ENERGY CENTER OF WISCONSIN

ECW Report Number 241-1

Central Air Conditioning in Wisconsin

A compilation of recent field research

May 2008, emended December 15, 2010

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This publication was funded in part by the State of Wisconsin, Wisconsin Department of Administration, Division of Energy, and was funded through the Focus on Energy Program. Focus on Energy is a public-private partnership offering energy information and services to energy utility customers throughout Wisconsin. The goals of this program are to encourage energy efficiency and renewable energy, enhance the environment and ensure the future supply of energy for Wisconsin. (800.762.7077, focusonenergy.com)

This publication is also based in part on work funded by the State Technologies Advancement Collaborative (STAC) administered by the National Association of State Energy Officials (NASEO). The STAC project, *Closing the Gap:Getting Full Performance from Residential Central Air Conditioners* included the following funding partners: New York State Energy Research and Development Authority (NYSERDA), the Florida Solar Energy Center (FSEC) and Advanced Energy of North Carolina, with supplemental funding from the California Energy Commission. The American Council for an Energy Efficient Economy (ACEEE) also participated in this effort.

Project Manager

Scott Pigg

Acknowledgements

A number of individuals directly contributed to the various research efforts documented in this report. These include the following:

Greg Dalhoff, Dalhoff and Associates, assisted with fieldwork for the STAC two-stage and sizing-swap samples, as well as preliminary data analysis for the two-stage sites.

John Viner, Building Services and Consultant, conducted the fieldwork for the 2005 field testing and monitoring component of the STAC project, led the fieldwork team for the 2007 Focus on Energy study, and conducted the high-intensity site fieldwork for that study.

Dan Cautley, Cautley Engineering, designed and constructed the monitoring rigs for the 2007 Focus on Energy study, provided on-going troubleshooting throughout the fieldwork, and assisted with on-site fieldwork.

Ingo Bensch, Energy Center of Wisconsin, managed the 2003 Appliance Sales Tracking Project, and the random-digit-dial recruiting for the 2007 Focus on Energy field study, which was implemented by **Leede Research** of Manitowoc Wisconsin.

John Flanagan, Western Wisconsin Technical College, **Cheyne Herro,** Van's Heating and Cooling of De Pere, Wisconsin (with in-office assistance from **Tiffani Hussin**), and **Scott Bradley**, Cardinal Heating and Cooling of Madison Wisconsin, conducted on-site testing for the 2007 Focus on Energy Study.

Sherry Benzmiller and **Jaime Barbian**, Energy Center of Wisconsin, provided data compilation and other support for the 2007 Focus on Energy project.

Scott Pigg, Energy Center of Wisconsin, managed the STAC and Focus on Energy research projects, contributed to fieldwork efforts, conducted the data analysis, and prepared this report.

Also, Wisconsin Energy Conservation Corporation staff provided valuable assistance and direction in shaping both the 2005 STAC and the 2007 Focus on Energy research, notably: **Wayne Deforest**, **Jill Cornelius**, **Sara Van de Grift** and **Ed Carroll**.

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REPORT SUMMARY

The report summarizes the results of several field studies involving residential central air conditioners in Wisconsin. The studies include:

- A 2007 study involving field measurements before and after making airflow and refrigerant charge corrections—and in some cases cleaning condenser coils;
- A 2005 field assessment of refrigerant charge, airflow and other parameters of new SEER 13+ systems;
- Field monitoring and experimental control of two-stage systems over the course of the 2004 – 2005 cooling seasons;
- Detailed monitoring at two sites where over-sized 3-ton systems where replaced with identical 2-ton systems to assess the impact of sizing on energy and indoor comfort; and,
- A large-sample 2003 telephone survey of air conditioning use the previous day.

A number of observations and conclusions can be made from the data from these studies:

THE WISCONSIN CENTRAL AIR CONDITIONING MARKET

- Somewhere between two-thirds and three-fourths of Wisconsin single-family homes have central air conditioning, and 2 to 3 percent of homes add central AC each year.
- The market share for high efficiency systems (SEER 14+) increased concurrently with introduction of the federal SEER-13 efficiency standard, and currently stands at about 15 percent.

SYSTEM SIZING

- The majority of Wisconsin central air conditioners have between 2 and 3 tons of capacity.
- While Manual J calculations indicate that most systems are appropriately sized to within ½ ton, monitoring data suggest that most systems are in fact oversized: empirically-based sizing estimates indicate that about a quarter of systems are appropriately sized, a third are oversized by ½ ton, and 40 percent are oversized by a ton or more. Only one of 39 systems evaluated appeared to be undersized.
- Experiments at two homes in which otherwise identical 2- and 3-ton systems were installed and monitored yielded inconclusive results as to whether down-sizing saves energy or affects humidity control. One of the sites showed no difference in weather-normalized energy consumption: reduced power requirements were almost exactly offset by increased run time. The other site showed some energy savings, but the difference was not statistically significant. The latter site also had higher indoor humidity with the smaller system, likely due to the fact that airflow provided by a 100kBtu/hr furnace could not be adjusted downward sufficiently to match

the 2-ton system. In contrast, the smaller system at the first site produced lower indoor humidity (relative to outdoor humidity levels) in hot weather.

BEHAVIORAL ASPECTS OF AC USE

- Most households operate their air conditioners at set points between 72 and 78°F, with an average setting of about 75°F.
- Both monitoring and survey data indicate that many households do not operate their air conditioners at times when cooling loads would otherwise warrant it. This discretionary use of air conditioners reduces operating hours by 25 to 33 percent on average.
- Monitoring data show numerous operating cycles for many sites in which the system runs 60+ minutes continuously after a period of four or more hours without operating at all. These events tend to be concentrated in the afternoon and early evening hours and many, if not most, likely result from occupants keeping the system turned off until later in the day. The prolonged recovery times from these events are estimated to add about 2.5 percent to diversified air conditioning electrical load on hot afternoons.

SEASONAL AND PEAK-DAY OPERATION OF AIR CONDITIONERS

- Average seasonal system operating time ranges from about 200 hours in northern areas to more than 400 hours in La Crosse—with an estimated statewide average of about 310 ± 50 hours. These estimates incorporate the effects of system over sizing and discretionary use of air conditioning by Wisconsin households.
- On afternoons that reach 90°F or higher, about one in five systems is not operating at all, and about 30 percent are running flat out; the rest are cycling on and off. The data suggest an overall average of about 50 percent duty cycle.

SAVINGS FROM TUNING AIR CONDITIONERS

- Before and after field EER measurements suggest that aggregate savings from tuning air conditioners is on the order of 5 ± 4 percent, a range that encompasses both tuning older existing systems and better installation practices of new systems. This averages blends many systems that have little or no potential for efficiency improvement from tune-up with a few systems where large savings can be achieved by correcting undercharged systems and reducing airflow.
- Field measurements across a range of refrigerant charge levels confirms that air conditioning systems that incorporate thermostatic expansion valves (TXVs)—which make up more than half of new systems sold in Wisconsin—are less susceptible to efficiency degradation from refrigerant charge errors.
- The diversified peak load impact from tuning air conditioners is unlikely to average more than about 50 Watts per system.

VARIABLE SPEED FURNACES AND AIR CONDITIONING

- Electrically efficient furnaces with electronically commutated motor (ECM) air handlers average about 190 watts less power per 1,000 cfm of airflow delivery than do standard PSC air handlers. This translates into about 170 ± 50 watts of diversified peak load reduction and 70 ± 20 kWh of seasonal electricity savings. Households that practice continuous-fan operation will see much higher savings.

- As-found measurements of airflow indicate that air conditioning systems with ECM furnace air handlers are no more likely to have appropriate airflow than standard furnaces with 3- or 4-speed PSC blower motors, despite the ability of ECMs to achieve a much wider range of airflow.

HUMIDITY CONTROL

- Among about 60 monitored homes, indoor humidity averaged about 47 percent during hours when the air conditioning system operated. Few homes showed average indoor humidity of more than 55 percent.

- Fewer hours of operation, higher building air leakage, and continuous-fan operation are all associated with poorer than average humidity control.

SEER RATING PROCEDURES AND WISCONSIN FIELD DATA

- Field data from Wisconsin central air conditioners suggests that the current test procedure for establishing SEER ratings does not reflect real-world operation in a number of respects:

    - Air handler static pressures and power requirements are considerably higher than those used in the current test procedure.

    - Typical cycling times are longer than those used to establish cycling performance degradation.

    - Indoor temperatures are lower than used in the test procedure.

    - The mid-load temperature used in the test procedure (82^{o}F) is slightly higher than the estimated seasonal average mid-load temperature for Wisconsin systems, which appears to average about 79^{o}F.

INTRODUCTION

Central air conditioning is an important and fast-growing electrical load among Wisconsin households. This report compiles the key findings from several recent research efforts to better understand the nature of central air conditioning electricity use in Wisconsin and explore the opportunities for improving the efficiency of this end-use.

This report covers a number of research efforts, ranging from field monitoring and testing, to compilation of distributor sales data, to telephone surveys of Wisconsin households. Three efforts for which results are presented here for the first time are as follows (in chronological order):

TELEPHONE SURVEY OF CENTRAL AC USE (2003)

During the summer of 2003, the Energy Center conducted a biennial telephone survey of appliance purchases, demographics and energy attitudes. As part of the effort that year, a subsample of respondents with central air conditioners (n=1,712) was queried about thermostat settings on the day prior to the survey. These data were later merged with weather station data, and used to better understand when and how people use their air conditioners under a range of outdoor conditions.

STAC PROJECT (2004-2006)

In collaboration with New York State Energy Research and Development Authority (NYSERDA), the Florida Solar Energy Center (FSEC), the American Council for an Energy Efficient Economy (ACEEE) and others, the Energy Center of Wisconsin received a federal grant under the State Technologies Advancement Collaborative (STAC) for a project titled "Closing the Gap: Getting Full Performance from Residential Central Air Conditioners." A key goal of the project was to explore improved regional performance of central air conditioners. Wisconsin's statewide Focus on Energy program provided additional in-kind funding through Wisconsin Energy Conservation Corporation for this project. The Wisconsin portion of this project involved three specific elements:

Field testing and monitoring of new systems

This effort was meant to gather better data on installation practices and use of new central air conditioners through on-site testing and subsequent monitoring of 50 new Wisconsin air conditioners. The sample focused on SEER 13+ systems that had received Focus on Energy rebates in the Madison, Wisconsin area. An initial sample of 13 systems monitored in 2004 turned out to be largely unusable due to an exceptionally cool air conditioning season. The results presented here focus on the 37 systems that were tested and monitored in 2005, which produced much more favorable conditions for cooling research.

On-site testing of these systems was primarily about checking refrigerant charge and air flow, but static pressures, air handler and compressor amps, and other parameters were also measured. The monitoring effort comprised tracking system on/off status and monitoring indoor temperature and relative humidity. In addition, data was gathered to allow for an independent system sizing analysis.

Field monitoring of two-stage systems

Under this task, two-stage central air conditioners were monitored to better assess the impact of two-stage operation on indoor comfort and efficiency. Twenty systems in the southern half of Wisconsin were originally targeted for monitoring, but the cool summer of 2004 led to extending the monitoring of some systems for two cooling seasons. Data collection issues further reduced the number of sites to 14.

Most of the systems were also subjected to special intervention in which the systems were forced to operate only on high-stage for a period of time. Some systems were forced to operate only on high-stage on some days of the week, while others were configured for two-stage operation for the first part of the summer and high-stage only operation for second part.

AC-sizing test homes

This task examined the comfort and energy efficiency implications of system sizing by directly measuring energy consumption and indoor psychrometrics in two homes with new oversized 3-ton air conditioners, then replacing these with more appropriate 2-ton systems. The original scope called for testing four homes, but Wisconsin's relatively short cooling season made it apparent that it would be a better allocation of the monitoring budget for this task to instead gather data over a longer time period for two sites.

FOCUS ON ENERGY FIELD RESEARCH PROJECT (2007)

This project was primarily concerned with assessing the savings from tuning refrigerant charge and airflow in Wisconsin central AC systems. A geographically stratified random sample of systems was recruited for three groups: (1) older systems; (2) new systems; and, (3) new, high efficiency (SEER 14+) systems. The first two groups were recruited via a random-digit-dial telephone procedure. The third group was obtained from a list of recent Focus-on-Energy rebate recipients. A total of 77 sites were evaluated.

Recruited sites were subjected to one of two protocols. For 61 *standard-protocol* sites, airflow and refrigerant charge were tuned sequentially while monitoring system energy input and cooling output. The remaining 15 sites were subjected to a day-long *high-intensity protocol*, which involved deliberately under- and over-charging the systems at different airflows while monitoring electricity consumption and output cooling. This protocol was used to better understand the effect of charge and airflow errors on system efficiency in a field context. Some older systems also received condenser coil cleans under both protocols.

Some of the standard-protocol sites also received data loggers to track post-tuneup system cycling and indoor psychrometrics.

In addition to the above projects, this report makes use of other air conditioning data that have been gathered and already reported separately. These include:

- **Furnace and Air Conditioner Tracking System (FACTS)** — this on-going effort (currently funded by Focus on Energy) has tracked the market for residential furnaces and central air conditioners in Wisconsin through cooperating equipment distributors (estimated to represent somewhat more than half of the Wisconsin market) since 1996.

- **Furnace Electricity Study** — this 2003 study (funded by Focus on Energy) looked at residential furnace electricity consumption for standard and two-stage systems with electrically-efficient electronically-commutated blower motors.

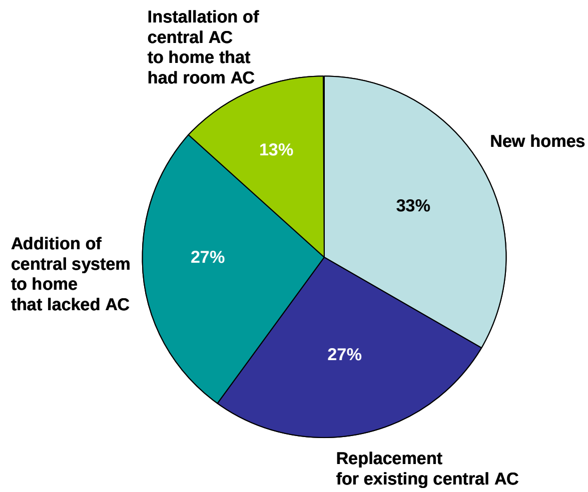
- **Survey of Fan Operation Practices** — as part of an evaluation of the savings from Focus on Energy rebates for electrically efficiency furnaces, Glacier Consulting conducted a telephone survey of furnace fan operation practices in 2003 and 2004.

The remainder of this report covers various topics related to the Wisconsin central air conditioner market, operating characteristics of these systems, and the potential for efficiency improvements.

THE WISCONSIN CENTRAL AIR CONDITIONING MARKET

In 1999, on-site audits of about 300 Wisconsin single-family, owner-occupied homes showed that just over half (53 ± 7 percent) had central, split-system air conditioning (Pigg and Nevius, 2000).[1] Subsequent statewide telephone surveys in 2001 and 2003 showed that about 4 percent of owners of existing Wisconsin homes purchased new central systems each year, and that fully two-thirds of these systems were installations in homes that lacked air conditioning (or that had previous used room units for space cooling), suggesting that the saturation of central air conditioning in single-family homes was increasing at 2.5 to 3.0 percentage points annually.[2] Indeed, a recent Energy Center of Wisconsin statewide telephone survey indicates about 72 (±5) percent saturation in Wisconsin.[3]

FIGURE 1, THE WISCONSIN CENTRAL AIR CONDITIONING MARKET.

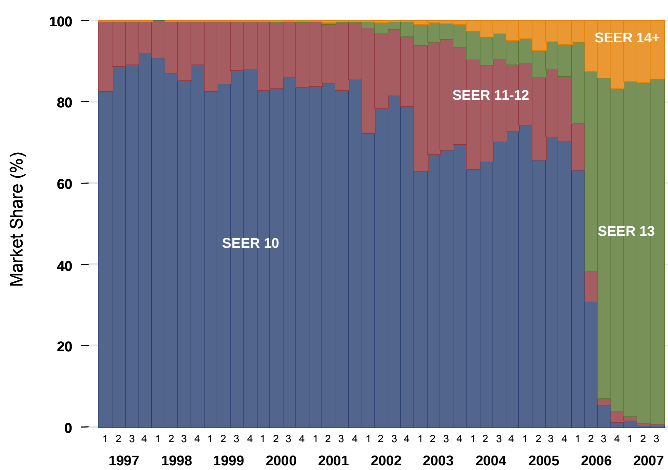


Source: ECW Appliance Sales Tracking Survey

Based on the above sources, it is estimated that 70,000 to 80,000 central air conditioners are installed in Wisconsin single-family homes each year, of which only about a quarter are replacements to existing systems (Figure 1).

Prior to implementation of the federal SEER-13 standard in 2006, the Wisconsin market was dominated by SEER 10 equipment (Figure 2).[4] SEER 13 has since become the new dominant category, but the market share for SEER 14+ equipment also increased concurrently with implementation of the new standard.

FIGURE 2, WISCONSIN QUARTERLY CENTRAL AC MARKET SHARE BY SEER LEVEL.



Source: FACTS project

[1] A similar study in 2004 involving on-site audits to a random sample of single- and multifamily rental propertied in Wisconsin showed that about 12 percent of Wisconsin's 658,000 rental housing units had central, split-system air conditioners.
[2] These Energy Center of Wisconsin Appliance Sales Tracking Surveys involved about 2,200 single-family, owner-occupied households.
[3] Midwest Energy Attitude Survey, November 2007. Results are for 223 Wisconsin homeowners.
[4] These data come from distributor-provided sales data under the Focus On Energy funded Furnace and AC Sales Tracking System (FACTS), which is estimated to track 50 to 60 percent of the total market.

The market share for SEER 14+ equipment is likely underestimated from these data, because they are based on distributor-reported condenser sales only, for which SEER ratings are based on the highest sales-volume combination of outdoor and indoor coils. Some installers over-size evaporator coils to achieve a higher SEER rating, and, more notably, the concurrent installation of a furnace with an electrically-efficient air handler alone can boost SEER from 13 to 14. About a quarter of Wisconsin furnace sales are electrically-efficient models. Five of 13 (38%) new systems that were recruited via random-digit-dial methods for the 2007 Focus field study were found to be SEER 14+ systems. Additionally, 24 of 40 (60%) of the new systems in the study had thermostatic expansion valves (TXVs), which are thought to improve the ability of the system to maintain efficiency levels in the face of refrigerant charge errors.

SIZING

The vast majority of Wisconsin central air conditioners fall in the range of 2 to 3 tons of cooling capacity, as Table 1 shows.

TABLE 1, DISTRIBUTION OF SYSTEM CAPACITY FOR THREE STUDIES.

System nominal output capacity (tons)	Residential Characterization Study (1999)	STAC (2005)	Focus (2007)	Combined Samples
1.5	6%	3%	9%	6%
2	41%	57%	33%	41%
2.5	24%	22%	33%	26%
3	23%	19%	20%	22%
3.5	4%	0%	1%	3%
4	2%	0%	3%	2%
5	1%	0%	1%	1%
Study n	157	37	76	270

Square footage data gathered from the 1999 statewide Residential Characterization Study shows that the median Wisconsin home with central air conditioning has about 700 square feet of finished living space (excluding basement areas) per installed ton of cooling capacity, and 90 percent of homes had between 500 and 1,000 square feet per ton. (The 31 new homes in the sample had a somewhat higher median of 760 square feet per ton and the 10 low-income homes had a median of about 600 square feet per ton.)

MANUAL J ESTIMATES OF SIZING

The Air Conditioning Contractors of America (ACCA) Manual J is the generally-recognized standard for sizing residential central air conditioners. Manual J calculations were performed for the 37 SEER 13+ STAC sites tested and monitored in 2005.[5] These calculations take into account wall and ceiling areas and insulation levels, window areas and orientation, as well as air leakage rates (blower door tests were conducted to quantify the latter). The systems were all installed in the Madison, Wisconsin metropolitan area, and design cooling loads were based on the Madison design values of 87°F dry-bulb and 72°F wet-bulb temperatures.

FIGURE 3, AGGREGATE CONTRIBUTION TO MANUAL J SENSIBLE COOLING LOAD, 37 SITES.

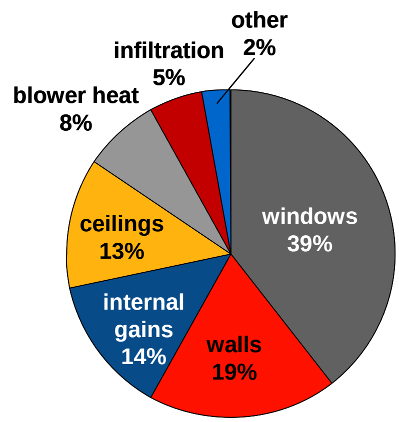


Figure 3 shows the aggregate contribution of various contributors to cooling load to the Manual J estimated design sensible cooling load

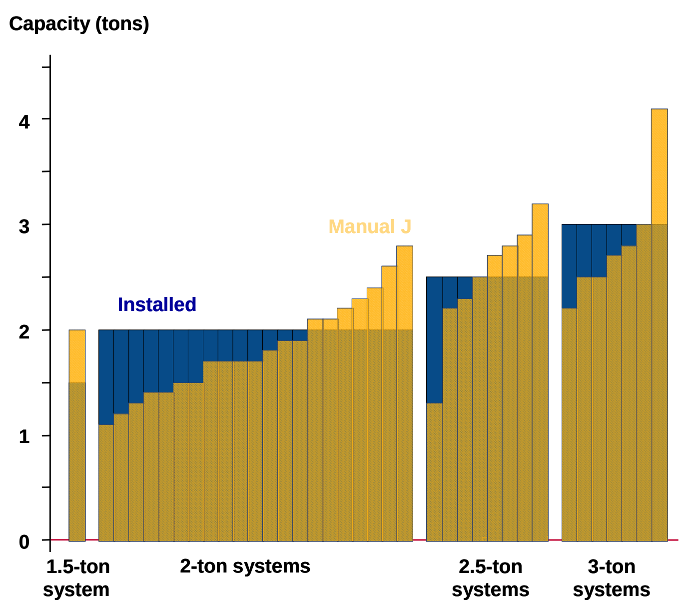
[5] The calculations were conducted using Right-Suit Residential, Version 6.0.27, using the 8th Edition version of Manual J.

across the 37 sites. Windows, walls, ceilings and internal heat gains make up the majority of the total cooling load.

