capacity. Each data point represents a monthly visit during the cooling season with any type of filter installed.

The median airflow rate measured across all systems was 176 m³ h⁻¹ kW⁻¹, with median airflow rates of 187 m³ h⁻¹ kW⁻¹ and 154 m³ h⁻¹ kW⁻¹ with low- and high-MERV filters installed, respectively. The low-MERV airflow rates were lognormally distributed with a GM of 194 m³ h^{-1} kW⁻¹ and a GSD of 1.42. Median airflow rates in individual systems were in the range of those recommended by manufacturers at one site, below at 9 sites, and above at 7 sites. Sites 16 and 17 had high airflow rates because their electronically-commutated motor (ECM) fans operated at higher speeds during the cooling mode. The wide range in airflow rates measured at Site 17 may be caused by inaccurate flow measurements because the unit had operating pressures near the lower limit of sensitivity of our instrumentation. For reference, Parker et al. (1997) measured a mean airflow rate of 155 m³ h⁻¹ per kW of rated cooling capacity in 27 residential systems in Florida (ranging from 63 to 247 m^3 h⁻¹ kW⁻¹) [10] and Proctor (1997) measured a mean airflow rate of 166 m^3 h^{-1} kW⁻¹ in 28 new residential systems in Arizona [11].

4.2.2. Recirculation rates

Table 3 shows recirculation rates estimated for the volumes that each individual system served, calculated using mean airflow rates measured across all filter installations with and without incorporating the mean fractional operation times from Table 2.

The median individual system recirculation rate was approximately $6 h^{-1}$ assuming the systems ran 100% of the time and 1.5 h^{-1} when averaged over the mean operation time. These rates, when

accounting for duty fraction, are considerably lower than some of those used in other investigations [15,17,20]. For comparison with typical air exchange rates, Murray and Burmaster (1995) reported air exchange rates in 2844 existing residences with an interquartile range from 0.32 to 0.87 h⁻¹ (median of 0.51 h⁻¹) [44]. Limiting values to those measured in the summer in Arizona, Florida, and (mostly) California, median air exchange rates were 1.10 h⁻¹ (with an interquartile range of 0.58–1.98 h⁻¹). More recently, Offermann (2009) reported median air exchange rates of 0.26 h⁻¹ in 108 new homes in California [45]. Air exchange rates were not measured in our study, but our estimated recirculation rates (median 1.5 h⁻¹) suggest that HVAC systems, even at low duty cycles, should be competitive as pollutant removal mechanisms relative to air exchange rates, depending on filter efficiency, filter bypass, duct leakage, window opening behavior, and individual system runtime.

Table 3 assumes that the individual systems contained within the same building (Sites 3 and 4 and Sites 16 and 17) act completely independently of each other. If those systems acted as one, operating at the same time and serving a completely mixed building volume, the volume-weighted average whole-house recirculation rates



Fig. 2. System airflow normalized by rated cooling capacity ($m^3 h^{-1} kW^{-1}$) measured at each monthly visit during the cooling mode (n = 114 visits). The dashed lines correspond to the range of airflow rates typically recommended by manufacturers (169–193 $m^3 h^{-1} kW^{-1}$). Boxes describe 25th, 50th, and 75th percentiles; whiskers describe 5th and 95th percentiles.

Table 3				
Estimated	individual	system	recirculation	rates

Site	When Operating, hr^{-1}	Averaged Over a Day, hr ⁻¹
1	4.6	0.8
2	5.3	0.6
3	4.6	0.5
4	10.8	2.7
5	6.4	2.5
6	4.9	1.6
7	4.3	0.9
8	3.8	1.3
9	7.9	1.5
10	4.7	0.7
11	6.8	3.8
12	6.0	1.2
13	4.3	1.7
14	7.0	2.4
15	8.3	2.8
16	6.7	0.9
17	32.5	4.5
Mean (std. dev.)	7.6 (6.7)	1.8 (1.2)
Median	6.0	1.5

would be 6.1 h⁻¹ and 11.4 h⁻¹, respectively, excluding duty cycle effects. In reality, airflows and duty cycles of those systems interact with each other, making them neither completely dependent on, nor independent of, each other. It should also be noted that the estimations in Table 3 assume zero duct leakage. Return duct leakage would provide a smaller fraction of recirculated air and effectively lowers the recirculation rate. Supply leakage would also decrease the recirculation rate estimate. Because most of the airflow measurements were taken at the air handler, supply leakage represents an unaccounted volume loss. Duct leakage on either side would also lead to increased air exchange rates because of the interaction of building pressurization with infiltration [26,36].

4.2.3. Fan power draws

Fig. 3 shows the ranges of values of fan efficacy measured during each monthly visit at each of the test sites in the cooling mode, defined as the system airflow divided by the power draw of the AHU fan.

Median efficacy values across all sites were approximately 3.4 m³ h⁻¹ W⁻¹, ranging from 2.0 to 6.6 m³ h⁻¹ W⁻¹. Rated filter efficiency had a small effect on fan efficacy values. Low-MERV and high-MERV filters had median efficacy values of 3.5 m³ h⁻¹ W⁻¹ and 3.3 m³ h⁻¹ W⁻¹, respectively. The median fan power draw across all sites with all filters was 519 W, ranging from 312 to 1040 W. These wide ranges of fan efficacy and power draw values are similar to those reported in Ref. [31,46,47], suggesting that AHU fans are similar across multiple locations in the current building stock. The widest range in efficacy was observed with the ECM fan at Site 17, although this variation is likely due to variations in the flow measurements previously discussed.

4.3. Condenser-compressor unit operation

4.3.1. Cooling capacities

Fig. 4 compares total cooling capacity (sensible + latent) to manufacturer-rated (nominal) cooling capacity of the outdoor unit, measured at four ranges of outdoor temperature: 20-25 °C, 25-30 °C, 30-35 °C, and 35-40 °C. For comparison, the AHRI standard 210/240 test for rating air-conditioning equipment in the cooling mode calls for testing outdoor compressor units at outdoor temperatures of both 28 °C and 35 °C. The bars represent the mean value calculated from measurements at steady-state operation and the error bars represent one standard deviation in each direction.



Fig. 3. Fan efficacy ($m^3 h^{-1} W^{-1}$) measured at each monthly visit during the cooling mode (n = 114 visits). Boxes describe 25th, 50th, and 75th percentiles; whiskers describe 5th and 95th percentiles. The dashed horizontal line represents the overall median value across all sites.

Cooling capacity was estimated by measuring the airflow rate through the AHU, the differences in temperature and humidity ratio across the cooling coil, and assumed constant values of air density, specific heat of air, and the latent heat of vaporization of water as described in Ref. [21].

Site 14 was excluded from this analysis because the outdoor condenser-compressor unit was replaced midway through the testing period. The mean total capacity of each system in all but one combination of site and temperature bin (Site 16, outdoor temperature 30-35 °C) was less than 100% of rated capacity. The mean percentage of rated capacity across all sites was 62%, 64%, 67%, and 67% for each outdoor temperature bin (20-25 °C, 25–30 °C, 30–35 °C, and 35–40 °C, respectively), which generally agrees with values from previous field studies [33]. The low relative values suggest that the majority of the test systems do not operate at rated capacity, which has implications for thermal comfort and energy consumption as equipment will operate longer than necessary in order to meet cooling loads. Longer operation times will also increase recirculation rates, which may positively impact IAQ as previously discussed. However, the median cycle length described above (8 min) is similar to the mean runtimes by correctly-sized units in James et al. (1997) [9], which suggests that systems were correctly sized relative to our test conditions and that low delivered capacity may have been accounted for in the design.



Fig. 4. Measured versus rated (nominal) capacity measured in 5 °C bins of outdoor temperatures; means are reported only if a temperature range had at least 50 data points recorded within its bin.

4.3.2. Outdoor unit power draw

Because knowledge about how outdoor condenser—compressor units perform outside of standard rating conditions is generally lacking [48], Table 4 describes the increase in the power draw of 16 of the 17 outdoor units (compressor + outdoor fan power) as a function of outdoor temperature (Site 14 is excluded again because of the replacement of the outdoor unit during the test period).