Figure 4 shows how the design cooling loads calculated with Manual J compare to the actual installed system capacity. On balance, more systems were oversized relative to Manual J cooling loads than undersized, with the average system being 12 percent oversized.

However, if one considers that air conditioners are generally only available in ½ ton increments of 1.5 tons of capacity and higher—and that one would install the next highest ½ ton size above the Manual-J estimated design load—a slightly different picture emerges, as shown in Table 2. This assessment suggests that about a third of systems are appropriately sized, and half are about evenly divided between being under- or over-sized by ½ ton. Fewer than 15 percent are mis-sized by one ton or more.[6]

FIGURE 4, MANUAL-J VS. INSTALLED CAPACITY FOR 37 NEW SYSTEMS.



[6] If, instead of rounding the Manual J loads *up* to the nearest ½ ton increment, one rounds to the *nearest* increment, the proportion of systems that are appropriately sized remains relatively unchanged at 30%, but about 40% of systems would be characterized as being over-sized by ½ ton, versus 20% of systems being under-sized by ½ ton, with the remaining 10% being mis-sized by 1 ton or more.

TABLE 2, SIZING ASSESSMENT OF 37 NEW SEER 13+ SYSTEMS.

Sizing Error Installed vs. Manual J[a]		**n**	**%**
Undersized by…	...1 ½ tons	1	3%
	…1 ton	3	8%
	…½ ton	9	24%
Appropriately sized		13	35%
Oversized by…	... ½ ton	10	27%
	…1 ton	1	3%
Total		37	100%

[a]Compares installed nominal capacity to Manual J (8th Edition) calculated cooling load rounded up to nearest ½-ton increment ≥ 1.5 tons

EMPIRICAL ASSESSMENT OF SIZING

The cycling data collected for the STAC and Focus monitoring sites allows for an empirical assessment of sizing, though not for all sites, because run-time for some systems is dominated by long runs of continuous operation from householders keeping the system turned off (or the thermostat set up) during part of the day. The empirical assessment of system sizing proceeded as follows:

1. Remove periods when the system ran for more than an hour continuously after not operating at all for four hours or more. The purpose of this step is to at least partly remove long runs that are due to occupant interaction at the thermostat rather than load on the system.

2. For each site, identify the three-hour window that represents the highest average recorded duty cycle. (The center of this window ranged from 1 pm to 7 pm, but was centered between 4 pm and 6 pm for 80 percent of the sites.)

3. For the three-hour peak load period, regress daily duty cycle against outdoor temperature, and use the resulting regression fit to project duty cycle at 87°F, representing a typical Wisconsin design condition.[7]

From this analysis, a system with a projected peak duty cycle at design conditions of 50 percent would be categorized as oversized by 100 percent, and one with a projected duty cycle of 100 percent would be considered to be appropriately sized.[8]

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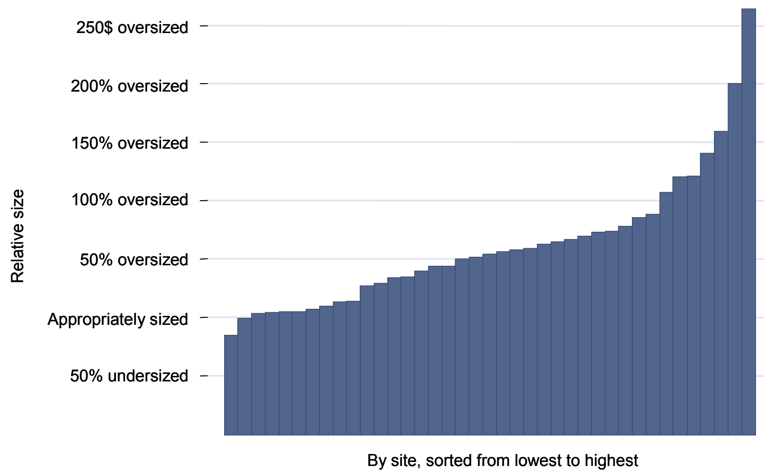
[7] ASHRAE 1% design conditions for major Wisconsin metropolitan areas range from 85°F (Green Bay) to 88°F (La Crosse).

[8] A potential problem with this approach is that because actual duty cycle is limited to 100 percent, the regression fits could understate the degree of undersizing for systems that are seriously undersized. However, visual assessment of the data indicates that this is not an issue for all but perhaps two sites.

Figure 5, Empirical estimates of relative sizing, STAC and Focus monitoring sites.



As Figure 5 shows, only one of the 39 sites amenable to this analysis appears to be undersized. Overall, about a quarter are approximately appropriately sized, and another quarter are more than 100 percent oversized; the remaining 50 percent are somewhere between about 25 and 100 percent oversized.

These empirical sizing estimates can be combined with the installed nominal tonnage of the systems to determine the empirical estimate of appropriate tons, rounded to the nearest ½ ton, and assuming a minimum available size of 1.5 tons (Table 3). The results indicate that about a third of systems are oversized by ½ ton, and another 40 percent are over-sized by a full ton or more. Keep in mind that these estimates are based on simply meeting peak three-hour loads at design conditions, and do not account for recovery from thermostat set-up.

TABLE 3, INSTALLED AND EMPIRICALLY-DETERMINED SIZING (STAC AND FOCUS SITES).

Empirically-determined size (tons)	Installed size (tons) 1.5	2	2.5	3	Total systems
1.5	4	12	10	3	29
2	0	4	0	3	7
2.5	0	1	2	0	3
Total systems	4	17	12	6	39
Oversized					
Appropriately sized					
Undersized					

SIZING SWAP EXPERIMENT

The research agenda for the STAC project included a direct test of the energy and comfort impacts of system sizing on a limited number of homes. The research plan called for monitoring new over-sized systems for a period of time, then swapping these out for properly-sized air conditioners, and comparing the two periods.

The original project schedule called for monitoring four homes over a single cooling season with the system swap-outs occurring midway through the cooling season. However, difficulty recruiting suitable sites for this test as well as contracting delays led to modifying the task to monitoring only two homes over two cooling seasons (2005 and 2006).

The test sites consisted of a 1970s home in North Prairie, Wisconsin (about 25 miles southwest of Milwaukee), and a newly built home south of Cross Plains, Wisconsin (about 10 miles west of Madison). In both cases, heating contractors had recommended installation of a 3-ton system, but sizing calculations suggested that a 2-ton unit (or smaller) would be adequate. The tests therefore comprised comparison of the performance of 2-ton systems compared to 3-ton systems. In both cases, new 2- and 3-ton systems of the same make and model were installed, differing only in the nominal capacity. The existing furnaces were not changed in either case.

North Prairie

The North Prairie home was built in the 1970s, and is owned by an independent home performance consultant who had implemented insulation and air sealing upgrades to improve the shell efficiency of the home. The home is located in a fairly open subdivision with large lots and little shading in the summer.

FIGURE 6, NORTH PRAIRIE TEST HOME.

The home had an existing 3-ton air conditioner. When the homeowner solicited bids for a new unit, the local heating contractor that was selected for the job recommended a new unit of the same capacity, though sizing calculations taking into account the shell upgrades suggested that a 1.5-ton system would be adequate.

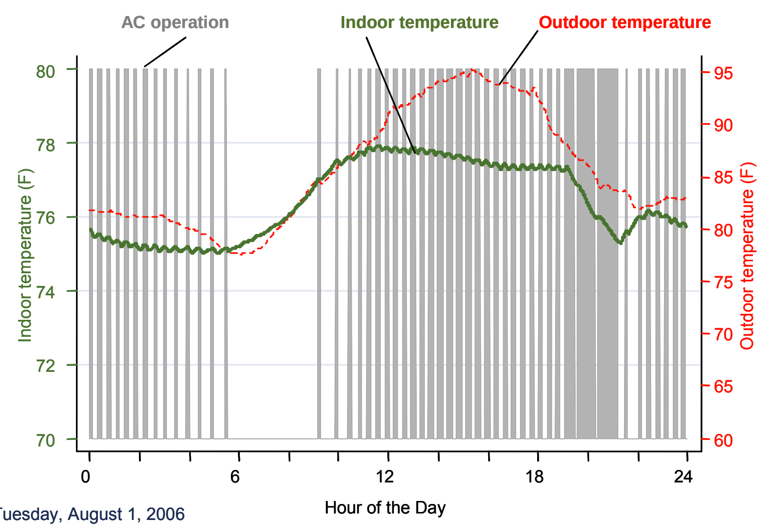
For the sizing experiment, the 3-ton unit recommended by the contractor was installed in mid-July 2005 and monitored for the remainder of the cooling season. The 2-ton system was installed in late September 2005, and was monitored over the 2006 cooling season.

Both systems were of the same make and general model, and differed only in nominal output capacity. The model installed was a SEER-13, non-TXV design using R-410a refrigerant. New matching evaporator coils were installed in both cases. The existing 80,000 Btu/hr, non-variable speed furnace remained in place throughout the monitoring.

Two adjustments were made to the 2-ton system. First, the installation contractor did not change the airflow setting for the furnace when the 2-ton system was installed, a fact that was overlooked until late June 2006. Measured wet-coil airflows showed a range of airflow from about 800 cfm to 1,050 across the four available speed settings on the furnace. The 3-ton system operated using the high-speed setting, which translates into 350 cfm per nominal ton. In late June 2006, the speed setting for the 2-ton system was switched from high to low, or from 525 cfm per nominal ton to about 400 cfm per ton. Most of the analysis results reported here for the two-ton system are based on data after the airflow setting was reduced.

Second, although a refrigerant charge test conducted by the heating contractor at the time of installation showed that the 2-ton system was properly charged, subsequent tests of output capacity showed the system to only be producing about 18,000 to 20,000 Btu/hr of cooling. This led to re-testing refrigerant charge in late July 2006, which showed that the system was very slightly undercharged. A small amount of refrigerant was added at this point, which increased the measured output of the unit about 1,000 Btu/hr. The heating contractor also injected dye into the system at this time: examination about one month later showed no signs of refrigerant leakage, and a capacity test at the end of the cooling season showed the unit to be providing about 21,000 Btu/hr of cooling capacity.

FIGURE 7, NORTH PRAIRIE TYPICAL HOT WEEKDAY OPERATION.



Examination of the monitoring data showed that during both summers the homeowner generally practiced a daytime thermostat setup of about 3F° from early morning until about 7 pm on weekdays (Figure 7): during this period, the system would only operate if the temperature exceeded about 77°F.

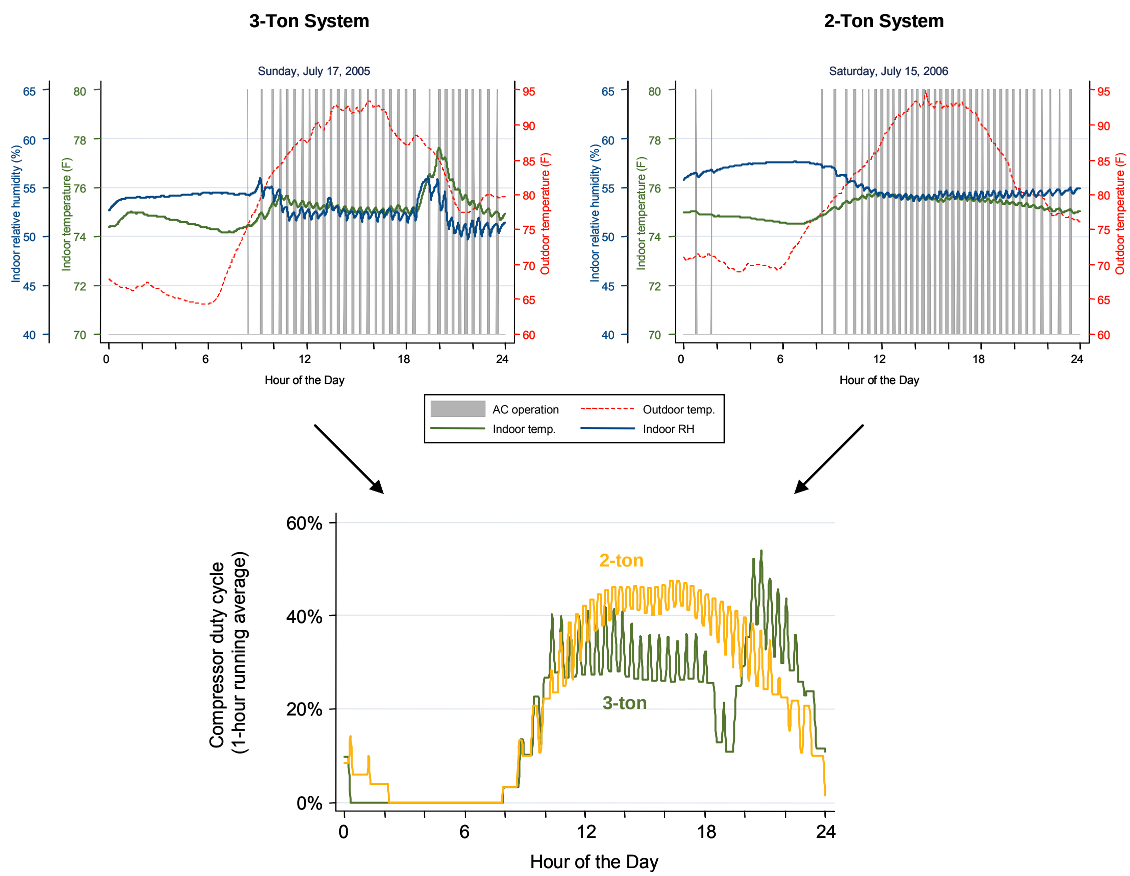
At other times (including the daytime hours on weekends), the thermostat set point was kept at about 75°F when the system was operated. As is not uncommon in Wisconsin, there were also some warm days when the system was not operated at all, as well as days when the system was turned off until late in the day. The results below are based on days when there was at least some AC operation.

The data collected from the site also indicated that the homeowner practiced some sporadic continuous-fan operation during the summer of 2005—though not during the summer of 2006. Specifically, there were 10 days in 2005 when the furnace fan was operated even when there was no call for cooling: these generally began in the afternoon or evening, and continued until the next morning. These periods are generally excluded from the analyses that follow.

Power data from the site show that the 2-ton compressor system drew 35 to 38 percent less power than the 3-ton system, and the air handler drew 29 percent less power when operated at low speed than when operated at high-speed. For the system as a whole, the 2-ton system operated at 65 percent of the power of the 3-ton system.

Due to its lower cooling output, the 2-ton system ran more hours under equivalent weather conditions (Figure 8). Note also from Figure 8 that despite outdoor temperature well above the design value of 86°F, the 2-ton system was able to recover from the afternoon temperature setup in a reasonable period of time.

FIGURE 8, NORTH PRAIRIE 2-TON AND 3-TON SYSTEM OPERATION ON SIMILAR DAYS.



When daily operating time is regressed against outdoor temperature for each of the systems—and then expanded to an estimate of seasonal operating hours (using the 1987-2006 average cooling season temperature distribution for Milwaukee)—the results suggest 57 percent (± 19%) more operating hours for the 2-ton system compared to the 3-ton system. This is close to what one might expect from a system that produces 2/3 the cooling output of the 3-ton system.

In fact, similar regressions of daily kWh versus outdoor temperature (Figure 10) suggest a nearly exact offset between reduced electrical load and increased run-time, with close to zero measured difference between the two systems: 4 out of about 520 seasonal estimated kWh or about 1 percent lower energy consumption for the 2-ton system.

Note however, that the scatter in the daily data is large enough to create about 13 percentage points of uncertainty in the difference in estimated seasonal energy consumption between the two systems. This means that one can only confidently conclude for this site that the energy savings for the 2-ton system, if any, are unlikely to be more than about 12 percent.

FIGURE 9, NORTH PRAIRIE DAILY SYSTEM OPERATING MINUTES VERSUS DAILY TEMPERATURE, 2- AND 3-TON UNITS.

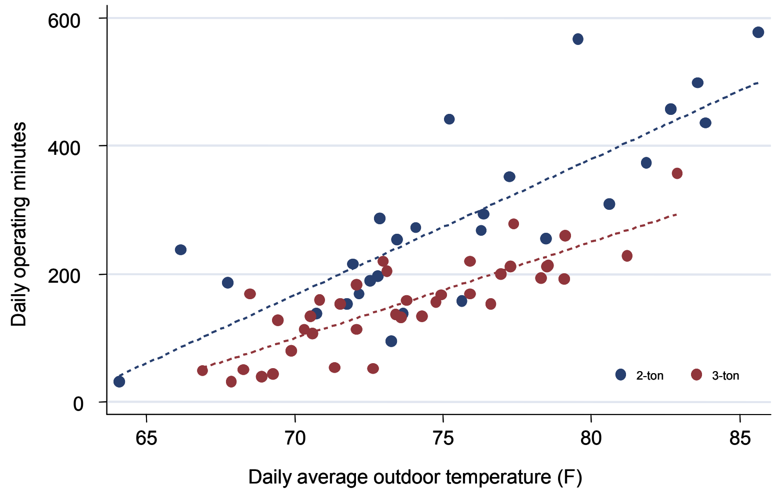


FIGURE 10, NORTH PRAIRIE DAILY KWH VERSUS DAILY TEMPERATURE, 2- AND 3-TON UNITS.

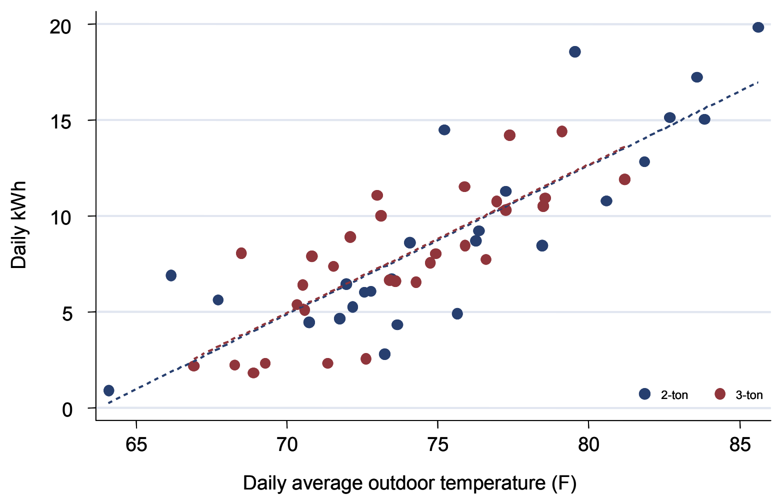
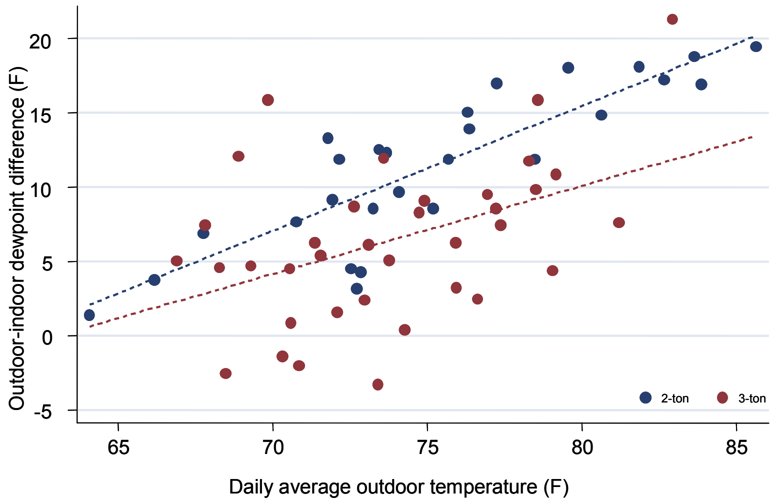


FIGURE 11, NORTH PRAIRIE OUTDOOR/INDOOR DAILY DEWPOINT DIFFERENCE FOR 2- AND 3-TON SYSTEMS (DAYS WITH 30+ MINUTES OF SYSTEM OPERATION).



Another hypothesis about appropriately-sized air conditioners is that they will do a better job of controlling indoor humidity due to longer operating times if those longer operating times increase the proportion of the time that condensate flows off the evaporator coil. Of course, the trade-off is that smaller systems remove humidity at a lower rate when they are operating.

For the North Prairie site, indoor relative humidity (on days when the system was operated) was no different between the two systems, averaging 49.7 (±0.7) percent for the 2-ton system and 48.9 (±1.2) percent for the 3-ton system at indoor temperatures of about 77°F.[9]

However, outdoor temperatures and humidity—the latter based on data from the nearby Waukesha airport— were both somewhat higher over the monitoring period for the 2-ton system. To control for these higher values we looked at the difference between outdoor and indoor dew point as a function of outdoor temperature. As Figure 11 shows, for both systems, the dew point depression is greater in hotter weather, when the system runs more hours and (probably more importantly) outdoor dew points tend to be higher. The difference between the two systems is statistically significant (at a 90% confidence level) for days that average about 70°F or higher.

Viewed in this way, one could say that the 2-ton system did a better job of controlling indoor humidity in the face of higher outdoor humidity.

[9] These humidities are based on measurements made on the second floor (which averaged about 2°F warmer than the first floor). Humidity at the 1st floor thermostat did show a statistically significant difference of about 3 percentage points, but that is thought to be due to the fact that different data loggers were used during the two monitoring periods at that location. Note also that—like many Wisconsin households—the homeowner operated a basement dehumidifier on a sporadic basis.

Finally, temperature sensors mounted at 1-, 4- and 7-feet above floor level on both the first and second floors showed slightly increased vertical stratification on the second floor for the 2-ton system (Table 4), though stratification was not large at this site to begin with, and the observed difference is small (though statistically significant).