Outdoor unit power draws were averaged during outdoor temperature bins of 1 °C (ranging from 21 °C to 41 °C) and a minimum of 100 data points were required in each bin. A linear regression was performed with power draw as the dependent variable versus each outdoor temperature bin as the independent variable. According to the regression slopes, the median increase in outdoor unit power draw was 1.6% per °C rise in outdoor temperature. Most of the coefficients of determination (R^2) were relatively close to unity, excluding Site 1, which had a two-stage compressor that operated and different speeds as needed, thus showing a nonlinear response. The power draw response to outdoor temperature of the systems generally fell within the range of those reported in other studies. Proctor (1998) (and references therein) reported that the energy efficiency ratio (EER) of a typical condenser-compressor unit decreased approximately 2.2% per °C increase in outdoor temperature [49]. More recently, Kim et al. (2009) reported that the compressor power draw of an 8.8 kW (SEER 13) residential heat pump increased approximately 2.9% per °C increase in outdoor temperature and was only a very weak function of indoor conditions [50].

4.4. Ducts

4.4.1. Duct leakage

Fig. 5 shows mean supply and return duct leakage fractions to the exterior of the building envelope measured at each site. Values for Sites 10 and 17 are not present because duct leakage tests were not performed at Site 10 due to scheduling conflicts and Site 17 had ducts located entirely inside conditioned space (exterior leakage was not measured).

The median supply and return duct leakage fraction across all sites where duct leakage testing was performed was 8% and 4%, ranging from 0% to 33% and from 0% to 17%, respectively. For comparison, Jump et al. (1996) reported mean supply and return leakage rates of 18% and

Table 4

Regression results of steady-state outdoor unit power draw versus outdoor temperature.

Site	Power Draw Increase per °C Rise in Outdoor	R ² of Linear Regression	95% C.I.
	Temperature		
1	3.9%	0.68 ^a	1.7-6.0%
2	1.5%	0.97	1.3 - 1.7%
3	2.5%	0.99	2.3-2.6%
4	2.7%	0.98	2.4 - 2.9%
5	1.4%	0.99	1.3-1.5%
6	1.0%	0.87	0.7-1.2%
7	0.7%	0.96	0.6-0.8%
8	2.1%	0.91	1.7-2.5%
9	1.7%	0.98	1.5-1.8%
10	2.7%	0.99	2.6-2.8%
11	1.3%	0.97	1.2-1.5%
12	2.3%	0.98	2.1-2.6%
13	1.9%	0.95	1.7-2.2%
15	1.1%	0.86	0.9-1.3%
16	1.1%	0.90	0.9-1.4%
17	1.3%	0.89	1.0-1.5%
Mean	1.8%		
Std. dev.	0.8%		
Median	1.6%		

^a Site 1 was the only system with a variable speed compressor and operated at two stages.

17%, respectively, in 27 residential systems in California [22]. In 28 new residential systems in Arizona, Proctor (1997) measured mean supply and return duct leakage of 9% and 5%, respectively [11]. More recently, Offermann (2009) reported median duct leakage of 10% in 138 systems in 108 new homes in California [45].

4.4.2. Operating pressures and pressure drops

An important parameter in determining the effect that an individual component has on airflow rates in an HVAC system is the fraction of total system pressure drop that can be attributed to that component. Fig. 6 shows the range of fractions of system pressure drop measured across three components at each test site in the cooling mode: low-MERV filters, high-MERV filters, and cooling coils. Because filters were left in place for three months, these filter measurements capture the effects of both initial filter design and dust loading while in use. Coil measurements are taken across all filter installations because there was no significant difference in coil pressure drop observed between filters, although there is considerably uncertainty in some of our coil measurements because of difficulties in locating pressure taps in some systems [21]. The median fractional pressure drops due to low-MERV filters, high-MERV filters, and coils across all sites were 21.5%, 31.4%, and 35.9%, respectively, as indicated by the three dashed lines. Fractional pressure drops across low-MERV filters were normally distributed with a mean (and standard deviation) of 23.6% (11.6%) and high-MERV filter pressure drops were lognormally distributed with a GM (GSD) of 31.6% (1.37). Coil pressure drops were neither normally nor lognormally distributed.

The overall median fraction of pressure drop across cooling coils was larger than the overall median pressure drop across either lowor high-MERV filters, which may help explain the lack of significant differences in energy consumption observed due to higherefficiency filters in [21], albeit with considerable uncertainty. Pressure drops across low- and high-MERV filters ranged from 2 to 174 Pa and from 37 to 145 Pa with medians of 35 Pa and 71 Pa, respectively. The wide range is due to variations in filter selection, individual system design, and filter dust loading (which is related to occupant activity, filter efficiency, system runtimes, indoor particle sources, the penetration efficiency of outdoor particles, and potentially return duct leakage). Similarly, coil pressure drops ranged widely from 3 to 192 Pa (with a median of 58 Pa), although the smallest values are likely due to unreliable pressure measurements.

Median return plenum operating pressures, measured with respect to ambient indoor pressure and including the pressure drop across the filter, were -63 Pa and -97 Pa with low- and high-MERV filters installed, respectively (ranging -14 to -174 Pa and -39 to -208 Pa). Return plenum operating pressures were lognormally distributed with high-MERV filters installed, with a GM (GSD) of 93 Pa (1.50). Median supply plenum pressures were 38 Pa and 32 Pa



Fig. 5. Supply and return duct leakage-to-the-exterior fractions (as a percent of total airflow rate) measured during the cooling mode.



Fig. 6. Filter and coil pressure drops as a fraction of total system pressure drop, measured at each monthly visit during the cooling mode. Dashed lines represent median fractional pressure drops for each of the three components: low-MERV filters, high-MERV filters, and cooling coils. Boxes describe 25th, 50th, and 75th percentiles; whiskers describe 5th and 95th percentiles ($n_{low-MERV}$ filter = 43; $n_{high-MERV}$ filter = 53; n_{coil} = 112).

measured with low- and high-MERV filters installed, respectively (ranging 4–79 Pa and 2–72 Pa). Inter-home variability was greater than intra-home variability with two different filters installed and the measured values of supply and return plenum operating pressures fall generally within the range of those reported by other studies. For example, Modera (1993) reported a mean supply plenum pressure relative to the occupied space of 46 Pa (ranging from 9 to 138 Pa) and a mean return plenum pressure of -88 Pa (ranging from - 14 to - 181 Pa) in 31 homes [36]. Parker et al. (1997)measured mean total system pressures of 112 Pa and 157 Pa in six new and eight existing residential systems in Florida, respectively [10]. Francisco et al. (1998) reported mean supply and return plenum pressures of approximately 50 and -58 Pa, respectively, in six residential heating systems [37]. Proctor and Parker (2000) reported total system pressures of 102–137 Pa from several studies, as measured across duct systems, registers, and filters, excluding that associated with the cooling coils [31].

4.4.3. Filter lifespan

Measured changes in pressure drop are directly related to changes in airflow rates. Because filters were typically left in place for three months at a time and building occupants operated their systems as usual, we are able to observe real-life loading of filters. Out of 64 filter installations (excluding Site 12, which had high-MERV filters installed on a different rotation schedule during the entire test period), filters were loaded enough (i.e., filter pressure drops were increased enough) to cause at least a 10% decrease in fan-only mode airflow rates (a measure that was conducted at every monthly visit, regardless of season) in only 11 installations (17%). Twice this occurred with a low-MERV filter, five times with a mid-MERV filter, and four times with a high-MERV filter. The 10% decrease in airflow is an arbitrary threshold, although it has been shown that decreases in airflow of up to 10% have not generally had large energy impacts [10,21,39]. Twice a filter pressure drop passed this threshold within one month, five times within two months, and four times within three months. Many filter manufacturers recommend replacing filters every 90 days, however, our results suggest that filter replacement schedules should be determined independently for individual systems based on operation time, system and building characteristics, and occupant activity levels.

4.4.4. System environmental conditions

Because supply plenum, supply register, and return plenum temperatures and humidity ratios were measured at each site during steady-state operation in the cooling mode, we can investigate the differences in those parameters across a variety of components within the air-conditioning systems. For example, the mean (\pm std. dev.) steady-state supply plenum, supply register, and return plenum temperatures across all sites were 14.3 \pm 3.1 °C, 18.0 \pm 3.1 °C, and 24.5 \pm 1.6 °C, respectively, which corresponds to a mean temperature rise in supply ducts of approximately 3.6 ± 2.9 °C and a mean temperature decrease across the AHU (fan + coil) of approximately 10.2 \pm 2.6 °C. Temperature gains in supply ducts due to conduction and conditioned air losses because of duct leakage were likely a significant source of cooling capacity degradation in these systems. Although the supply register measurements were made only at one register and may not represent the temperature delivered from every register, temperature increases in supply ducts would result in a mean heat load from the duct system of approximately 2.2 \pm 1.8 kW, or 17 \pm 12% of rated cooling capacity. The mean increase in supply ducts of 3.6 °C was nearly two times greater than the nearly 2 °C rise in temperature measured in a single residence from the upstream portion of a repaired supply duct passing through an unconditioned attic to a supply register on a hot day by Parker et al. (1993) [34]. The mean decrease of 10.2 °C across AHUs is comparable to a 10 °C difference under normal operating conditions in [34] and an 11.0 °C temperature differential across the evaporator coil measured in laboratory tests of a 12.3 kW residential unit at standard conditions [51].