Table 4, North Prairie vertical temperature stratification, by floor, for 2- and 3-ton systems.

F°	First Floor		Second Floor	
	4' vs. 1'	7' vs. 1'	4' vs. 1'	7' vs. 1'
2-ton	+1.18 (±0.05)	+1.73 (±0.09)	+0.73 (±0.06)	+1.35 (±0.11)
3-ton	+1.18 (±0.06)	+1.65 (±0.07)	+0.37 (±0.03)	+0.86 (±0.07)
Difference	0.00 (±0.08)	+0.08 (±0.10)	+0.36 (±0.07)	+0.48 (±0.13)

Note: uncertainties are 90% confidence intervals based on daily averages

CROSS PLAINS

FIGURE 12, CROSS PLAINS SITE.

The Cross Plains home is a newly-constructed (in 2005) home on a rural lot (Figure 12). Though the home is located on a wooded lot, the lot is exposed enough to allow some direct solar gain on the home from mid-morning through early evening during the summer.

The builder's heating contractor had already delivered a 3-ton, SEER-10 unit to the job site when the homeowners agreed to participate. For the purposes of the research project, a 3-ton, SEER-14 unit by the same manufacturer was substituted and installed in June 2005. In mid-August the 3-ton unit was replaced with a 2-ton version of the same system, including replacement of the evaporator coil. The second system was monitored through the end of the 2006 cooling season. Both units had evaporator coils with TXVs, and both systems used R-410a refrigerant.

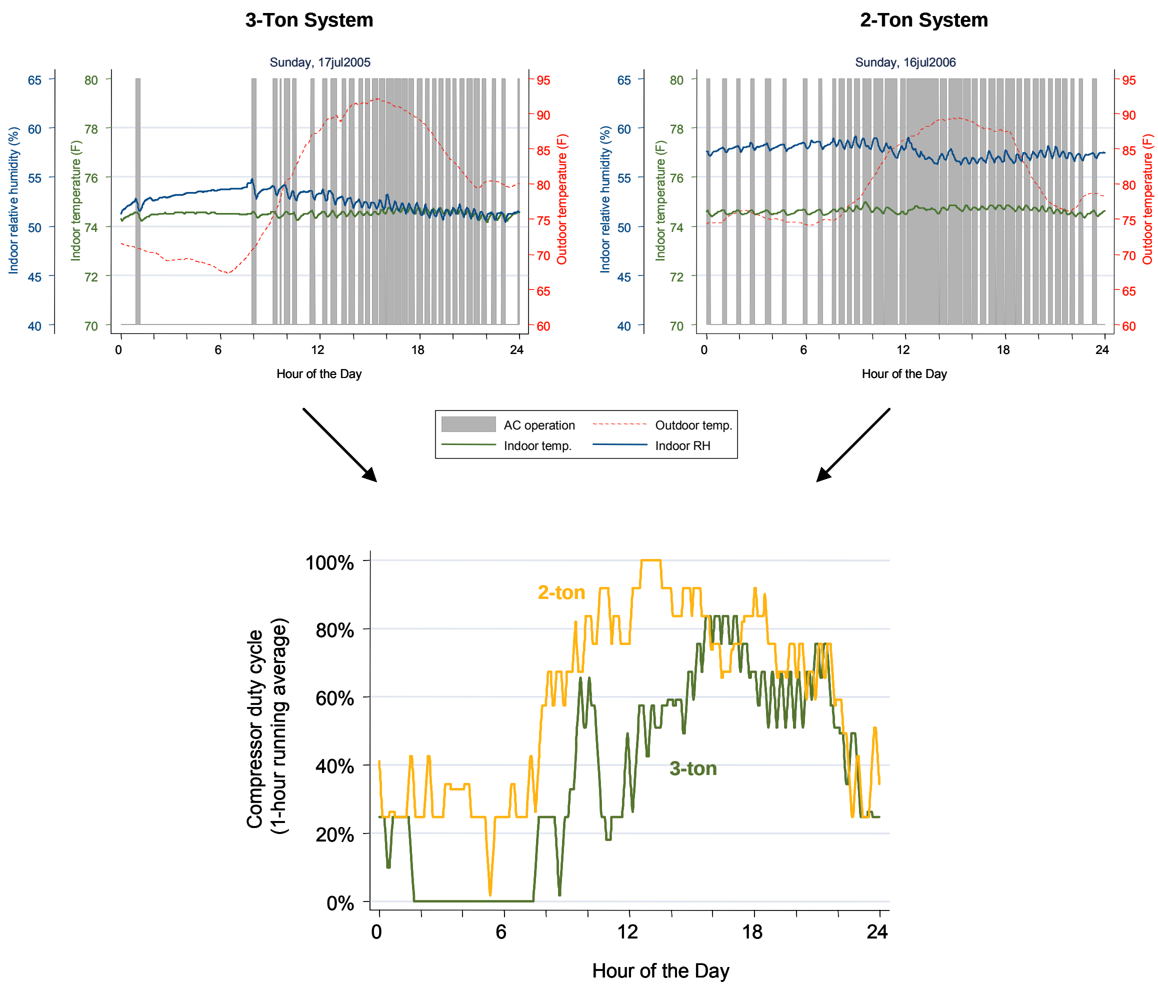
A 100,000 Btu/hr, non-ECM furnace remained in place throughout the study. The four available fan speeds on the furnace produced only a narrow range of airflow—from 1,130 cfm at low speed to 1,200 cfm at high speed. This meant that while airflow for the 3-ton system could be provided at about 400-425 cfm per nominal ton, airflow for the 2-ton system could not be brought below about 570 cfm per nominal ton.

[Note: furnace cycling data collected over the 2005/2006 heating season indicate that the furnace is significantly oversized relative to design loads: the data indicate about 9.3 hours of operation (39% duty cycle) on days that average -10^{o}F. However, many Wisconsin furnaces are similarly oversized. Also, the occupants practiced an overnight thermostat setback (of about 5^{o}F) for the first part of the heating season: the cycling data suggest that the morning setback recovery takes about 2 hours on cold days.]

The monitoring data showed that when operating the air conditioning, the occupants typically maintained an indoor temperature of 74 to 75^{o}F, with a weekday daytime thermostat set-up of 3 to 5^{o}F until late afternoon. For the purposes of the research project, the homeowners agreed to maintain a constant setting of 75^{o}F during part of each cooling season. The analysis that follows treats these periods separately.

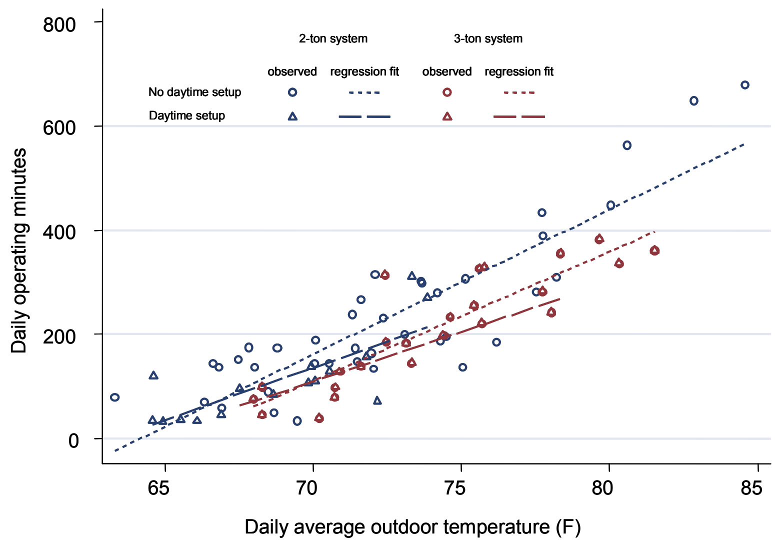
Operation of the systems was generally comparable on similar days, with higher duty cycle evident for the 2-ton system in the afternoon (Figure 13).

FIGURE 13, CROSS PLAINS SITE 2- AND 3-TON SYSTEM OPERATION ON SIMILAR DAYS (NO THERMOSTAT SET-UP).



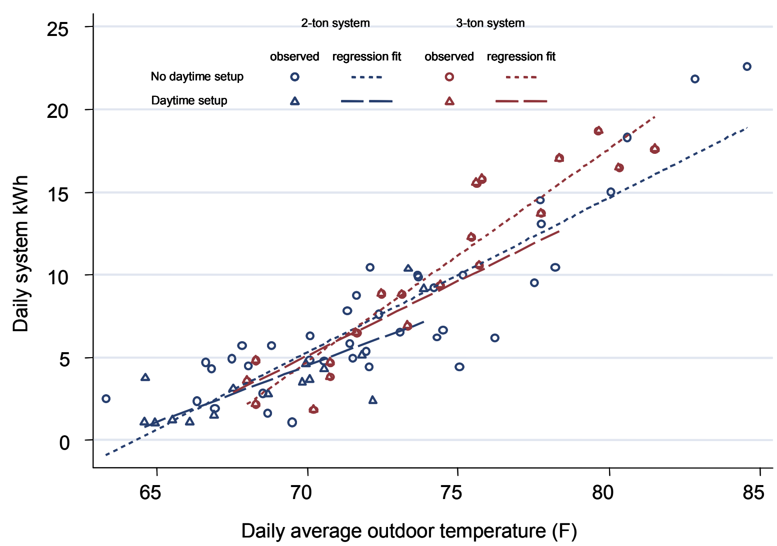
Regression of daily operating minutes versus outdoor temperature indicates longer operating time for the 2-ton system (Figure 14). The data also suggest that for both systems, operating hours are lower when a daytime temperature set-up is practiced. When the regression fits shown in Figure 14 are expanded to estimates of seasonal operating hours (using 1987-2006 cooling season temperature data for Madison), the results indicate 19 (±27) percent longer operating time for the 2-ton system when daytime set-up is practiced and 33 (±17) percent longer operating time without set-up. Note that only the latter is statistically significant.

FIGURE 14, CROSS PLAINS SITE DAILY OPERATING MINUTES VERSUS DAILY OUTDOOR TEMPERATURE, 2- AND 3-TON SYSTEMS, WITH AND WITHOUT AFTERNOON TEMPERATURE SET-UP (DAYS WITH 30+ MINUTES OF SYSTEM OPERATION).



The compressor for the 2-ton system drew about 68 percent of the power of the 3-ton system, and air handler power for the 2-ton system was about 75 percent that of the 3-ton system. Taken together, the 2-ton system drew about 70 percent of the power drawn by the 3-ton system.

FIGURE 15, CROSS PLAINS SITE DAILY KWH VERSUS DAILY AVERAGE TEMPERATURE, 2- AND 3-TON SYSTEMS, WITH AND WITHOUT TEMPERATURE SET-UP (DAYS WITH 30+ MINUTES OF SYSTEM OPERATION.

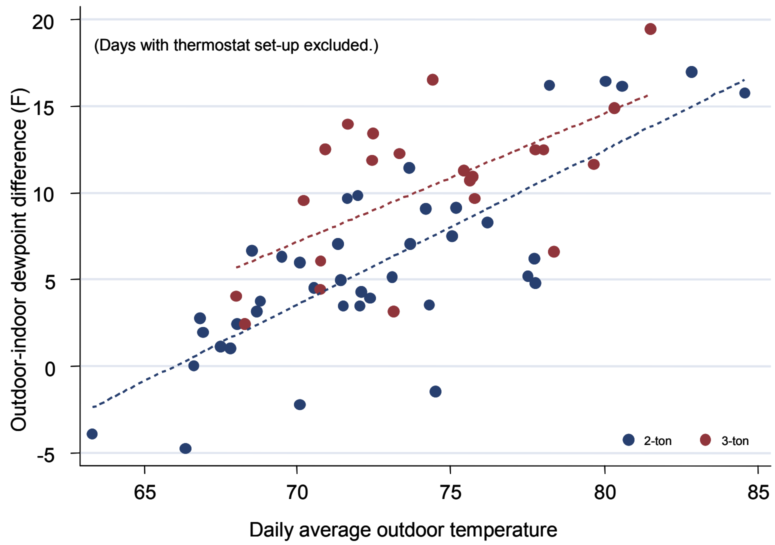


When daily kWh consumption is regressed against outdoor temperature for the two systems and the two thermostat setting modes (Figure 15), and then expanded to estimates of seasonal energy consumption, the results indicate a 16 (±20) percent reduction in seasonal energy consumption when thermostat set-up is practiced, and a 5 (± 11) percent reduction when the thermostat is maintained at a constant temperature. The size of the confidence intervals on these figures—which span both negative and positive savings—precludes definitive conclusions about whether the 2-ton system saves energy.

Relative humidity (at the thermostat) averaged 4.8 (±0.7) percentage points higher under the 2-ton system (59.0 ±0.5 percent) compared to the 3-ton system 54.2 ±0.4 percent) on days when the systems were operated at least 30 minutes.[10] (Relative humidity also averaged 2 to 4 percentage points higher for days when a daytime set-up was practiced.)

Plots of the difference between outdoor and indoor dew point versus outdoor temperature also indicate that the 2-ton system did not do as good a job of removing humidity as the 3-ton system (Figure 16). The most likely explanation for this is the high airflow per ton of output capacity for the 2-ton system due to the large furnace at the house.

FIGURE 16, CROSS PLAINS SITE DAILY OUTDOOR-INDOOR DEWPOINT DIFFERENCE FOR 2- AND 3-TON SYSTEMS (DAYS WITH 30+ MINUTES OF SYSTEM OPERATION AND NO DAYTIME THERMOSTAT SET-UP.)



Finally, measurements of vertical temperature stratification on the first and second floors showed no statistically (or practically) significant differences in the amount of stratification between the two systems (Table 5). This is not surprising given the small difference in airflow between the two systems, and the fact that both high and low return registers are located throughout the house.

Table 5, Cross Plains vertical temperature stratification, by floor, for 2- and 3-ton systems.

F°	First Floor		Second Floor	
	4' vs. 1'	7' vs. 1'	4' vs. 1'	7' vs. 1'
2-ton	+0.92 (±0.03)	+1.32 (±0.03)	+0.81 (±0.09)	+1.20 (±0.15)
3-ton	+0.92 (±0.03)	+1.36 (±0.05)	+0.78 (±0.10)	+1.29 (±0.16)
Difference	0.00 (±0.04)	-0.04 (±0.06)	+0.03 (±0.16)	+0.08 (±0.23)

Note: uncertainties are 90% confidence intervals based on daily averages

[10] Note also that the homeowners operated a basement dehumidifier connected to a floor drain throughout both cooling seasons.

Similarly ambiguous results were found in three Florida houses where a comparable protocol was implemented by the Florida Solar Energy Center as part of the overall STAC project (Sonne et al., 2006). While there are important housing stock differences between these two efforts—most notably that ductwork in Florida homes is generally in unconditioned attic spaces, versus basement and conditioned spaces for Wisconsin homes—neither effort yielded clear evidence of significant energy savings or improvement in humidity control from correcting over-sizing issues with central air conditioners.

The Florida study notes that manufacturers have reduced cycling losses in efforts to improve SEER ratings, thereby reducing the potential for savings from right-sizing equipment. As with the case of the Cross Plains home here, the Florida study also noted issues with the ability to adjust airflow to appropriate levels if the air handler is mismatched to the air conditioning system.

BEHAVIORAL ASPECTS OF CENTRAL AC USE

Wisconsin's cooling season extends from mid-May to mid-September, and can be characterized by intermittent periods of hot and humid weather interspersed with milder conditions. Some households choose to gut it out rather than run the air conditioner during some hot spells. Summer is also prime vacation time, and air conditioners are often turned off when people are away. Both of these factors contribute to less air conditioning use than would be the case if all households practiced set-it-and-forget-it behavior at the thermostat.

It is no surprise, though, that the probability that a household will operate their air conditioner on a given day is highly correlated with outdoor temperature, as Figure 17 shows for both the monitoring data from the 2005 STAC and 2007 Focus study groups and the 2003 AST survey data. The shape of the curves—which come from logistic regressions of whether the central AC is used on a given day versus outdoor temperature for individual sites—are different for the survey data and monitoring data, but there are qualitative differences between these two: the monitoring data reflects whether the air conditioning actually ran during a given day, while the survey data are based on whether the respondent reported having the thermostat set in cooling mode or turned off. The latter results in higher probabilities at cooler temperatures as the system may not operate at all on cool days even if the system is enabled in cooling mode.

FIGURE 17, LOGISTIC REGRESSION ESTIMATES OF PROBABILITY OF AC USE VERSUS OUTDOOR TEMPERATURE, MONITORING AND SURVEY RESULTS.

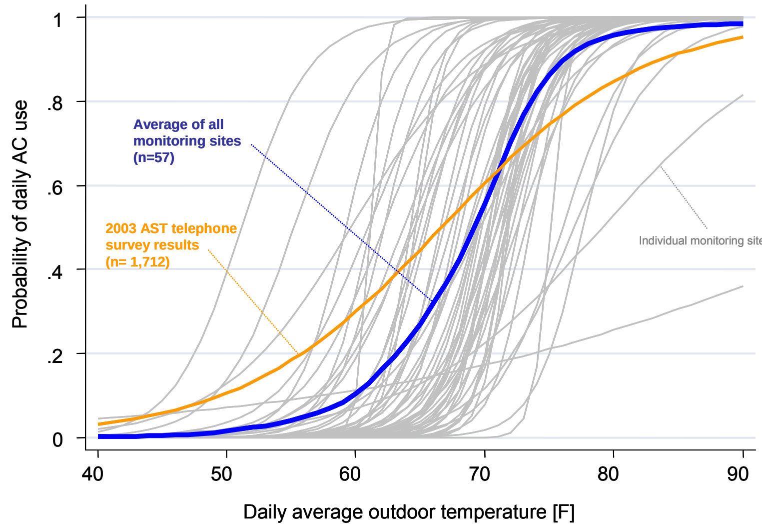
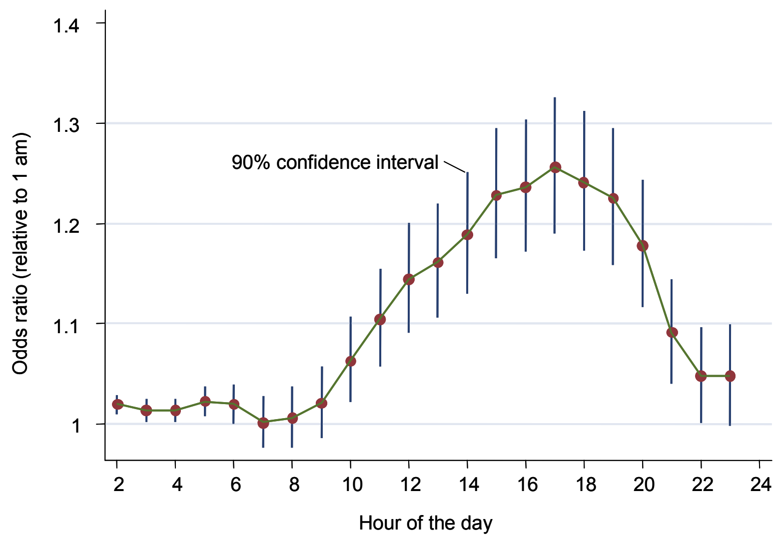


FIGURE 18, RELATIVE PROBABILITY OF AC USE BY HOUR OF THE DAY (2003 AST SURVEY).



The large sample size for the 2003 AST survey data allowed for the development of more complex regression models of air conditioning use incorporating additional potential explanatory variables such as the outdoor temperature on previous days, home age and type, household income, day of the week, time of day, and other factors (see Appendix B). These models suggest that in addition to the temperature on the day in question:

- high temperatures on prior days increase the likelihood of central AC use;
- homes with one or two occupants are significantly less likely to use the central AC
- occupants of 5+ unit apartment buildings are more likely to use their central AC (though number of such respondents in the survey was relatively small);
- households with annual income of $75,000 or more are somewhat more likely to use their central AC.

The AST survey queried respondents about whether the thermostat was set in cooling mode on an hourly basis over the course of the day preceding the call. This allowed for exploring a model that looked at the relative probability of AC use by time of day. As Figure 18 shows, compared to night-time hours, thermostats for central air conditioners are 20 to 30 percent more likely to be set in cooling mode during the afternoon and evening hours.

For households that did have their thermostat set in cooling mode, the 2003 AST survey asked about hourly temperature settings. As Figure 19 shows, most households with the cooling system enabled reported keeping the thermostat set at a temperature between 74^{o}F and 79^{o}F, with an average of about 75^{o}F (Table 6).

FIGURE 19, SELF-REPORTED THERMOSTAT SETTINGS BY HOUR OF DAY OUTDOOR TEMPERATURE CATEGORY (2003 AST SURVEY).

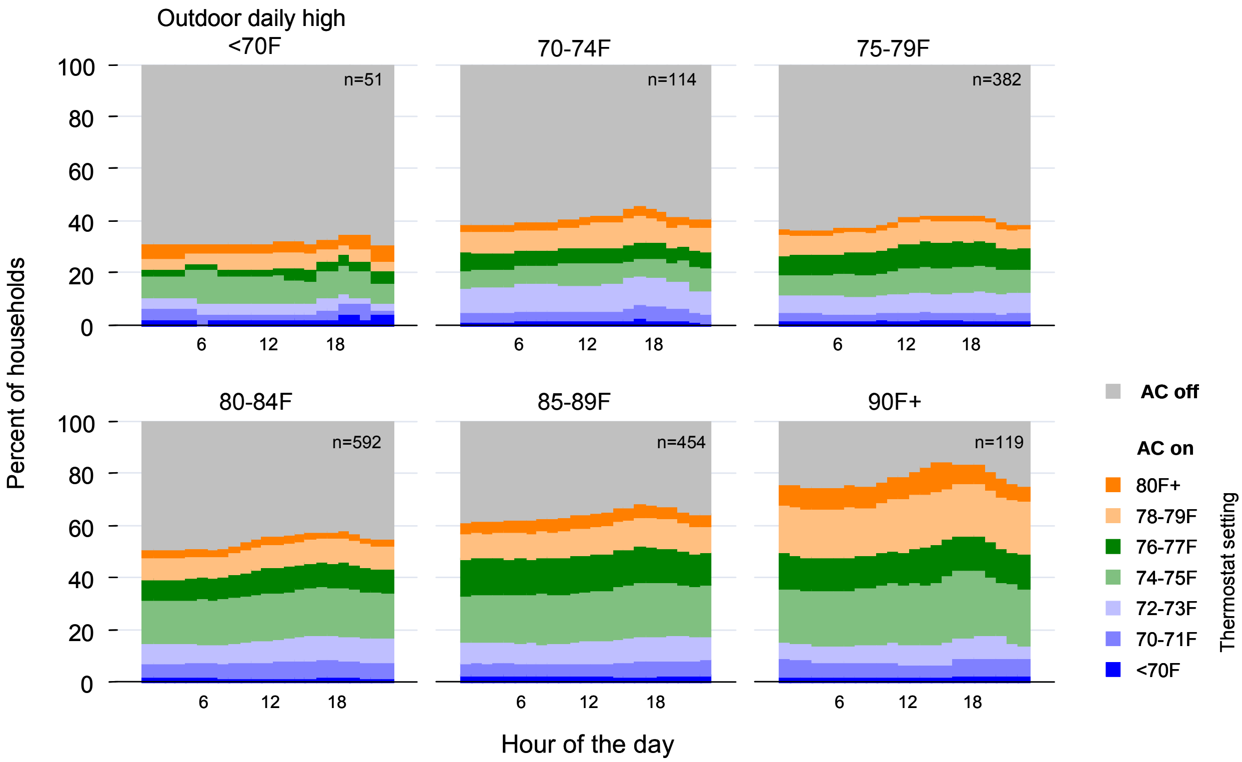
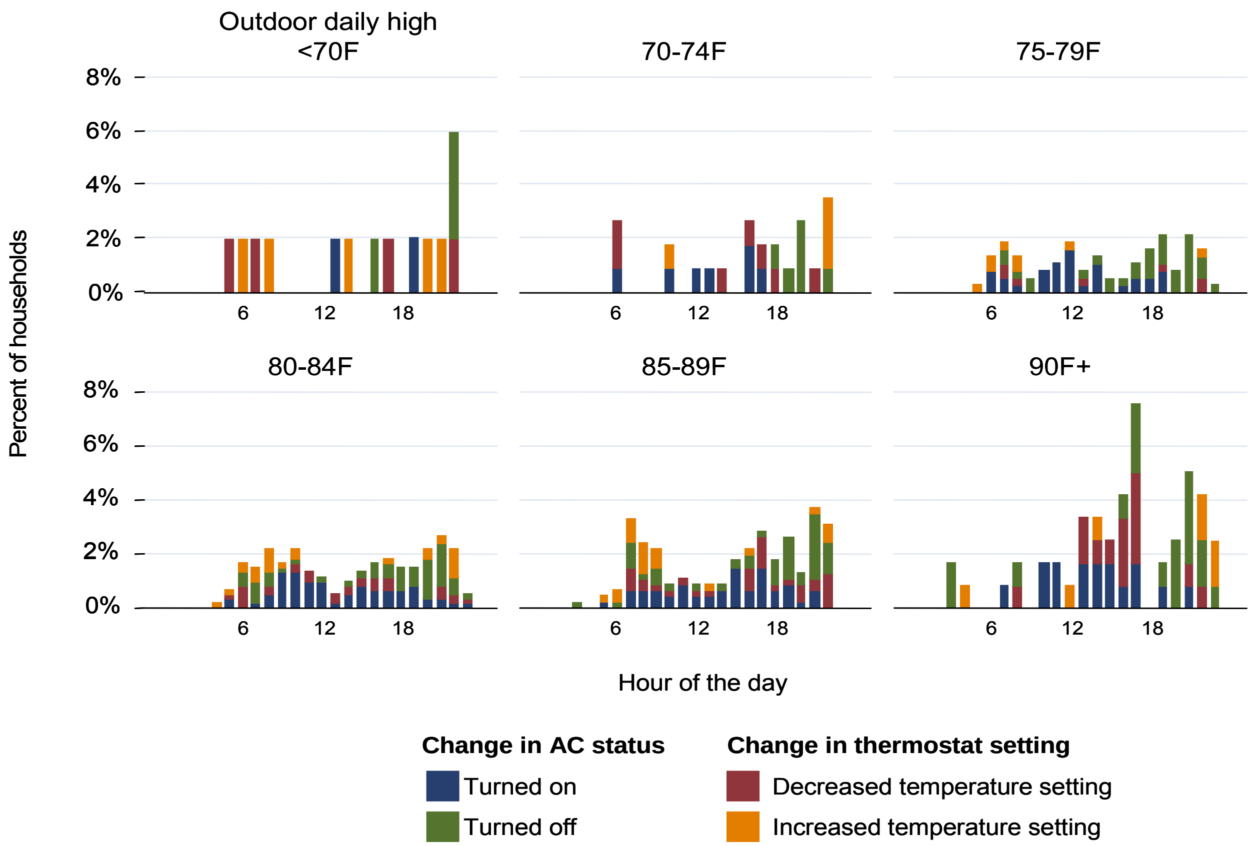


TABLE 6, MEAN SELF-REPORTED THERMOSTAT SETTING BY OUTDOOR TEMPERATURE CATEGORY (2003 AST SURVEY).

Outdoor high temperature for the day (F)	n	Mean reported thermostat setting (with 90% conf. interval)
<70F	51	75.0 ± 1.1
70-74F	114	74.4 ± 0.6
75-79F	382	74.7 ± 0.3
80-84F	592	74.7 ± 0.2
85-89F	454	75.1 ± 0.3
90F+	119	75.5 ± 0.5

In terms of reported changes at the thermostat, only a small minority of households reported switching the thermostat from or to cooling mode, or adjusting the thermostat temperature setting (Figure 20).

FIGURE 20, REPORTED CHANGES TO THERMOSTAT SETTINGS BY HOUR OF DAY AND OUTDOOR TEMPERATURE CATEGORY (2003 AST SURVEY).

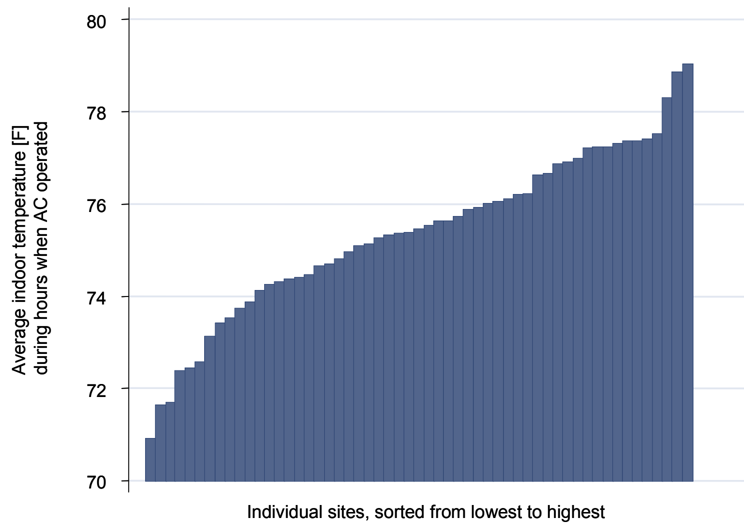


The temperature monitoring data at the thermostat for the 58 STAC and Focus sites provide empirical field data on thermostat settings and behavior. In aggregate, the monitoring data confirm an average set point of about 75°F, with about 90 percent of the values falling between 71.8°F and 78.5°F (see also Figure 21). Outdoor temperature is a statistically significant (but weak) predictor of indoor temperature at the hourly level in these data: each 1°F increase in outdoor temperature is associated on average with a 0.046 ± 0.018°F increase in indoor temperature.

Temperature loggers were also placed on the second floor of 12 two-story homes in the 2007 STAC monitoring sample. On average across all sites, temperatures on the second floor averaged 2.6°F higher than the temperature at the second floor thermostat during hours when the cooling system operated. Among the individual sites, the average difference between second-floor and first-floor temperature during hours with AC operation ranged from less than 1°F to more than 6°F. In hot weather (≥ 85°F), five of the 12 sites had second-floor temperatures that were 3°F or more higher than the first floor, and all but one showed a 2°F or more difference.

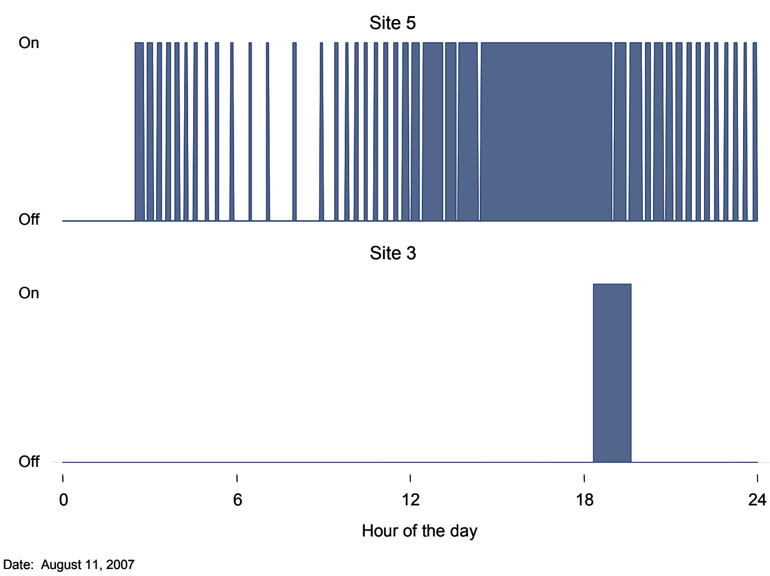
Also notable from the monitored cycling data for the 58 sites is the significant number of sudden and prolonged periods of cooling-system operation after a significant period without operation. Such events are likely the result of the occupants delaying operation of the system until later in the day after indoor temperatures have already climbed. Figure 22 shows the cycling behavior of two sites in the Madison area on the same day: the cooling system for the first site gradually builds to a prolonged period of operation in the afternoon, but is preceded by many shorter cycles; the second site shows only a single relatively long period of operation in the early evening. It is likely that the first system was cycling under thermostat control throughout the day, while the second system was switched off until the occupants returned home in the evening.

FIGURE 21, RECORDED AVERAGE INDOOR TEMPERATURE (AT THERMOSTAT) DURING HOURS WHEN AC WAS OPERATED, BY SITE (55 MONITORING SITES).



When these sudden-on events are defined as 60 minutes or more of continuous operation following four hours or more without system operation, such events make up 23 percent of the aggregate system run time across the 58 sites and about half of all operating cycles that last for 60 minutes or more (see also System cycling behavior, page 52).

FIGURE 22, CYCLING STATUS OF TWO SYSTEMS ON THE SAME DAY (2007 FOCUS MONITORING DATA).

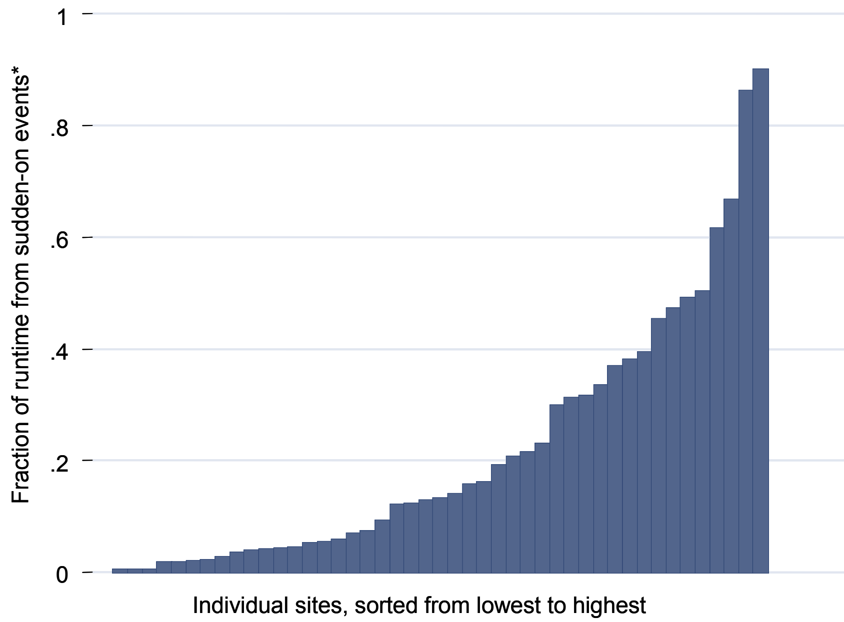


The degree to which households practice this type of behavior varies, as Figure 23 shows: such events are rare occurrences for some households, but account for nearly all system operation for others.

The operating hours for these sudden-on events tend to be concentrated in the afternoon and evening hours, as one might expect (Figure 24). Analysis of hot-day operation suggests that these events increase diversified air conditioning load by about 2.5 percent (see page 32).

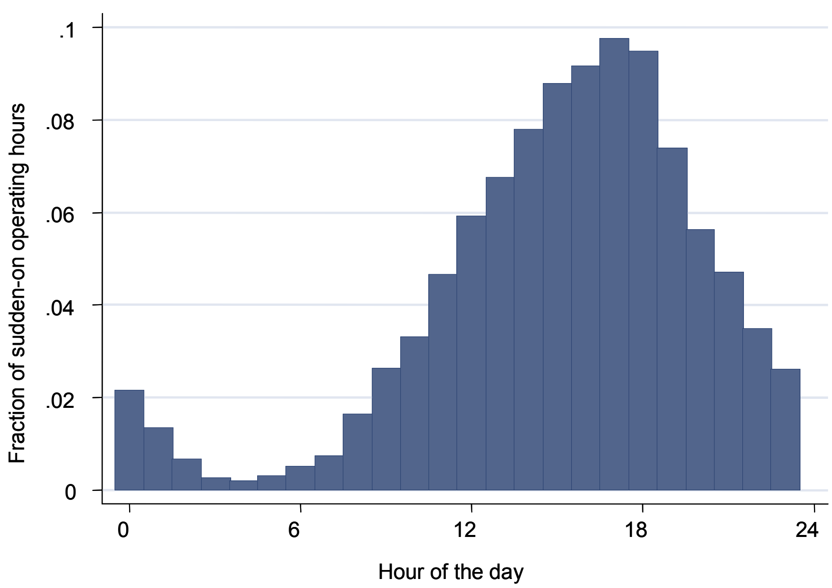
While the definition of sudden-on events used here probably contains some cycling events that are purely load-driven—and almost certainly excludes events that are related to occupant intervention at the thermostat—it seems clear that occupant interaction with the thermostat is an important aspect of air conditioning cycling behavior.

FIGURE 23, FRACTION OF TOTAL OPERATING TIME ATTRIBUTABLE TO SUDDEN-ON EVENTS, BY SITE (MONITORING SITES).



*Defined as 60+ minutes of continuous operation following 4+ hours of no operation
(Sites with < 50 hours of total operation excluded.)

FIGURE 24, HOURLY DISTRIBUTION OF SUDDEN-ON OPERATING HOURS (MONITORING SITES).

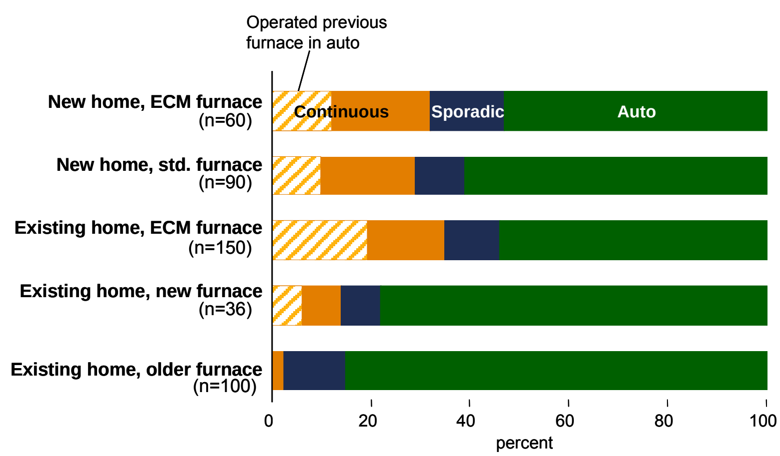


How people operate their air handlers is also a factor during the cooling season. Most thermostats allow for fan-auto and fan-on settings. In fan-auto mode, the air handler (which is typically a gas-fired furnace for Wisconsin households) only operates when the cooling system is operated. When set for fan-on, the air handler operates continuously regardless of whether the cooling system is operating. As Shirey and Henderson (2004) have shown, continuous fan operation in this manner may hurt the dehumidification capability of the cooling system, because moisture stored on the indoor cooling coil is quickly re-evaporated into the home.

In the 2003 AST survey, 12.3 ± 1.4 percent of respondents indicated that they ran their air handler continuously part or all of the previous day: the majority of these (9.4 ± 1.3 percent) reported running the air handler continuously the entire day, with only about 3 ± 0.7 percent of respondents reporting running the fan continuously for part of the day. There was no significant relationship between outdoor temperature and whether the fan was operated continuously. We also found little relationship between fan operation practices and other demographic and housing data gathered for the survey, with two exceptions: respondents who reported having 5 or more bedrooms had about twice the incidence rate of fan-on operation as other households, and none of the 36 respondents who lived in a mobile home reported fan-on operation on the day in question. Both of these differences are statistically significant, and suggest that home size plays a role in fan operation practices. Anecdotally, it seems that occupants of split-level homes are more likely to practice continuous-fan operation as they try to move cooler lower-level air around the upper levels.

The 2003/2004 survey of fan operation practices conducted by Glacier Consulting (Pigg and Talerico, 2004) provides additional detail on cooling-season fan operation practices in Wisconsin (Figure 25). That survey shows higher rates of continuous fan operation among new homes and homes with electrically-efficient electronically commutated motors (ECMs). Moreover, a significant proportion of households with new furnaces reported that they had operated their previous furnace in fan-auto mode.

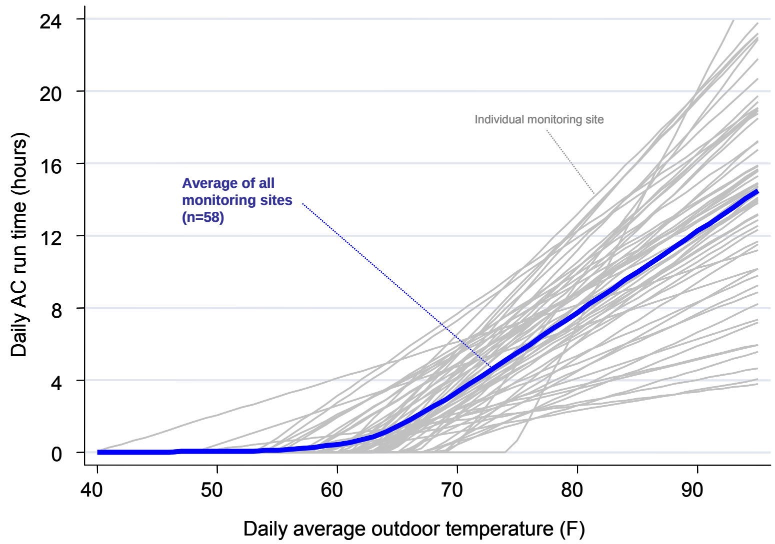
FIGURE 25, REPORTED COOLING-SEASON FAN OPERATION PRACTICES (2003/04 FAN OPERATION SURVEY).



SEASONAL OPERATING HOURS

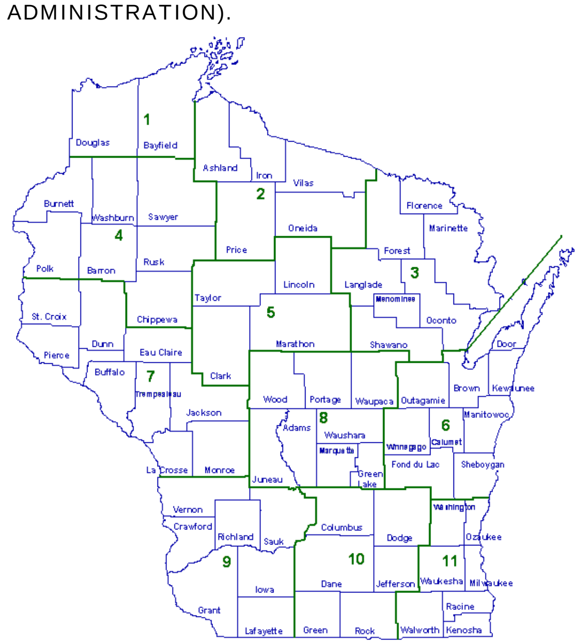
The 58 sites monitored for the 2003 STAC and 2007 Focus studies indicate that for most sites, the number of daily operating hours is reasonably linear with respect to daily outdoor temperature. Across the monitored sites, the data suggest that the average system will begin to show some daily run hours when the daily outdoor temperature exceeds about 65^{o}F, corresponding to a typical daily high of about 75^{o}F (Figure 26). This suggests that standard cooling degree days based on a 65^{o}F base temperature are an appropriate metric for cooling energy use estimates.

FIGURE 26, MODELED OPERATING HOURS VERSUS OUTDOOR TEMPERATURE (58 MONITORING SITES).



Seasonal operating hours can be estimated from these linear models by combining the estimated daily hours of operation at any given outdoor temperature with a seasonal bin distribution outdoor temperature. Logistic models of the probability of system use versus outdoor temperature can also be incorporated into the analysis to account for discretionary use of air conditioners. When applied to the individual site models (using 15-year average temperature distributions for the nearest weather station) the resulting estimates of seasonal operating hours range from less than 50 hours to more than 1,000 hours, with an average of about 325 hours.

FIGURE 27, WISCONSIN CLIMATE ZONES (SOURCE: WISCONSIN DEPARTAMENT OF ADMINISTRATION).