The mean (±std. dev.) steady-state return plenum and supply register humidity ratios across all sites were $10.0 \pm 1.6 \text{ g kg}^{-1}$ and $8.8 \pm 1.6 \text{ g kg}^{-1}$, respectively. The combination of return ducts, cooling coils, and supply ducts provided dehumidification to reduce the mean indoor humidity ratio by approximately $1.2 \pm 0.9 \text{ g kg}^{-1}$. Using the median measured values for airflow rates, temperature differences, and humidity ratio differences, latent capacity accounted for approximately 20% of total capacity in the test systems, on average (equivalent to a sensible heat ratio, or SHR, of approximately 0.8). The median measured SHR was on the upper end of those typically reported in residences [28].

4.5. Occupant influences and overall performance

4.5.1. Fractional operation times

This section explores key factors that affected system operation fractions (i.e., duty cycle) in the test systems. First, Fig. 7 describes how operation time increases in response to both outdoor temperature and indoor-outdoor temperature differentials, using Spearman's rank correlation coefficients (a non-parametric measure of statistical dependence) for each full hour of cooling cycles observed across all sites (N = 3070). Then linear regressions of hourly duty fraction are performed versus the difference between the mean hourly outdoor and indoor temperatures.

Hourly fractional operation times were more strongly correlated with differences between outdoor and indoor temperatures $(\rho = 0.66)$ than outdoor temperature alone $(\rho = 0.50)$ across all sites. The median increase in hourly operation fraction is approximately 6.0% per °C increase in indoor-outdoor temperature difference, ranging from 2.4 to 11.3% per °C per site. Coefficients of determination (R^2) from the table in Fig. 7 range between 0.6 and 0.8 for 14 of the 17 sites, suggesting that approximately 60-80% of the variation in hourly duty fraction can be explained by indoor-outdoor temperature differences for most of the test systems. For comparison, Thornburg et al. (2004) reported an approximately 1.8% increase in operation time per °C increase in daily mean outdoor temperature with similar confidence in their correlations ($R^2 = 0.61$), although their measurements occurred during relatively mild climate conditions (daily mean temperatures during cooling operation ranged from approximately 17 °C–27 °C) [41]. Other factors that can affect duty fractions include the relative of size of the system capacity compared to the cooling load, indoor heat gains, and the insulating properties of the building envelope.

To explore some other important factors known to affect fractional operation times of systems, Table 5 shows Spearman's rank correlation coefficients between the mean hourly duty fractions from Table 2 against six independent variables of interest measured at each site: return leakage fraction, supply leakage fraction, system size, mean outdoor temperature, mean airflow rate, and mean indoor endpoint temperature (a surrogate for thermostat settings). A Spearman's rank correlation coefficient (ρ) is a non-parametric measure of statistical dependence between two variables that is appropriate for small sample sizes. A value of +1 for ρ establishes a perfect direct relationship and a value of -1 establishes a perfect inverse relationship between the two variables.

Table 5 shows the strongest association with mean hourly duty fraction is mean indoor endpoint temperature ($\rho = -0.797$). Endpoint temperatures are a surrogate for thermostat settings and were flagged in the dataset as the temperature in the return plenum measured at the end of an air-conditioning cycle when the thermostat is satisfied and the outdoor unit terminates operation. Treating indoor endpoint temperatures independently, there is less than a 1% probability that duty fractions are not associated with indoor endpoint temperatures. The negative association between mean indoor endpoint temperature and operation time is intuitive: a lower thermostat set point will increase runtime. The next strongest association with mean hourly duty fraction is supply duct leakage ($\rho = 0.482$). There is only an approximately 6% chance that supply leakage fraction and duty fraction are independent. The two variables are intuitively positively associated as energy wasted due to supply leakage cause longer runtimes.