The usage models also allow each site to be modeled using weather data for other locations. When all sites are modeled in each of Wisconsin's 11 climate zones (Figure 27), the estimates shown in Table 7 are obtained, which suggest typical annual operating hours ranging from under 200 in the far north to more than 400 hours in the La Crosse area, with a weighted statewide average of 311 ± 44 hours.[11]

[11] A caveat regarding this analysis is that it presumes that sizing practices and occupant discretionary-use practices are comparable across regions.

TABLE 7, SEASONAL COOLING DEGREE DAYS AND ESTIMATED AIR CONDITIONER OPERATIONG HOURS, BY CLIMATE ZONE.

Climate zone	Weather station	Cooling degree-days[a]	Estimated average seasonal AC operating hours
1	Ashland	300	171 ± 31
2	Rhinelander	368	203 ± 35
3	Rhinelander/ Green Bay average	434	234 ± 38
4	Spooner	496	265 ± 40
5	Marshfield	486	259 ± 40
6	Green Bay	479	256 ± 40
7	Eau Claire	556	293 ± 42
8	Hancock	537	284 ± 42
9	La Crosse	840	430 ± 52
10	Madison	624	327 ± 45
11	Milwaukee	699	361 ± 46
State[b]		593	311 ± 43

[a]May through September, 1993-2007
[b]Based on housing-unit weighted average of climate zones using Census 2000 data. Alternative weighting schemes using Census 2000 population weights or Focus on Energy 2006 Efficient Heating and Cooling program participation by climate zone produces estimates that are within 1% of value shown above.

All of these estimates account for the fact that people do not always operate their cooling systems when outdoor conditions would otherwise warrant it. If this aspect of the calculations is omitted, estimated seasonal operating hours are roughly one-quarter to one-third higher, suggesting that discretionary use of air conditioning by Wisconsin households has an impact of this magnitude on seasonal run times.

These estimates of operating hours are less than those shown for Wisconsin on a widely-used map of cooling load hours published by the Air Conditioning and Refrigeration Institute (ARI), which shows 600 cooling load hours along Wisconsin's southern border and about 400 hours through the northern part of the state. The two are not incompatible, however: cooling load hours assume proper sizing of equipment, and do not account for discretionary use of equipment. If one considers that the empirical estimates of sizing developed here (see page 11) suggest 50 percent oversizing on average, and that discretionary use reduces operating hours by about 25 percent, then actual operating hours should be about half of total cooling load hours.

The models for seasonal operating hours can also be used to gauge the mid-load temperature (i.e., the outdoor temperature that divides seasonal operating hours in two halves). The federal test procedure for determining SEER uses 82 ^{o} F as a national estimate of mid-load temperature. The cooling models from the data here suggest that average mid-load temperatures for Wisconsin are somewhat below this value, ranging from 77 ± 0.7 ^{o}F in the north to 80 ±0.5 ^{o}F in the south, with a weighted statewide average of 79 ± 0.5 ^{o}F. This would suggest that—all other factors being equal—Wisconsin air conditioners should perform slightly more efficiently than the SEER ratings would suggest, because their operation is slightly more concentrated at cooler temperatures.

PEAK-DAY OPERATION

The operation of residential central air conditioners is of particular interest for utility load planning, because central air conditioning use is highly coincident with utility system peak.

Figure 28 shows the hourly distribution of system duty cycle (the percent of each hour that the system operated) for the 60 sites in the 2005 STAC and 2007 Focus monitoring samples on days that reached 90°F or more. During the peak operating period between about 3 pm and 7 pm, about 80 percent of systems were operating, and about 30 percent of all systems were running flat out (Figure 29). When these systems are included with the 51 percent of systems that are cycling, the average duty cycle is 55 percent (33 minutes/hour). The average duty cycle for systems that were operating during this time was 70 percent (42 minutes/hour).

FIGURE 28, SYSTEM CYCLING BY HOUR OF DAY ON DAYS THAT REACHED 90F+ (60 STAC AND FOCUS MONITORING SITES).

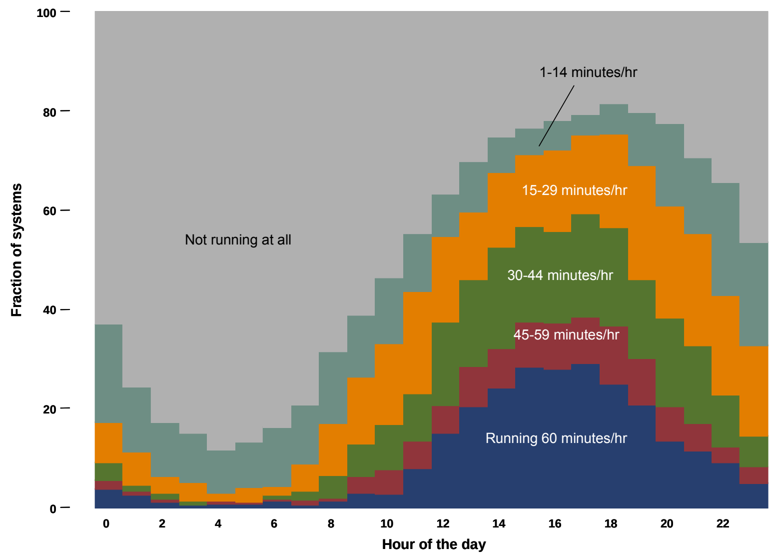
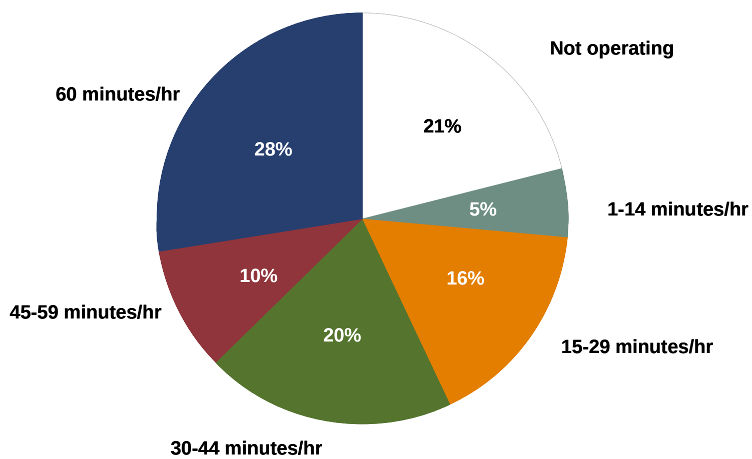


FIGURE 29, DISTRIBUTION OF CYCLING BETWEEN 3-7 PM ON DAYS THAT REACHED 90F OR HIGHER (60 STAC AND FOCUS MONITORING SITES).



Note, however, that this group of air conditioners is heavily weighted toward newer efficient units: 48 of the 60 sites included here are new SEER 13+ systems.

As noted previously, the data showing sudden-on cycles that last 60+ minutes following four or more hours of no operation suggest an element of occupant interaction with the systems (see page 23, Behavioral Aspects of Air Conditioning Use). Extending that analysis to system-peak conditions, the data suggest that sudden-on operation represents about 10% of the aggregate load during the hours between 3 and 7 pm. If these systems were operated earlier in the day, the empirical sizing analysis suggest that on average they would be cycling at about 75 percent duty cycle during the peak afternoon and evening hours instead of running flat out. This suggests that households delaying operation of the system (or reducing thermostat settings later in the day) adds about 2.5 percent to the aggregate air conditioning load during system peak, or about 70 watts of diversified peak load per unit, assuming about 2,800 watts for the average system at system peak.

TUNE-UP SAVINGS

As noted earlier, the 2007 Focus field research was oriented toward assessing the savings potential from tune-up of Wisconsin central AC systems by conducting field measurements of system EER before and after each adjustment.[12] That effort included older systems, new standard-efficiency systems, and new high-efficiency systems. The last group received Focus on Energy rewards that carry a nominal contractor requirement for charge and airflow verification on installation: the program receives charge and airflow verification data for about half of the rewards processed under the program.

The 2005 STAC study group also looked at refrigerant charge and airflow for 37 new SEER 13 and SEER 14 systems, but did not involve adjustments or field measurements of EER.

REFRIGERANT CHARGE

In terms of refrigerant charge, the field data from the two studies generally confirm what has been reported elsewhere in the country: a significant proportion of central AC systems are not charged properly, with the dominant charge error being that of undercharging (Table 8).

TABLE 8, AS-FOUND REFRIGERANT CHARGE ERROR (FOCUS AND STAC SAMPLES).

	Focus 2007			
As-found charge[a]	Older Systems (n=21)	New SEER 10-13 (n=10)	New SEER 14+[b] (n=30)	**STAC 2005** (new SEER 13-14) (n=33)
Overcharged	10%	0%	7%	6%
Slightly overcharged	5%	0%	7%	3%
OK	29%	30%	63%	15%
Slightly undercharged	14%	10%	10%	24%
Undercharged	43%	60%	13%	52%
Overall mischarged	71%	70%	37%	85%

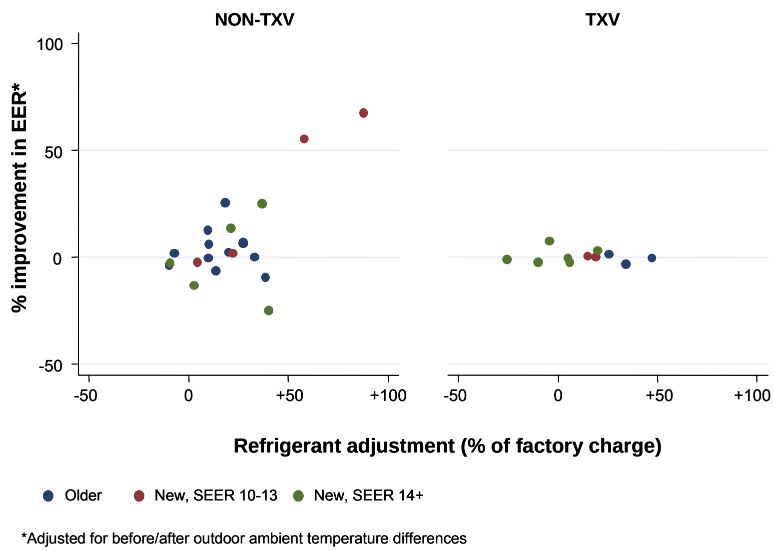
[a]The categorization of “slight” charge error differs for the Focus and STAC studies. For the Focus study (where charge was corrected), “slight” mis-charge is defined as charge adjustment of <10% of factory charge. For the STAC study, “slight” charge adjustment is defined as a 5-9F difference between measured and target superheat for non-TXV systems. Charge error for TXV systems for this study were defined as: <6F measured subcooling, undercharged; 6-9F, slight undercharge; and, ≥15F of subcooling, overcharged.

[b]This group received Focus on Energy rebates that included a “best practices” installation requirement addressing refrigerant charge and airflow.

While this would seem to imply significant savings potential from correcting charge errors, field EER measurements made for the Focus study (see Appendix A) suggest that the savings are modest in most cases. As Figure 30 shows, only a small proportion of refrigerant adjustments had a substantial impact on

[12] The equipment and methods used for these measurements—and issues associated with field EER measurements—are detailed in Appendix A.

FIGURE 30, FIELD EER IMPROVEMENT VS. REFRIGERANT ADJUSTMENT, NON-TXV AND TXV SYSTEMS (FOCUS 2007).



measured EER, and these were all for non-TXV systems. TXV systems—which made up about half of the new units tested in the 2005 STAC project, and 60 percent of the new systems tested in 2007—showed no change in EER on average after adjusting refrigerant charge. The non-TXV systems averaged about 8 percent savings, but this derives mostly from a few substantially undercharged systems with large corresponding improvement in EER.[13] Only about a quarter of systems requiring charge adjustment had charge errors of 30 percent or more. Downey and Proctor (2002) reported even fewer systems with substantial charge errors for a large population of California systems.

Test data from the Focus high-intensity sites (in which EER was measured at multiple charge levels for 15 sites) tends to confirm the assertion that the efficiency of TXV systems is less sensitive to charge error than non-TXV systems (Figure 31). In fact, on average the test data from these sites replicates very well meta-analysis of bench test data reported by Proctor (2000) (Figure 32).

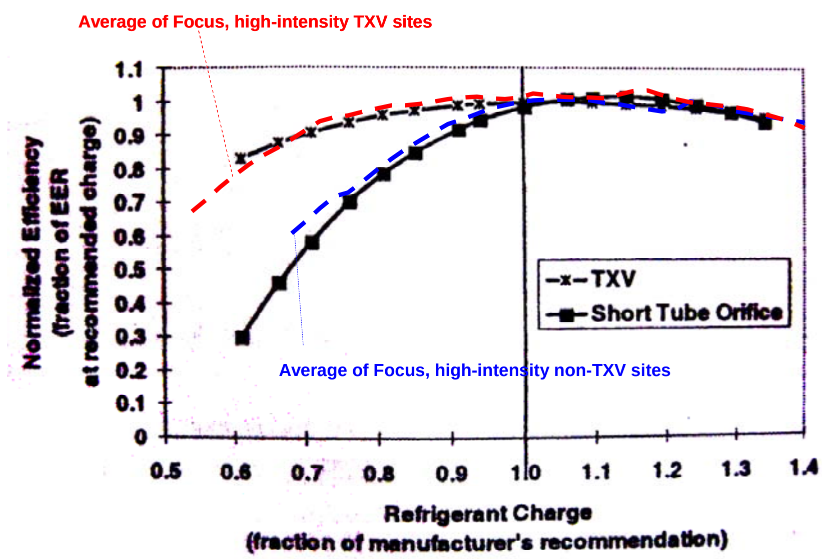
This finding stands somewhat in contrast to data reported by Mowris et al. (2004), who found no difference in tune-up savings between TXV and non-TXV systems. However, those data came from California, and involved mostly add-on TXVs in hot attics: in contrast, most TXVs tested here were integral to the evaporator coil and all were located in cool basements.

Though the sample sizes involved in the 2005 and 2007 studies are relatively small, the distribution of charge error appears to reasonably match those found in larger study groups (e.g., Downey and Proctor, 2002). If one combines this charge error distribution with the performance curves in Figure 32—together with an assumption that the majority of new units are TXV systems—aggregate savings from tuning refrigerant charge appears to be on the order of 3 to 5 percent.

[13] That some systems showed negative savings from refrigerant adjustment is likely due partly to changing outdoor conditions during the testing (though a temperature correction was used to try to compensate for this—see Appendix A), and uncertainties inherent in measuring cooling output. It is possible however, that a few sites tested under marginal conditions may have been mis-tuned.

FIGURE 31, FIELD RELATIVE EER VERSUS CHARGE ERROR (2007 FOCUS HIGH INTENSITY SITES).

FIGURE 32, AVERAGE RELATIVE FIELD EER VS. CHARGE ERROR FOR TXV AND NON-TXV SYSTEMS (COLOR), SUPERIMPOSED ON FINDINGS FROM PROCTOR (2000).

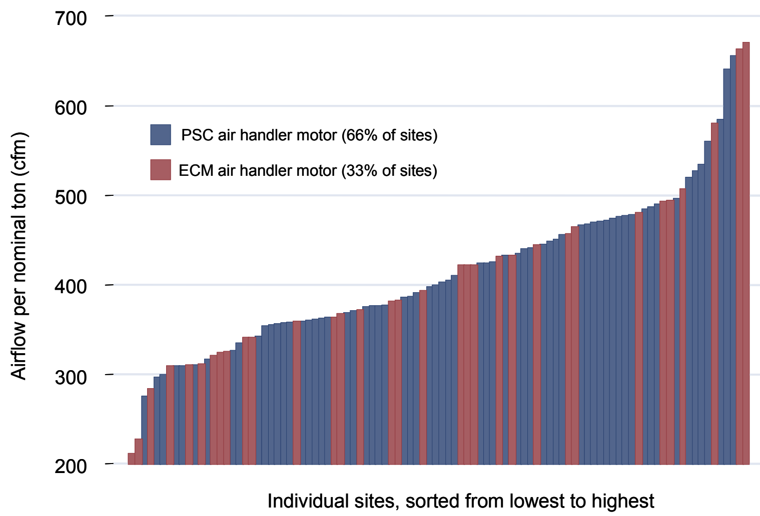


(Sites 101 and 108 excluded from above averages for data quality reasons.
Two-stage systems at Sites 109 and 115 also are excluded here.)

AIRFLOW

Airflow measurements (made with calibrated flow plates) at the 2005 STAC and 2007 Focus sites indicate that on average, Wisconsin central AC systems achieve about 400 cfm of airflow per nominal ton (Figure 33). The on-site testing revealed no difference in as-found airflow between systems with standard permanent magnet, split capacitor (PSC) blower motors and those with more advanced electronically commutated DC motors (ECM), despite the fact that the latter offers a wider range of available airflow settings.

FIGURE 33, AS-FOUND AIRFLOW (2005 STAC AND 2007 FOCUS SITES).



There is a correlation between the cooling capacity of the air conditioning system and airflow, however (Table 9): small systems tend to have excess airflow, and larger systems are more likely to have low airflow.

Also notable is that while high-efficiency systems that received Focus on Energy rewards that nominally require refrigerant charge and airflow adjustment showed lower frequency of refrigerant charge errors, they were no less likely to have airflow issues.

TABLE 9, AS-FOUND AIRFLOW, BY SYSTEM NOMINAL CAPACITY AND SYSTEM TYPE (2005 STAC AND 2007 FOCUS SITES).

	N	Mean cfm/ton (w 90% conf interval)	% <350 cfm/ton	% >450 cfm/ton
Nominal tons				
(2005 STAC + 2007 Focus)				
1.5	7	495 ± 81	0%	71%
2	40	439 ± 26	12%	42%
2.5	31	384 ± 20	32%	19%
3+	20	379 ± 25	30%	15%
Overall	98	413 ± 15	21%	32%
System age and SEER (2007 Focus only)				
Older	21	377 ± 28	33%	19%
New, SEER 10-13	10	393 ± 30	20%	0%
New, SEER 14+	30	420 ± 30	30%	33%
Overall	61	400 ± 18	30%	23%
Air handler type (2005 STAC +2007 Focus)				
ECM air handler	33	404 ± 30	33%	73%
PSC air handler	65	419 ± 16	15%	66%

In the 2007 Focus study, airflow adjustments were made to 20 systems with as-found airflow outside the range of 350-450 cfm per ton. (Six additional systems with low airflow could not be adjusted because the air handler was already at its highest speed setting.) These were divided about evenly between airflow increases (n=9) and airflow decreases (n=11), and most resulted in about a 100 watt change in air handler power consumption for each 100 cfm change in airflow (Figure 34).

Measured before and after EERs for these systems show that while *decreases* in airflow consistently increased EER (10 of 11 cases), in six of nine cases where airflow was *increased*, the measured EER of the system fell (Figure 35). This suggests that in at least some cases, the increased air handler power consumption from increasing airflow may more than offset the improvement in cooling output. The average EER improvement from these airflow adjustments was +5.6 percent (with a statistical confidence interval of ± 4.3 percentage points). However, this overall average derives from a mean EER improvement of +11.8 ± 4.9 percent for the 11 sites where airflow was decreased and -2.1 ± 5.2 percent for the 9 sites where airflow was increased.

FIGURE 34, CHANGE IN AIR HANDLER POWER VS. CHANGE IN AIRFLOW (FOCUS 2007 SITES).

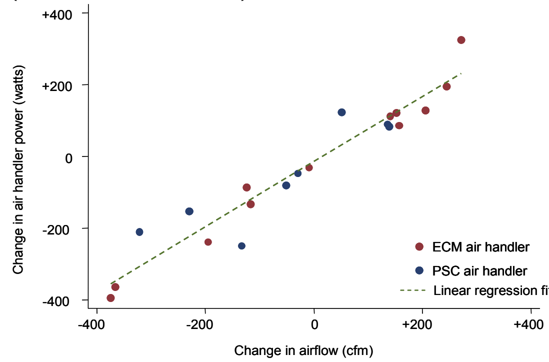
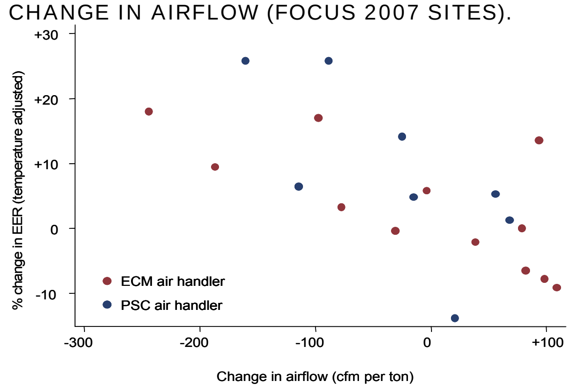


FIGURE 35, CHANGE IN FIELD EER VS. CHANGE IN AIRFLOW (FOCUS 2007 SITES).



Airflow adjustments to 1.5- and 2-ton systems appear to be particularly beneficial, because these are almost always reductions in airflow, and more than 40 percent of systems in this size range have high airflow. Of the 12 systems in this size range in the 2007 Focus study that received airflow reductions, nine involved reductions averaging about 200 cfm with an average of about a 200-watt reduction in air handler power draw. Based on typical seasonal operating hours (see *Seasonal Operating Hours*, page 30), this would translate into about 60 kWh of air handler energy savings. Similarly, the average airflow reduction to a 1.5- or 2-ton system would be expected to produce a bit more than 100 watts of peak load relief, given a typical duty cycle of about 50 percent (*Peak-Day Operation*, page 30)

CONDENSER COIL CLEANS

Condenser coils (and in one case the evaporator coil) were cleaned for all older systems in the 2007 Focus study (regardless of visual appearance). Savings from these cleans averaged about 7 percent (Table 10). Some units with visibly dirty coils showed no savings, and one site with relatively clean coils had more than a 10 percent measured improvement in EER. The latter may owe to issues with measuring EER in the field. The sites with the most visibly fouled coils showed the most savings, however (Figure 36).

TABLE 10, FIELD EER CHANGES FROM CONDENSER COIL CLEAN (FOCUS 2007).

Site	Change in Field EER[a]	Technician notes
55	-4.4%	"Back side some matting; otherwise clean"
56	+0.5%	"Clear and semi-dirty"
54	+0.5%	"Very dirty"
101	+1.0%	"Some fouling on back side; otherwise clean"
35	+2.4%	"Slight cottonwood seed"
69	+4.3%	"Dirty / Couple bushes"
62	+5.3%	
29	+9.0%	"Dirty"
102	+11.4%	"Clean"
103	+11.6%	"Some fouling"
104	+14.8%	"Substantial fouling; condenser partly buried in mulch"
28	+25.5%	"Dirty"
Mean	+6.8%	

[a]Adjusted for outdoor temperature (see Appendix A).

FIGURE 36, UNITS WITH LARGEST EER IMPROVEMENT FROM CONDENSER COIL CLEAN (FOCUS 2007).

Site 28

Site 103

Site 104

OVERALL SAVINGS FROM TUNE-UP

Table 11 summarizes the tune-up adjustments made at the 61 Focus sites, along with the mean overall improvement in EER from these adjustments. Overall, the data suggest something on the order of 5 percent average savings from tune-up efforts, albeit with 4 percentage points of uncertainty. Older systems and new standard efficiency systems were more likely to require refrigerant adjustment than new premium efficiency systems (which have nominal reward program requirements for charge and airflow adjustment), but the last group was more likely to require airflow adjustment.

Interpretation of the results is made difficult by the small sample for new standard-efficiency systems (n=10). Though this group averaged about 13 percent improvement in EER, that figure has 21 percentage points of uncertainty associated with it, meaning one can only reliably conclude that the savings in this group is somewhere between about -8 and +32 percent.

TABLE 11, OVERALL TUNE-UP ADJUSTMENTS AND MEAN EER IMPROVEMENT (61 FOCUS SITES).

Type of adjustment		**Older systems (n=21)**	**New SEER 10-13 systems[a] (n=10)**	**New SEER 14+ systems[b] (n=30)**	**All systems (n=61)**
	Airflow	6 (28%)	0 (0%)	14 (47%)	24 (39%)
	Refrigerant	15 (71%)	7 (70%)	11 (37%)	33 (54%)
	Coil clean	8 (38%)	0 (0%)	0 (0%)	8 (13%)
	Filter replacement	1 (5%)	0 (0%)	3 (10%)	4 (7%)
	Other	1 (5%)[c]	1 (10%)[d]	0 (10%)	2 (3%)
Number of adjustments made					
	0	0 (0%)	2 (20%)	9 (30%)	11 (18%)
	1	14 (67%)	8 (80%)	16 (53%)	38 (62%)
	2	3 (14%)	0 (0%)	3 (10%)	6 (10%)
	3	4 (19%)	0 (0%)	2 (7%)	6 (10%)
Mean % EER improvement	Adjusted systems	3.1 ± 6.1%	13.3 ± 20.9%	4.6 ± 3.9%	5.4 ± 4.1%
	All systems	3.1 ± 6.1%	10.6 ± 16.2%	3.2 ± 2.7%	4.4 ± 3.4%

[a]Two systems were SEER 10; 8 were SEER 13.

[b]About half (n=13) of this group of participants received Focus on Energy rebates that included a "best practices" installation requirement addressing refrigerant charge and airflow.

[b]Opened registers.

[c]Closed bypass damper for humidifier.

High Savers

Eight sites in the 2007 Focus study showed EER improvements of 15 percent or more following tune-up. As Table 12 shows, the contributors to large improvements in EER were about evenly divided between refrigerant adjustment and airflow adjustment. Only two of the eight sites had multiple significant interventions. Notably, three of the eight highest savers were high-efficiency systems with nominal reward-program requirements for proper refrigerant charge and airflow.

TABLE 12, SYSTEMS WITH 15+ PERCENT TUNE-UP SAVINGS (FOCUS 2007).

Site	System Description	% EER improvement	Adjustments (key savings contributor underlined)
41	3-ton, non-TXV, non-ECM, R-22, SEER 10	68%	Corrected 88% undercharge.
65	2-ton, non-TXV, non-ECM, R-22, SEER 13	55%	Corrected 58% undercharge.
69	1.5-ton, non-TXV, non-ECM, R-22, SEER 10	31%	Corrected 33% undercharge. Reduced airflow 131 cfm for 245 watt reduction in air handler power. Cleaned condenser coil.
28	2.5-ton, non-TXV, ECM, R-22, SEER 10	27%	Cleaned condenser. Corrected 27% undercharge. Small airflow adjustment had negligible impact.
15	2-ton, TXV, non-ECM, R410a, SEER 14	26%	Reduced airflow 320 cfm for 210 watt reduction in air handler power.
64	2.5-ton, non-TXV, non-ECM, R-22, SEER 10	25%	Corrected 18% undercharge.
63	3-ton, non-TXV, non-ECM, R-22, SEER 14	25%	Corrected 37% undercharge.
34	2-ton, TXV, ECM, R410a, SEER 14	18%	Reduced airflow 195 cfm, for 240 watt reduction in air handler power. Correction of 4% overcharge also improved EER slightly.

Tune-up Impacts on Peak Load

While a 5 percent improvement in system efficiency can be expected to result in a comparable reduction in seasonal energy consumption, the same cannot be said about tune-up impacts on peak loads. This is partly due to the impact mitigation that arises from some systems not being in operation at all during utility system peak, but the fact that some systems are running at 100 percent duty cycle during system peak also plays a role. Efficiency improvement from tune-up arises from either increasing the cooling output of the system or reducing the input power (or a combination thereof). Reducing input power requirements to a system that runs flat out during system peak will contribute commensurately to peak load reduction. But simply increasing the cooling output of such a system will not contribute to peak load reduction unless the cooling output increase causes the system to cycle during system peak. The situation is complicated further by considering that increasing cooling output through tune-up adjustments may shorten the length of time that a system will run flat out, or may be enough to cause such a system to cycle instead of running at 100 percent duty cycle.

When assumptions about the likely range of values for the various factors above are combined with the sampling uncertainty for tune-up efficiency improvement from the Focus 2007 study, the range of estimated mean peak wattage reductions shown in Table 13 is obtained. Overall, this analysis suggests that positive savings from system tune-up are likely, but that these savings are unlikely to average more than 50 Watts per system across a population of units that receive attention to refrigerant charge and airflow.

TABLE 13, ESTIMATED RANGE OF PEAK LOAD SAVINGS FROM SYSTEM TUNE-UP (FOCUS 2007).

	Mean peak load reduction at 93^{o}F (watts)
Combined sample (n=61)	+1 to +54
Older systems (n=21)	-46 to +66
New, SEER 10-13 systems (n=10)	-30 to +134
New, SEER 14+ systems (n=30)	+5 to +56

Range above is the 5th to 95th percentile for Monte Carlo model incorporating sampling uncertainty, plus the following assumed ranges:

- percent of systems not operating at peak: 15 to 25%
- percent of systems running flat out at peak: 25 to 35%
- average duty cycle for cycling systems at peak: 40 to 60%
- percent of flat-out system affected by tune-up: 10 to 40%
- flat-out system impact as a percent of mean performance improvement: 25 to 75%
- system power variation with ambient temperature: 0.7 to 0.8% per F^{o}
- EER variation with ambient temperature: 0.1 to 0.2 EER points per F^{o}

TWO-STAGE SYSTEMS

The STAC research project included monitoring of a number of two-stage systems. The original research plan called for monitoring 20 such systems over a single cooling season, but recruiting issues and the cool weather encountered in 2004 led instead to monitoring 12 systems in 2004, of which three agreed to continue monitoring in 2005. Five additional sites were monitored in 2005 (2 sites) and 2006 (3 sites).[14] Unfortunately, data logging and other site issues made three of the sites unusable for analysis.

To better gauge the effect of two-stage capability, the sites were experimentally forced to run only in high stage during part of the cooling season. For the 2004 sites, this was accomplished by installing a seven-day programmable timer to force high-stage-only operation for three days of the week, with two-stage operation allowed for the remaining four days. For three sites monitored in 2005 and 2006, a split-season approach was used in which the system was allowed to operate in two-stage mode for the first part of the cooling season, and then was re-configured for high-stage-only operation for the remainder of the cooling season. Two sites could not be easily reconfigured for high-stage only operation, and were allowed to operate in two-stage mode throughout the monitoring period.

HUMIDITY AND TEMPERATURE

The indoor psychrometric data (recorded at the thermostat location) suggest that two-stage capability by itself does not confer a systematic advantage in terms of humidity control: none of the sites where the split-week approach to mode control was used showed a statistically (or practically) significant difference in average indoor temperature or humidity between two-stage operation and high-stage-only operation (Table 14).

There were significant differences in indoor conditions between the two modes of operation for the split-season sites, but the two modes were not well balanced in terms of outdoor conditions for these sites, so it seems more likely that the observed differences in indoor conditions were due to confounding effects of weather rather than the operating mode of the air conditioner. (In contrast, the average—and distribution of—outdoor temperature was well matched between the two operating modes for all of the split-week sites.)

[14] With one exception (Site 12) the monitored systems used reciprocating compressors that drew about ½ of high-stage power in low-stage operation; Site 12 had a scroll compressor that drew about 70 percent of high-stage power in low-stage operation.

TABLE 14, AVERAGE DAILY INDOOR AND OUTDOOR CONDITIONS FOR TWO-STATE SITES (DAYS WITH 30+ MINUTES OF SYSTEM OPERATION).

	Days of data		Outdoor Temperature (°F)		Indoor conditions[a] Temperature (°F)		Relative humidity (%)		Dewpoint (°F)	
Site	Two stage	High stage only	Two stage	High stage only	Two stage	High stage only	Two stage	High stage only	Two stage	High stage only
(Split-week operation[b])										
1[c]	26	9	71.1	71.4	75.4	75.0	50.5	49.8	55.7	55.0
2	37	12	71.7	69.9	71.4	71.1	52.6	54.2	53.0	53.4
3	35	17	67.6	67.2	68.4	68.2	44.0	43.0	45.6	44.8
4	35	25	71.1	71.3	69.9	69.9	51.3	51.4	51.0	51.1
5[d]	22	15	71.1	70.8	69.5	69.8	62.5	62.8	56.1	56.5
6	22	15	73.0	72.2	76.5	76.2	48.6	51.2	55.5	56.7
7	49	34	73.1	73.0	70.0	70.1	51.5	51.0	52.4	52.2
8	95	73	71.3	70.9	75.9	75.8	49.7	49.6	56.2	56.0
9	58	41	75.1	74.7	76.1	76.2	51.3	50.6	57.3	57.0
(Split-season operation)										
10[d]	33	40	***73.7***	***69.2***	**72.8**	**71.7**	50.2	50.0	***54.2***	***53.2***
12	12	10	***77.0***	***72.7***	74.2	75.0	67.5	67.8	63.3	64.1
13	30	40	***77.0***	***62.6***	***76.3***	***71.5***	***57.2***	***60.8***	***60.0***	***57.2***
(Two-stage only)										
14	43		73.2		74.0		51.2		55.4	
16	21		68.8		75.1		52.3		56.4	

[shaded] = statistically significant difference at 90% confidence level

[a]At thermostat.

[b]Two-stage operation on Monday, Tuesday, Thursday and Saturday; high-stage-only operation on Wednesday, Friday and Sunday.

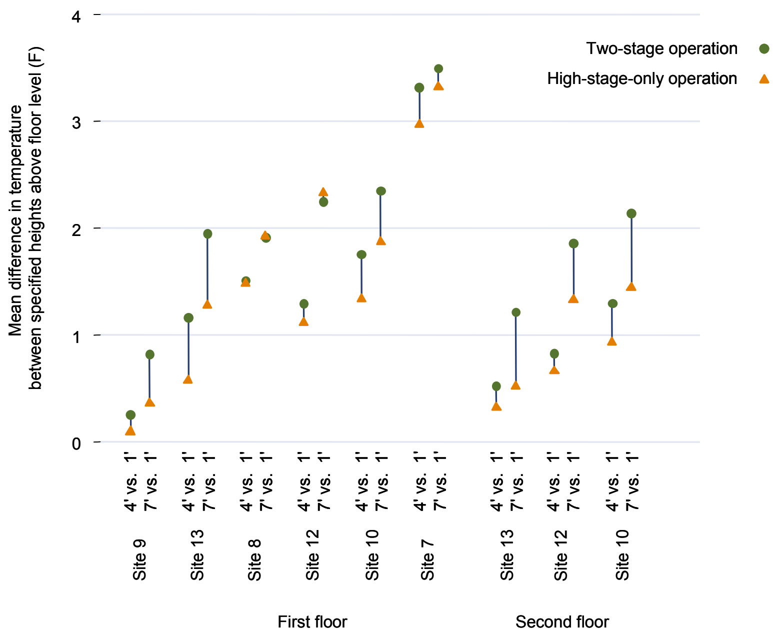
[c]Practiced continuous-fan operation for part of the monitoring period.

[d]Practiced continuous-fan operation throughout the monitoring period.

Also noteworthy is that 5 of the 14 monitored sites (35%) maintained an average indoor temperature of 72°F or less, and three (21%) averaged less than 70°F indoor temperature on days when the system was used. In contrast, only about a quarter of the 58 single-stage sites monitored in 2005 and 2007 showed average indoor temperatures below 72°F, and only two of 58 (3%) maintained an average indoor temperature of less than 70°F. This could be a sign that purchasers of two-stage systems are more likely to be heavy air conditioning users who desire cooler-than-average indoor conditions, or it could simply be an artifact of cooler weather in 2004 when the majority of the data were collected (though Site 7 showed comparably low indoor temperatures in *both* the cool 2004 summer and the considerably hotter 2005 cooling season).

The monitoring in 2005 and 2006 included the installation of temperature probes to monitor indoor temperature at 1 foot, 4 feet and 7 feet above the floor to gauge stratification effects. As Figure 37 shows, two-stage operation had slightly higher vertical temperature stratification than did high-stage-only operation—though the observed amount of stratification was not particularly large in general. The finding is consistent with the notion that the reduced airflow associated with low-stage operation results in less thorough mixing of conditioned air in the living space.

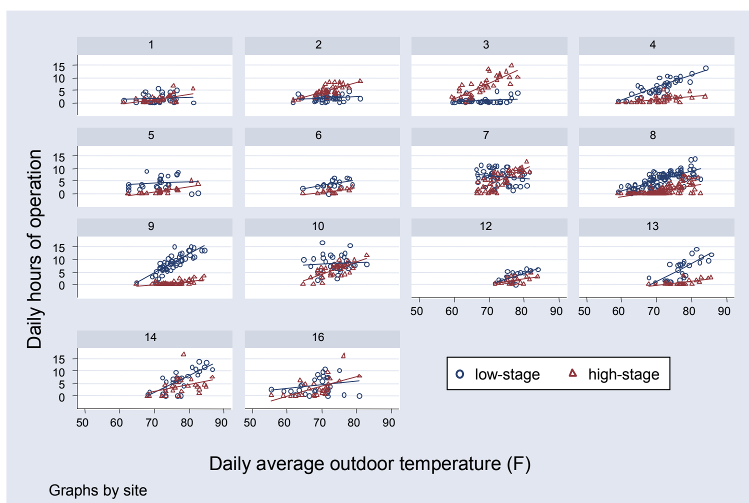
FIGURE 37, TEMPERATURE STRATIFICATION FOR TWO-STAGE SITES, BY MODE OF OPERATION (DAYS WITH 30+ MINUTES OF SYSTEM OPERATION.



STAGING

Linear models of hours of operation in low- and high-stage versus outdoor temperature (Figure 38) suggest a wide range across the sites in both seasonal operating hours and the proportion of the time that the systems operate in high-stage, the latter ranging from less than 10 percent to almost 90% (Table 15).

FIGURE 38, DAILY HOURS OF OPERATION (LOW- AND HIGH-STAGE) FOR TWO-STAGE SITES IN TWO-STAGE MODE.



This suggests that the amount of high-stage operation is situationally specific to both the relative sizing of the system and how the system is used by the occupants. Of the three sites with more than 50 percent high-stage operation, two (Sites 1 and 2) have run-time and indoor temperature patterns that suggest the occupants either keep the system off or set up the thermostat temperature until late afternoon and early evening. The third (Site 3) maintains a very low indoor temperature (Figure 39). However, other sites with similar characteristics (e.g., Site 10) do not have as high proportions of high-stage operation.

TABLE 15, OPERATING HOURS (LOW- AND HIGH-STAGE) FOR TWO-STAGE SITES IN TWO-STAGE MODE.

		Observed					Estimated seasonal average[c]	
			Operating hours			high stage	Total	high stage
Site[a]	t-stat type[b]	Days of data	low-stage	high-stage	total	operation (% of total)	operating hours	operation (% of total)
12	N	35	64	31	95	32%	146	32%
13	P	39	154	17	171	10%	212	10%
1	P	93	49	44	94	47%	258	53%
9	P	133	468	32	500	6%	268	6%
6	N	91	94	20	115	18%	297	22%
16	P	86	119	89	209	43%	328	49%
14	P	49	175	100	275	36%	336	38%
5	P	82	103	25	128	19%	364	24%
2	P	96	71	175	246	71%	532	73%
8	P	145	551	123	674	18%	649	20%
4	N	87	235	49	284	17%	717	18%
3	P	65	36	242	278	87%	740	88%
7	P	54	307	237	544	44%	871	45%
10	P	34	284	196	480	41%	1155	36%
						Mean	491	37%
						median	350	34%

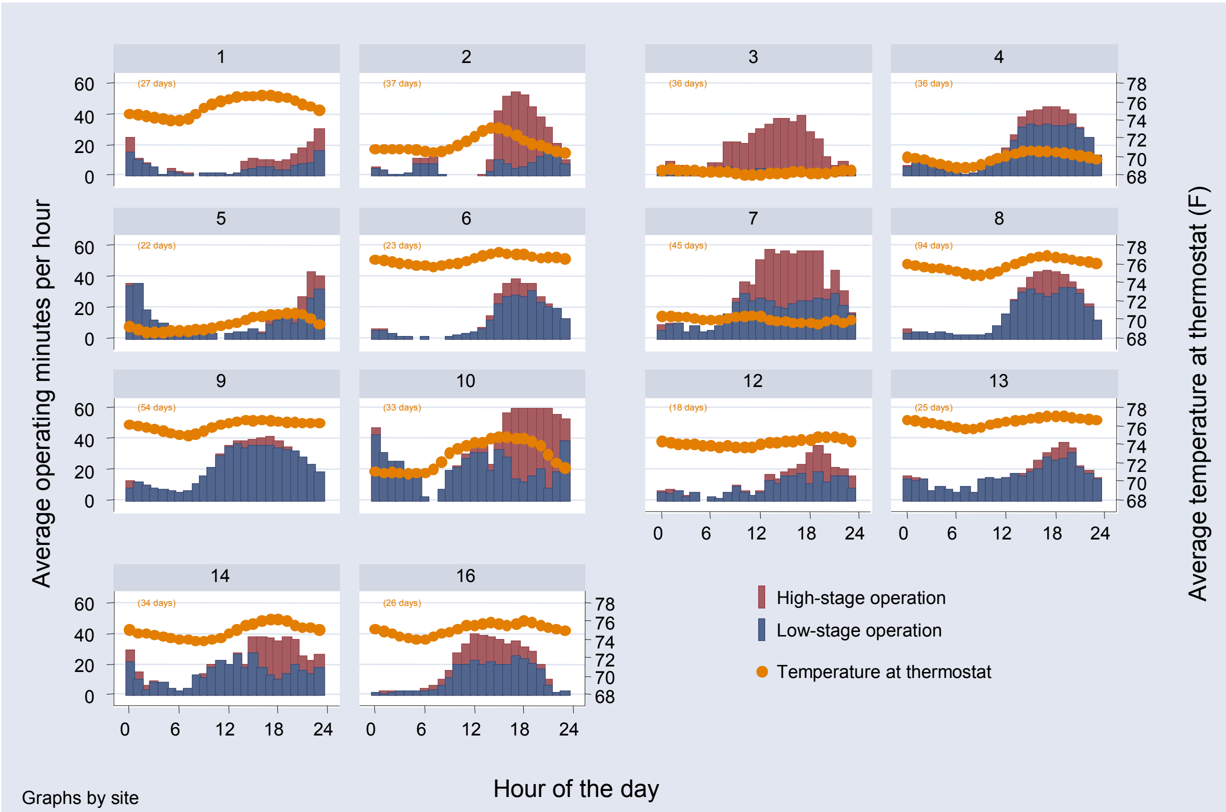
[a]In ascending order by estimated seasonal operating hours.

[b]Thermostat type: N = non-proprietary staging (calls for second stage after fixed period of low-stage operation or if temperature error exceeds fixed level); P = uses proprietary algorithms for staging control.

[c]Based on linear regressions of daily hours of operation for each stage versus outdoor temperature. Includes logistic fit of probability-of-use vs. outdoor temperature. Normalized to 1987-2006 distribution of daily outdoor temperature for Madison or Milwaukee, depending on site location.

The type of thermostat may also play a role: most of the systems studied here used thermostats with proprietary staging control algorithms (these thermostats also have a humidity-sensing function used to adjust cooling-speed airflow). Some two-stage thermostats, however, switch to high-stage operation only after a fixed period or if the temperature error (the difference between the actual space temperature and the set point) exceeds a fixed level.

FIGURE 39, OVERALL AVERAGE HOURLY OPERATION PROFILES (AND INDOOR TEMPERATURE) FOR TWO-STAGE SITES IN TWO-STAGE OPERATION, BY SITE (DAYS WITH 30+ MINUTES OF AC OPERATION).



ENERGY USE

Energy consumption was not directly measured for these sites, but can be reasonably inferred from monitored compressor and air-handler amp draws combined with one-time measurements of voltage and power factor (see Appendix A).

Linear models of daily energy use versus outdoor temperature (Figure 40), combined with 1987-2006 distributions of cooling season daily temperature (and individual logistic models of probability of system use versus temperature) yield the seasonal energy consumption estimates shown in Table 16 (only ten of the 14 sites had sufficient data for this analysis).