Mean hourly duty fraction appears to have the weakest association with return duct leakage ($\rho = 0.057$) and system size normalized by floor area served ($\rho = 0.138$). Duty cycle fractions appear to be negatively correlated with mean outdoor temperature ($\rho = -0.267$), but their probability of independence is greater than 50% and the differences in outdoor temperatures are small. Higher duty fractions were negatively correlated with airflow rates ($\rho = -0.200$), suggesting systems ran longer with lower mean airflow rates. However, the association is not particularly strong (probability of independence of 42%), which emphasizes the negligible effect of higher-efficiency filters found in [21].

Interestingly, the correlations emphasize the potential importance of supply leakage relative to return leakage. However, the lack of association of return leakage with operational fractions may have occurred because return leakage fractions were generally small in these systems. Previous studies have shown that excessive return



Increase in hourly duty fraction per °C rise in average hourly indooroutdoor temperature difference

outuooi	temperature unierence					
Site	%per °C	R ²	N (hours)			
1	6.0%	0.71	175			
2	3.7%	0.33	180			
3	2.9%	0.68	215			
4	7.2%	0.68	226			
5	9.3%	0.69	222			
6	9.1%	0.80	161			
7	4.7%	0.71	204			
8	7.3%	0.62	164			
9	6.0%	0.69	175			
10	4.9%	0.73	171			
11	11.3%	0.67	211			
12	4.5%	0.68	91			
13	7.9%	0.61	218			
14	7.1%	0.78	173			
15	9.2%	0.63	182			
16	4.0%	0.41	152			
17	2.4%	0.22	150			
Average	6.3%	Total	3070			
Median	6.0%					

Fig. 7. Hourly duty cycle response to climate conditions.

Table 5

Spearman's rank correlation coefficients for mean duty cycle fraction.

Mean Duty Cycle Fraction	Return Leakage Fraction	Supply Leakage Fraction	System size, Normalized by Floor Area	Mean Outdoor Temperature	Mean Airflow Rate, Normalized by Rated Capacity	Mean Indoor Endpoint Temperature
Spearman Correlation Coefficient, ρ	0.057	0.482	0.138	-0.267	-0.200	-0.797
Probability of Independence	83.3%	5.8%	75.0%	55.0%	42.2%	0.8%

duct leakage can lead to substantial energy penalties [24,36]. The correlations also intuitively suggest that system runtime is associated more closely with thermostat settings than any of the other variables. This suggests that those concerned with reducing energy consumption in residential air-conditioning systems in similar climates may prioritize increased thermostat settings and supply duct sealing, although further proof is warranted in more systems. Increased thermostat settings would only address sensible loads and could lead to moisture and comfort problems, especially in this hot and humid climate. There are also many other ways to reduce energy consumption in residential and light-commercial buildings, including reducing heating and cooling loads by building envelope improvements, increasing appliance and equipment efficiency and installation, and addressing occupant behavioral patterns. Ultimately, these results cannot be considered conclusive, as the variables of interest are not necessarily entirely independent of each other. However, this exploratory analysis provides an indication of the important parameters affecting duty cycle fractions and the methods should be used in larger samples.

4.5.2. Occupant thermostat settings

Fig. 8 shows distributions of minimum indoor temperatures reached during air-conditioning cycles in the test systems. Actual thermostat set points depend on the dead-band area and anticipation of each thermostat, or the range that the actual temperature is allowed to overshoot the set temperature to avoid rapid oscillations in cycling. Dead-band values are generally assumed to be 0.5–1 °C, although little is known about actual values and accuracies. Fig. 8a shows a histogram and cumulative distribution function of minimum indoor temperatures reached for each cycle measured across all sites (weighting all data equally). Fig. 8b is a box plot of minimum indoor temperatures reached at each site, along with the number of cycles at each site used in the plot.