The results are inconclusive in terms of whether 2-stage operation results in energy savings: four sites had statistically significant lower seasonal energy consumption estimates under two-stage operation, but two sites also had statistically significant *higher* consumption estimates in this mode—and four sites had differences that were not statistically significant. Averaging the point estimates of savings across the ten sites results in only a small (and not statistically significant) difference between two-stage and high-stage-only operation.

FIGURE 40, DAILY KWH VERSUS OUTDOOR TEMPERATURE FOR TWO-STAGE SITES, BY MODE OF OPERATION (DAYS WITH 30+ MINUTES OF AC OPERATION).

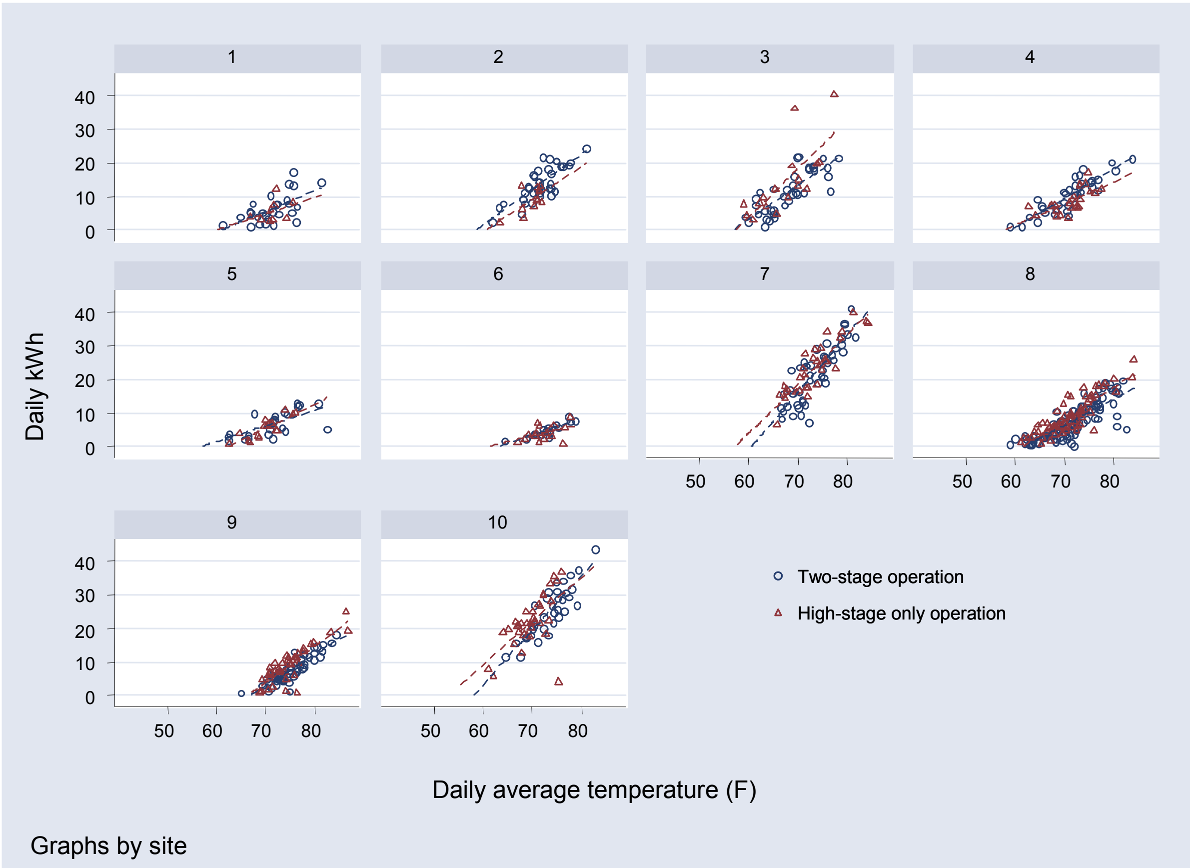


TABLE 16, ESTIMATED SEASONAL ENERGY USE, BY MODE OF OPERATION.

Site	Two-stage operation (kWh)	High-stage-only operation (kWh)	Difference (kWh)	Difference (%)	Statistically significant difference?
1	490 ± 82	410 ± 161	79 ±180	19% ±51%	N
2	1,129 ± 69	901 ± 198	228 ±210	25% ±29%	Y
3	1,106 ± 93	1,523 ± 274	-417 ±289	-27% ±14%	Y
4	942 ± 59	759 ± 78	183 ±98	24% ±15%	Y
5	468 ± 64	482 ± 59	-13 ±87	-3% ±18%	N
6	309 ± 26	257 ± 61	52 ±66	20% ±30%	N
7	1,576 ± 81	1,683 ± 78	-107 ±112	-6% ±6%	N
8	689 ± 44	868 ± 51	-179 ±68	-21% ±7%	Y
9	217 ± 15	268 ± 25	-50 ±29	-19% ±9%	Y
10	1918 ± 108	2,129 ± 166	-211 ±197	-10% ±9%	Y
		Average:	-43 ±113	0.3% ±12%	N

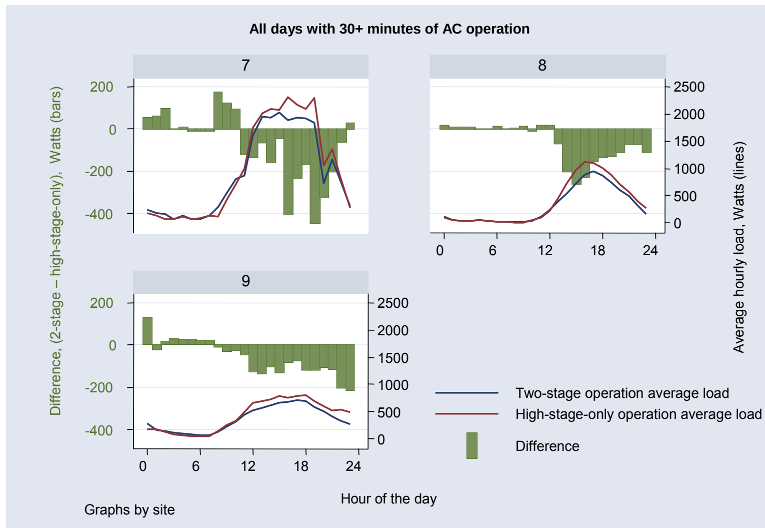
Note: All uncertainties (and statistical significance) are at 90% confidence level. Uncertainty for overall averages represent sampling uncertainty for point estimates.

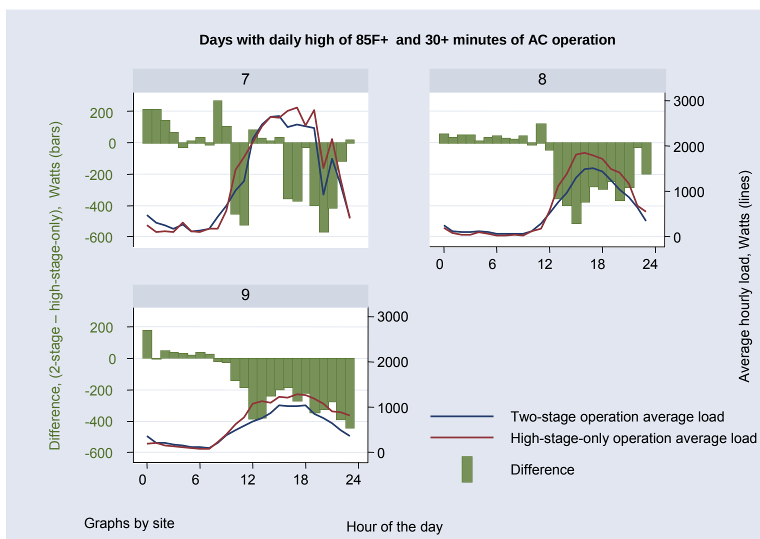
HOURLY LOAD PROFILES

Only the three sites monitored in 2004 and 2005 under the split-week approach had sufficient data under similar weather conditions to allow for assessment of differences in hourly electrical load between two-stage and high-stage-only operation.

As Figure 41 shows, the three sites showed 200 to 400 watt lower hourly electrical loads in two-stage operation during the peak afternoon and evening hours, even on hot days. This suggests that two-stage systems can provide peak demand savings in at least some cases.

FIGURE 41, AVERAGE HOURLY LOAD PROFILES (AND DIFFERENCE BETWEEN MODES) FOR THREE TWO-STAGE SITES, OVERALL AND FOR HOT DAYS.





IMPACT OF ECM FURNACES ON AC EFFICIENCY

A previous study of electricity consumption by new furnaces in Wisconsin (Pigg, 2003) suggested that furnaces with ECMs have wattage draws that are 150 to 175 watts lower than furnaces with standard PSC air handler motors. Furnace power data from the Focus 2007 study confirm this finding, with ECM air handlers averaging about 35 percent lower power consumption (Table 17).

Cooling savings from this difference manifests in two ways: not only does the lower power consumption reduce air handler power, but there is less motor heat to be removed by the cooling system as well. Given compressor COPs of 2.5 (for older systems) to 3.5 (for new premium efficiency systems) under peak conditions, the overall impact on cooling efficiency would be roughly 25 to 40 percent higher than the direct air handler power savings.

When the previously-described proportions of systems operating and running at 100 percent duty cycle are factored in, the mean diversified peak impact of ECM air handlers is estimated at about 170 ± 50 watts. Over a typical Wisconsin cooling season (300 hours of system operation), savings of about 70 ± 20 kWh could be expected (for households that do not practice continuous-fan operation).

Note, however, that one postulated benefit of ECM furnaces is *not* supported by the 2005 STAC and 2007 Focus field study data: namely, that the wider airflow range of ECMS—and the fact that airflow can be specified via on-board dipswitches—will result in more appropriate airflow settings by installers. As Table 9 shows, ECM air handlers were somewhat more likely to have either high or low airflow than PSC systems (though only the proportion with lower airflow is statistically significant).

TABLE 17, AIR HANDLER POWER FOR PSC AND ECM AIR HANDLERS (FOCUS 2007).

	Mean air handler power (watts per 1000 cfm)		
	PSC (n=37)	**ECM** (n=24)	**Difference**
As-found	528 ± 35	341 ± 43	187 ± 60
Post-adjustment	517 ± 33	320 ± 40	197 ± 51

Note also that the monitoring data from the STAC and Focus research support field research from other parts of the country showing that air handler power requirements (and static pressures) are considerably higher than assumptions used in the federal test procedure for SEER. That procedure assumes 350 Watts per 1,000 cfm, a value that this field data shows as being achieved by the ECM systems but not by the more common PSC furnaces. (Appendix C provides additional data on air hander power and static pressures found in this field research.)

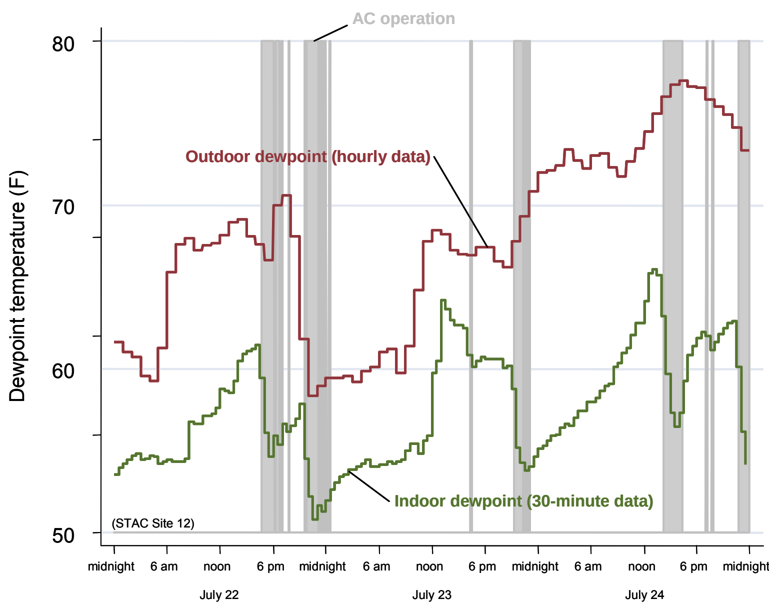
HUMIDITY CONTROL

Humidity control is an oft-cited concern for central air conditioning systems. Systems that cannot maintain acceptable indoor humidity will be less likely to be judged acceptable by users, even if they can maintain the desired temperature set point. The monitoring data collected for the STAC 2005 and Focus 2007 studies allow for an assessment of differences in humidity control across a variety of sites.

In aggregate, during hours when the system operated, indoor humidity averaged 47 percent across the 61 sites with recorded humidity data. Only a small proportion (12 percent) of sites averaged 55 percent relative humidity or higher, and none averaged more than 60 percent relative humidity during hours of system operation.

The monitoring data generally showed that indoor relative humidity tends to track outdoor humidity when the air conditioning is not operating, but drops quickly when the system operates (Figure 41). A typical 2-ton system removes about ½ gallon of condensate every 30 minutes in steady-state operation, enough to reduce indoor humidity in the average home from about 65 to 50 percent.

FIGURE 42, EXAMPLE OF INDOOR HUMIDITY VARIATION OVER THREE DAYS.



The difference between indoor and outdoor dew point is a useful indicator of the ability of the system to provide humidity control. When the sites are compared in this way, average daily hours of run-time is clearly an important predictor of humidity control (Figure 43): systems that operate fewer hours have indoor humidity that is closer to outdoor humidity than systems that operate more hours. This indicates that system sizing is important for humidity control.[15]

As Figure 43 also shows, households that practice continuous-fan operation also tend to be on the low end of humidity reduction for a given number of operating hours. This arises because continuous-fan operation re-evaporates moisture stored on the evaporator between cycles, which both directly adds to the amount of humidity in the home and increases the amount of operating time required to re-saturate the coil in subsequent cooling cycles before condensate drains off the coil.

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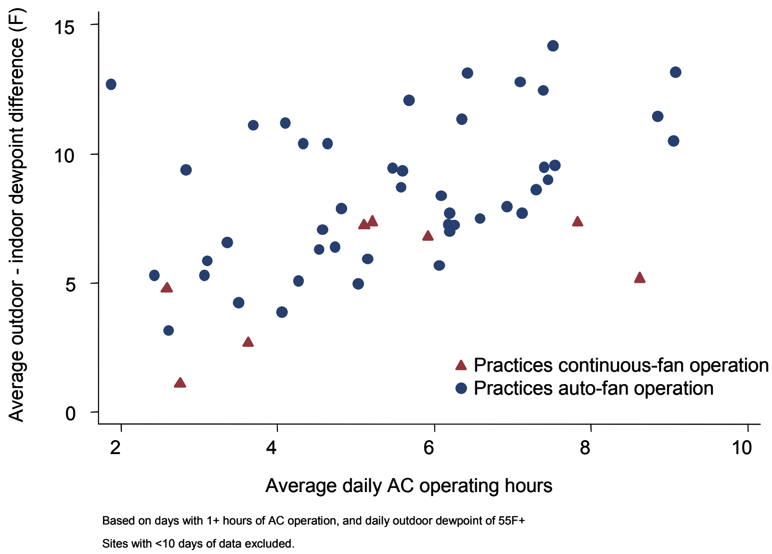
[15] This holds true even when accounting for the fact that outdoor dewpoints tend to be higher when outdoor temperatures are higher and air conditioners operate more hours.

FIGURE 43, DEWPOINT DEPRESSION VS. DAILY OPERATING HOURS (STAC 2005 AND FOCUS 2007 SITES).



Moreover, blower door test data for the 2005 STAC sites indicates that air leakage is an important (and statistically significant) predictor of indoor humidity (Table 18): after controlling for differences in run-time, systems operating in leakier homes reduce humidity less than do those in tighter homes.

Interesting, what is *not* a statistically significant predictor of indoor humidity in this analysis is airflow (Table 18). This is not to say that airflow has no role in indoor humidity control (one of the sizing-swap homes showed a difference in indoor humidity that is likely due to airflow issues—see page 18). Rather, the data suggest that system sizing, buildings air leakage and fan operation practices are perhaps more important than airflow in this regard.

TABLE 18, REGRESSION MODEL OF DEWPOINT DEPRESSION.

Dependent variable: site Δ dew point (outdoor minus indoor) average of days with 1+ hours of AC operation and outdoor dew point of 55F of higher

n = 35 sites
R^2=0.363

Parameter	Coefficient (with 90% confidence interval)	t-statistic
Avg. daily run-time hours	0.533 ± 0.471	1.92
Estimated natural air changes per hour (blower door)	-5.05 ± 3.67	-2.33
Binary variable for continuous-fan operation	-3.84 ± 2.38	-2.74
Air handler cfm per ton of capacity (x 1,000)	1.22 ± 7.54	0.27
Constant	6.84 ± 4.39	2.64

SYSTEM CYCLING BEHAVIOR

The 58 STAC and Focus sites monitored in 2005 and 2007 afford a detailed examination of system cycling by virtue of the data loggers that date- and time-stamped the beginning and end of each cycle. Analysis of the roughly 28,500 operating cycles recorded across 61 sites shows that while operating cycles of less than 30 minutes make up almost 90 percent of the total cycles, they constitute less than half of the aggregate operating time (Table 19).[16] In aggregate the monitoring data indicate that half of total operating time occurs at cycle lengths of 51 minutes or longer, and only about 12 percent of aggregate operating time is attributable to cycles that last less than 10 minutes. As noted previously (see *Behavioral Aspects of Central AC Use*, page 23), long operating cycles from occupants turning on the system or reducing the thermostat set point later in the day is a significant factor in air conditioning operation for some households.

The distribution of cycle length is of importance for SEER ratings, which incorporate a cycling degradation coefficient (C_d) to account for reduced performance under cycling conditions. The SEER test procedure uses a 6-minute-on, 24-minute-off cycling test to assess cycling degradation. Only three percent of the aggregate monitored run-time from the STAC and Focus sites came from cycles that were 6-minutes or less in length. This suggests that from a cycling standpoint, the current SEER procedure may underestimate seasonal efficiency.

TABLE 19, DISTRIBUTION OF CYCLES FOR STAC AND FOCUS MONITORING SITES.

Cycle length	% of observed cycles	% of aggregate operating time
<5 min.	12.4%	1.9%
5-9 min.	32.5%	10.7%
10-14 min.	26.7%	14.5%
15-29 min.	16.7%	14.9%
30-59 min.	5.7%	10.6%
60-179 min.	4.2%	19.1%
180+ min.	1.9%	28.2%

Based on aggregate analysis of 28,487 operating cycles recorded for 61 sites with 10+ days of data in 2005 and 2007.

[16] These data are not normalized for weather, and the amount of data collected per site varies from less than 20 hours of operating time to more than 400 hours (with a median of about 170 hours). The runtime-weighted average outdoor temperature for the data set is 78°F, which is close to the estimate of weather-normalized seasonal mid-load temperature.

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APPENDIX A — FIELD MONITORING AND TESTING DETAILS

This appendix describes in more detail the procedures and equipment used for testing and monitoring for the various efforts described in this report.

STAC PROJECT FIELD RESEARCH PROJECT (2007)

Field testing and monitoring of new systems

Field tests of as-found refrigerant charge were made using a Honeywell Service Assistant for refrigerant line pressure and temperature, an Energy Conservatory True Flow Air Handler Flow meter and an Energy Conservatory DG-700 digital manometer (for airflow and static pressure measurements). Supply and return dry-bulb and wet-bulb temperatures were measured at a single location each using a Testo Model 605-H2 Humidity Wet Bulb Stick. Compressor and air handler current draw was measured with a standard clamp-on AC ammeter.

Field monitoring of system cycling was accomplished at each site by installing a relay on the Y and common terminals in the furnace cabinet, with the relay switch terminals connected to a Hobo H8 State logger that recorded the date and time each time the thermostat initiated or terminated a call for cooling. These data loggers can record about 1,000 on/off cycles without requiring a download. In cases where the occupants reported using continuous-fan operation during the summer, a similar arrangement was used to monitor fan-on status via the thermostat G terminal.

Indoor temperature and humidity was logged at the thermostat by hanging a Hobo H8 temperature/RH data logger from the thermostat. The data loggers recorded a snapshot of temperature and humidity every 30 minutes, with 82 days of storage capacity.

Two-stage systems

Monitoring of the two-stage systems included the parameters listed in Table 20. In addition, spot measurements of system airflow, power consumption and cooling output were made at most sites. Airflow was measured using the Energy Conservatory True Flow air handler flow meter; power consumption was measured using a Dent Instruments Elite Pro data logger. Supply- and return-air psychrometrics were measured using a Testo Model 605-H2 Humidity Wet Bulb Stick.

To experimentally force high-stage operation on some days of the week, a 7-day programmable Grasslin timer was wired into the thermostat terminal at the furnace. The timer was configured so that on three days of the week (Wednesday, Friday and Sunday), any Y1 calls from the thermostat (low-stage cooling) were re-directed to the Y2 terminal (high-stage cooling); for the remaining days of the week, the system was allowed to operate in two-stage mode.

TABLE 20, STAC STUDY TWO-STAGE SYSTEM MONITORING POINTS AND SENSOR DESCRIPTION.

Outdoor temperature	Hobo Pro data logger with external sensor, recording 5-minute snapshot data.
Indoor living space (and basement) temperature/RH	Hobo H8 temp/RH logger at thermostat (or basement), recording 15-minute snapshot data.
Indoor temperature stratification, 1', 4' and 7' above floor (2005-2006 monitoring only)	Hobo TMC*x*-HA and TMC*x*-HB temperature sensors connected to Hobo H8 4-channel data logger, recording 5-minute snapshot data.
Compressor and air handler amps	Hobo 0-20 amp CT connected to Hobo H8 4-channel datalogger, recording 90-second snapshot data.
Compressor and air handler status	Relay wired to thermostat Y and G terminals, connected to Hobo State logger.
Supply and return air temperature (2005 and 2006 monitoring only)	Hobo Pro data logger with external sensor, recording 2-minute snapshot data.
Condensate production	Tipping bucket rain gauge mechanism connected to Hobo U11 event logger.