The median end-of-cycle indoor temperature recorded was 24.8 °C across all sites, with the 25th and 75th percentiles falling between 23.5 °C and 25.7 °C, respectively. Given the likely dead-band values of 0.5-1 °C, the median thermostat setting for all of the 17 test systems can be estimated to be between 25 °C and 26 °C. These values are in general agreement with many rule-of-thumb

values and those recommended by governmental agencies. However, Fig. 8b shows that a wide variation exists across individual sites in our study. Median endpoint temperatures between individual sites ranged from approximately 22.5 °C to over 27 °C, with light-commercial sites having statistically significant lower thermostat settings (Mann–Whitney–Wilcoxon P < 0.0001).

Finally, Fig. 9 shows mean fractional operation times in response to both time of day and outdoor temperature. Values are averaged for each hour of the day in the study and across all residential and light-commercial systems in the study. Error bars represent one standard deviation in each direction.

Operational times generally trend with outdoor temperature as the systems respond to meet the coincident cooling load. Mean hourly fractional operation times are similar between residential and light-commercial systems from 5 PM to 7 AM. However, lightcommercial systems ran up to 30-150% more often than residential systems during typical business hours (10-30% more absolute time from 8 AM to 4 PM). Assuming constant airflow rates and aircleaner efficiencies, longer operation times lead to greater recirculation rates. Thus, if in-duct air cleaners or filters in HVAC systems are relied upon to deliver clean air to occupied spaces, these results suggest that occupants may be more protected from indoor airborne pollutants by longer operation times in lightcommercial buildings than in residences in this sample. However, this relationship is only true if other parameters are held constant, including indoor pollutant sources, penetration of outdoor pollutants, air exchange rates, deposition rates, and indoor volumes. Additionally, the filters used in these systems are designed only to capture particulate matter. No additional protection would be offered against gas-phase pollutants.

5. Limitations

One limitation of this investigation is that the test systems were chosen as a sample of convenience and not necessarily as a representative sample of all small systems in the U.S. However, the test systems varied widely in age, size, efficiency, and operational characteristics, which is typical for the U.S. building stock. Another limitation of this study is that the measurements herein focus only



Fig. 8. Distribution of minimum indoor temperatures reached at the end of cycles for (a) all sites (N = 3658 cycles) and (b) each site individually (with the number of cycles per site).



Fig. 9. Mean hourly fractional operation time, averaged across 8 residential and 9 light-commercial systems in this study.

on cooling system operation in a hot and humid climate, which will differ from fan-only and heating operation, and from cooling operation in other types of warm climates. According to ASHRAE Standard 169, Austin has 938 annual heating degree-days (HDD, base temperature of 18 °C) and 3984 annual cooling degree-days (CDD, base temperature of 10 °C) [42]. Although not representative of the entire U.S., the size of the population that lives in climate zone 2a in the U.S. is approximately 33 million (~11% of the population) [52]. Many of the variables measured herein fall in the same ranges as those measured in other parts of the country.

Given the shortfalls of many actual operational characteristics relative to design or ideal conditions measured herein and in other studies, we recommend that our collection and analysis methods be used to collect similar data across more locations in the U.S. to capture the effects of other climates, construction practices, and occupant behavior. A fully assembled dataset of similar measurements across the U.S. building stock can provide further insight into how residential and light-commercial HVAC systems use energy and affect IAQ.

6. Conclusions

This paper strengthens the knowledge base of smaller HVAC systems by characterizing a variety of operational characteristics measured in 17 existing residential and light-commercial air-conditioning systems in Austin, TX. We report an analysis of a previously collected dataset of a variety of measurements taken over 3100 h of air-conditioning operation, characterizing key operational characteristics and exploring factors that affect building energy consumption and IAQ. Key findings include:

- Measured airflow rates were outside of the range recommended by most manufacturers for almost every system.
- Recirculation rates are considerably lower than values used in many other lab and modeling studies, although recirculation through AHUs was still likely competitive with air exchange rates as a removal mechanism for indoor pollutants.
- Actual measured cooling capacities were only 62–67% of rated cooling capacities on average.
- Filter pressure drops increased enough during 3 months of dust loading to decrease airflow rates at least 10% in only 17% of filter installations.
- Hourly fractional operation times increased approximately 6% for every °C increase in indoor—outdoor temperature difference.

- Mean indoor endpoint temperatures (a surrogate for thermostat settings) and supply duct leakage fractions were most associated with longer operation times.
- There was a wide distribution in indoor endpoint temperatures across individual sites, and light-commercial systems generally had lower thermostat settings and longer operation times during certain parts of the day, on average.

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