Sizing-swap sites

For the two sites that were monitored as part of the sizing swap experiment, the parameters shown in Table 21were monitored.

TABLE 21, STAC STUDY SIZING-SWAP SITE MONITORING POINTS AND SENSOR DESCRIPTION.

Outdoor temperature	Hobo Pro data logger, recording 10-minute snapshot data.
Indoor living space (1st and 2nd floors) temperature/RH	Hobo H8, U12 or Pro temp/RH logger, recording 5-minute snapshot data.
Basement temperature/RH	Hobo U8 temp/RH logger, recording 15-minute snapshot data
Indoor temperature stratification, 1', 4' and 7' above floor	Hobo TMC*x*-HB temperature sensors connected to Hobo U12 4-channel data logger, recording 2-minute snapshot data.
Compressor and air handler power	Ohio Semitronics watt-hour transducer (1 wh/pulse) connected to Hobo microstation
Compressor and air handler amps	Hobo 0-20 amp CT connected to Hobo H8 4-channel datalogger, recording 90-second snapshot data.
Compressor and air handler status	Relay wired to thermostat Y and G terminals, connected to Hobo State logger.
Supply duct pressure	Autotran Series 7000, 0-1" in. H_20, pressure transducer, connected to Campbell 21X data logger recording 15-minute average data (data recorded only during air handler operation)

Supply and return air temperature	Field-fabricated thermocouple junctions connected to Cambell 21X data logger recording 15-minute average data (data recorded only during air handler operation).
Solar gain	Hobo H8 temperature/light loggers placed in various windows, recording 15-minute temperature and light intensity snapshot data.
Condensate production	Tipping bucket rain gauge mechanism connected to Hobo microstation.

FOCUS ON ENERGY FIELD RESEARCH PROJECT (2007)

Two similar protocols were used for this project: (1) a standard protocol was employed for the majority of sites that received as-found testing followed by adjustments to refrigerant charge and airflow and/or compressor coil clean; (2) a high-intensity protocol was used for a small number of sites where refrigerant charge and airflow was deliberately run through a range of values to assess the impact on field-measured EER.

Standard Protocol

1. Install data logging equipment and monitoring points (described below). Initiate data collection.

2. Operate unit for a minimum of 20 minutes. During this period: measure as-found airflow and calibrate to supply plenum static pressure; measure air velocity at supply and return dry-bulb and wet-bulb locations.

3. Mark a 5-minute as-found test period. Also hand-record refrigerant line temperatures and pressures, superheat/subcool, airflow, and condenser entering and exit temperatures.

4. If site is older unit, clean condenser. Run unit for a minimum of 15 minutes, and mark a 5-minute test period.

5. If airflow adjustment is required (airflow >450 cfm/ton or <350 cfm/ton), adjust airflow to as close to 400 cfm/ton as possible. Run unit for a minimum of 15 minutes, and mark a 5-minute test period. Re-measure airflow and air velocity at DB/WB probe locations.

6. If refrigerant charge adjustment is needed (for non-TXV units, use standard superheat table, for TXV use manufacturer's instructions for required subcooling or Lennox approach temperature), adjust refrigerant charge. Run unit for a minimum of 15 minutes, and mark a 5-minute test period.

7. If occupants agree, install monitoring to record unit on/off cycles and indoor temperature and RH.

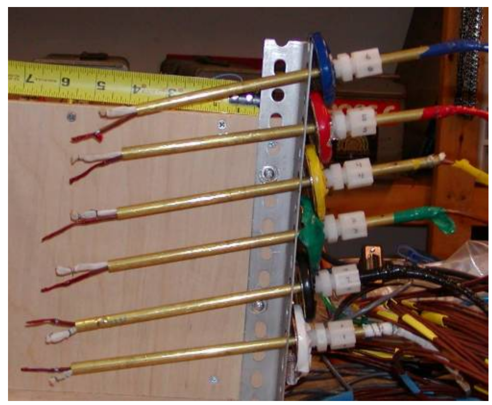
8. Record additional site data, including system information, line-set length, and duct dimensions.

Data collection during testing was accomplished with two custom-built rigs employing Campbell 21X data loggers connected to laptop computers. All sensors (described below) were connected to the data logger, which scanned the channels and recorded (and displayed on the laptop screen) values every 10 seconds under the main execution loop. Data collection was continuous throughout each site visit, but analysis was based on defined 5-minute test periods following a 15-20 minute period of operation after each adjustment.

For airflow measurements, the main execution loop was interrupted (after inserting the flowplate in the filter slot), and the datalogger sequentially recorded 10 pressure readings from the True Flow metering plate and the supply plenum static pressure (over the course of about 30 seconds). Calculation routines built into the data logger programming then calibrated airflow with the supply plenum static pressure, and displayed calculated airflow based on subsequent supply plenum static pressure readings.

The main execution routine was also interrupted to record air velocity data at the DB/WB probe locations using a hot-wire anemometer. In this case, the operator held a button to record the anemometer reading once per second until adequate data were captured.

FIGURE 44, DRY-BULB WET-BULB PROBES USED FOR FOCUS RESEARCH.



Sensors and monitoring points are described in Table 22. Of particular note are the supply- and return-air wet-bulb sensors, which were constructed of brass tubes from which both the dry-bulb and wet-bulb thermocouples protruded by one to two inches. Post-hoc data processing and cross-checks showed significant discrepancies in many cases between latent cooling calculations based on the wet-bulb flows (and airflow) and the recorded condensate production rate. Subsequent bench testing revealed that the wet-bulb depression for the sensors was typically 1 to 2^{o}F high.

Based on this finding, latent cooling output for analysis was calculated from condensate production rates—where these were visually determined to be stable during the test period (87% of tests). In the minority of cases where condensate production was not stable during the test period, the relationship between DB/WB-based cooling and condensate-based latent cooling at other times for the site was used. In a small number of cases (8 of 129 tests) where no condensate data was available, a generic correction to the DB/WB-based values was used.

The post-testing cycling and indoor temperature/RH monitoring for these sites for the remainder of the season was done in the same way as the STAC project field research, using Hobo State loggers to date/time stamp cooling cycles and Hobo temperature/RH loggers to record indoor conditions at the thermostat.

TABLE 22, FOCUS STUDY STANDARD-PROTOCOL MONITORING POINTS AND SENSOR DESCRIPTION.

Outdoor temperature	Radiation-shielded Omega 10K ohm tubular-style thermistor on tripod three feet above ground near compressor
Condenser “radiant” temperature	Average of three Omega 10K ohm thread-mounted thermistors affixed to black metal washers, mounted on three sides of compressor
Refrigerant high- and low-side pressures	Setra # 209, 0 to 500 psi pressure transducers with T-connection from service ports.
Refrigerant high- and low-side temperatures	Omega 10K ohm low-mass adhesive-mount thermistors, (replaced due to durability problems with disk-style thermistors) affixed to refrigerant lines at compressor
Air pressure	Cole Parmer 90080-02 barometer, hand recorded
Supply and return air dry- and wet-bulb temperatures	Custom thermocouple probes in six locations (see Figure []). Four supply locations, two return locations were monitored: typically two supply probes in each of two supply trunk lines and two return probes in the return plenum.
Static pressures—four locations: (1) upstream of filter, (2) blower cabinet, (3) upstream of evaporator coil, and (4) downstream of evaporator coil	Two pressure transducers used in parallel for low and high pressure ranges – Setra #264, 0-0.5” H_2O, and Dwyer #616, 0-2” H_2O with manifold arrangement and control algorithm to cycle among monitoring points (along with a zero reading).
Condensate production	Adam CPWPlus electronic scale, 15 or 35 kg capacity, modified to directly read analog load cells with bucket to collect condensate. Accumulated weight over time recorded.
Compressor and air handler power	Ohio Semitronics model WL50 watt-hour transducers (1 watt-hour per pulse). Two wire wraps through compressor CT and four through the air handler CT to amplify signal.
Compressor and air handler current	Onset 20-Amp CT (0-2.5V output). Two wire wraps through air handler CT to amplify signal.
Airflow	Energy Conservatory True-Flow Air Handler Flow meter (in filter slot) connected to Dwyer and Setra pressure transducers via manifold arrangement (see *Static Pressures* above). 10 readings taken over 30 seconds to calibrate airflow to supply plenum duct static pressure. (The True Flow was only inserted in the filter slot during airflow measurements.)
Air velocity	Sierra Instruments Model 662 hot-wire anemometer, spot measurements at supply and return dry-bulb / wet-bulb monitoring locations.

High-Intensity Protocol

The purpose of the high-intensity protocol was to measure the impact of variation on refrigerant charge and airflow on measured EER. A monitoring set-up similar to that for the standard protocol was used (differences are described below). Once the monitoring was in place, refrigerant charge and airflow were adjusted in as many combinations as could be accommodated within an 8-hour period of time.

The specific charge and airflow levels varied by site, but generally involved testing EER at refrigerant charge levels from 50 percent undercharge to 50 percent overcharge in increments on 10 to 20 percent charge. For each charge adjustment, the amount of refrigerant added or removed was measured with an electronic charge scale. For the final test of each day, the system was tuned as closely as possible to the manufacture's specifications. This tuned condition was used as the basis for determining the charge error for all prior tests.

Airflow was typically adjusted to two or three levels for each site, though not for all charge levels. The goal was to have one set of tests across charge levels at approximately 400 cfm/ton, and obtain additional tests at airflows as far above and/or below this value as could be attained with the existing air handler.

Due to the large number of tests being performed, the run-out time between adjustments was reduced to 4 to five minutes (and the length of each test period was reduced from five minutes to one minute). Tests with longer run-out times indicated this to be an adequate amount of time, except perhaps at very low charge levels when only part of the evaporator coil was being utilized: in that case, residual moisture on the part of the evaporator coil no longer being cooled may cause the amount of sensible cooling to be overestimated.

Monitoring equipment for the high-intensity sites differed somewhat from standard-protocols. The key differences are as follows:

- Static pressures (and airflow measurements from the True Flow device) were measured with an Energy Conservatory Automated Performance Testing (APT) system, and were recorded to a separate laptop at one-second intervals using the Energy Conservatory's Teclog software.

- Electrical power, current, voltage and power factor for the compressor and (separately) the air handler were monitored as 3-second averages using a Dent Instruments Elite Pro data logger.

- Hobo Pro temperature/Rh data loggers were placed in the return plenum to monitor return air conditions, recording 3-second interval data. Hobo Pro data loggers with external probes were also used to monitor dry-bulb temperature in duct take-offs not covered by the four thermocouple probes, as well as monitoring suction line temperature at the evaporator coil outlet.

- Condenser outlet temperature was recorded at two locations using 12-bit temperature probes connected to a Hobo Microstation recording data at one-second intervals (a third condenser outlet temperature was recorded via a thermocouple connected to the outdoor Campbell described below). A pyranometer mounted on a tripod and connected to the microstation also recorded solar radiation.

- A separate (outdoor) Campbell 21x data logger was used to record data on refrigerant line temperatures and pressures, and radiation-shielded outdoor temperature. Three thermocouples connected to this data logger also recorded (unshielded) condenser air inlet temperature on three sides of the condenser.

For the high-intensity sites, return-air humidity was based on the Hobo Pro data rather than the thermocouple DB/WB probes.

Post processing of the data for both the standard and high-intensity protocol sites included the following:

- Calculation of supply- and return-air psychrometrics (relative humidity, absolute humidity enthalpy and density) using dry- and wet-bulb sensor data and hand-recorded air pressure.

- Weighting of individual supply- and return-probe values using duct dimensions and recorded air velocities to obtain estimates of weighted-average overall supply and return air values.

- Adjustment of indicated airflow for air density.

- Calculation of sensible and latent cooling based on supply and return psychrometrics (or condensate production rate) and airflow.

- Calculation of EER based on calculated cooling output and measured electrical power consumption.

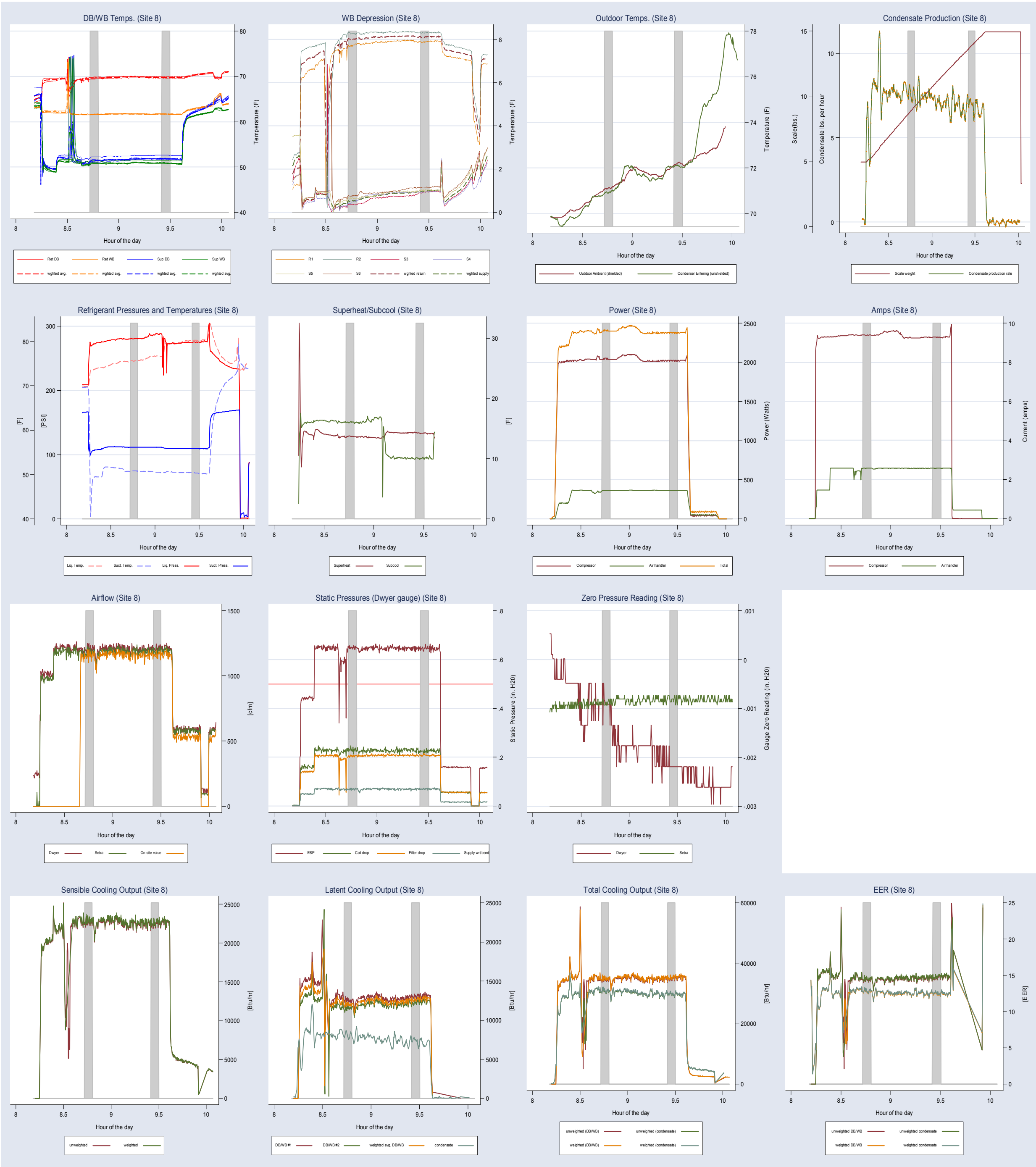
- Calculation of superheat and subcooling values based on recorded refrigerant line temperatures and pressures and recorded refrigerant type.

- Adjustment of recorded static pressures for instrument zero drift over the course of monitoring.

(In addition, it was discovered partway through the fieldwork that the algorithm for cycling through static pressures and True Flow pressure reading was producing erroneous readings for the flowplate, due to inadequate delay time between readings. Later bench testing produced a reliable correction algorithm that could be applied post hoc to the data to correct these errors.)

The data for each site were also visually reviewed for reasonableness and completeness (see Figure 45).

FIGURE 45, SAMPLE DATA PLOTS FOR FOCUS SITE TEST DATA (GRAY SHADING INDICATES TEST PERIODS).



Additional adjustments were needed to account for the fact that outdoor temperature (and hence compressor load and system output) varied from test to test. Compressor power draw increases with increasing temperature at the condenser coil, and hence efficiency drops. To weather-normalize EER for differences in outdoor temperature from test to test a generic adjustment of 0.15 EER per ^{o}F was used, based on analysis of EER and SEER ratings for approximately 600 systems reported in a central air conditioner database maintained at the California Energy Commission.[17] To (separately) normalize compressor power draw to a common outdoor temperature, and adjustment of 0.75 percent per ^{o}F was used, based on analysis of the variation in current draw with outdoor temperature for the STAC two-stage sites.

[17] Available from http://www.energy.ca.gov/appliances/appliance/excel_based_files/Central_Air_Conditioners/. The analysis converted SEER to EER @ 82^{o}F by removing the part-load adjustment to SEER then examined EER @ 82^{o}F to EER @ 90^{o}F to gauge the change in EER per ^{o}F. Note that this analysis was conducted using data available in early November, 2007. Starting in late December, many new SEER 13+ models were added to the database: analysis of the more recent data suggests that a more appropriate EER temperature dependence for new systems is about 0.18 per ^{o}F. Applying this difference to the new systems in the study has only a minor effect on the results, however.

APPENDIX B — PROBABILITY MODELS OF AC USE (2003 AST SURVEY)

TABLE 23, LOGISITIC MODEL OF DAILY AC USE.

Dependent variable: binary indicator of thermostat set to cooling at any time on day question

Number of obs	=	1569
LR chi2(23)	=	202.26
Prob > chi2	=	0.0000
Log likelihood	=	-933.54856
Pseudo R2	=	0.0977

Independent variable	Description	Odds-ratio	Std. error	Z-score	P>|z|	90% confidence interval	
High	high temp (F) for day	1.09	0.01	7.18	0.00	1.07	1.11
high_lag1	high temp (F), prior day	1.06	0.01	4.58	0.00	1.04	1.08
high_lag2	high temp (F), 2 days prior	0.99	0.01	-0.86	0.39	0.97	1.01
high_lag3	high temp (F), 3 days prior	1.06	0.01	4.52	0.00	1.04	1.08
weekend	1 if weekend day	0.99	0.13	-0.09	0.93	0.79	1.23
decade90s	1 if home built 1990 or later	1.22	0.17	1.45	0.15	0.97	1.53
decade80s	1 if home built 1980-1989	0.78	0.15	-1.32	0.19	0.57	1.06
decadepre50s	1 if home built prior to 1950	0.86	0.14	-0.93	0.35	0.65	1.12
bed	# of bedrooms	1.00	0.08	-0.06	0.95	0.88	1.13
townhouse	1 if townhouse	1.42	0.59	0.85	0.40	0.72	2.83
mobilehome	1 if mobile home	1.23	0.46	0.55	0.59	0.66	2.28
multismall	1 if 2-4 unit multifamily	1.62	0.52	1.51	0.13	0.96	2.75
multilarge	1 if 5+ unit multifamily	2.32	0.89	2.21	0.03	1.24	4.35
hhsize1	1 if single-person household	0.54	0.13	-2.65	0.01	0.36	0.79
hhsize2	1 if 2-person household	0.68	0.13	-2.05	0.04	0.50	0.93
hhsize6plus	1 if 6+ person household	1.05	0.42	0.12	0.91	0.54	2.02
kids	1 if kids in household	0.88	0.17	-0.68	0.50	0.64	1.20
seniors	1 if senior(s) in household	1.14	0.17	0.85	0.40	0.89	1.46
own1	1 if home is owned	1.42	0.38	1.31	0.19	0.91	2.22
edulow	1 if < college education level	0.84	0.11	-1.3	0.20	0.68	1.05
eduhigh	1 if advanced college degree	0.77	0.13	-1.49	0.14	0.58	1.03
lowincome	1 if < $20k yearly income	0.70	0.18	-1.4	0.16	0.46	1.06
highincome	1 if $75k+ yearly income	1.38	0.20	2.25	0.02	1.09	1.75

TABLE 24, LOGISITIC MODEL OF HOURLY AC USE.

Dependent variable: binary indicator of thermostat set to cooling at any time on hour in question

Number of obs	=	35827
Log pseudolikelihood	=	-22537.664
Pseudo R2	=	0.0810

(note: standard errors adjusted for clustering on respondent)

Independent variable	Description	Odds-ratio	Std. error	Z-score	P>|z|	90% confidence interval	
high	high temp (F) for day	1.08	0.01	6.95	0.00	1.06	1.10
high_lag1	high temp (F), prior day	1.05	0.01	4.23	0.00	1.03	1.07
high_lag2	high temp (F), 2 days prior	0.99	0.01	-0.62	0.54	0.97	1.01
high_lag3	high temp (F), 3 days prior	1.04	0.01	3.46	0.00	1.02	1.06
hr2	hour = 2 am	1.02	0.01	3.32	0.00	1.01	1.03
hr3	hour = 3 am	1.01	0.01	1.89	0.06	1.00	1.03
hr4	hour = 4 am	1.01	0.01	1.89	0.06	1.00	1.03
hr5	hour = 5 am	1.02	0.01	2.55	0.01	1.01	1.04
hr6	hour = 6 am	1.02	0.01	1.66	0.10	1.00	1.04
hr7	hour = 7 am	1.00	0.02	0.11	0.91	0.98	1.03
hr8	hour = 8 am	1.01	0.02	0.33	0.75	0.98	1.04
hr9	hour = 9 am	1.02	0.02	0.95	0.34	0.99	1.06
hr10	hour = 10 am	1.06	0.03	2.51	0.01	1.02	1.11
hr11	hour = 11 am	1.10	0.03	3.76	0.00	1.06	1.15
hr12	hour = 12 pm	1.14	0.03	4.66	0.00	1.09	1.20
hr13	hour = 1 pm	1.16	0.03	5.04	0.00	1.11	1.22
hr14	hour = 2 pm	1.19	0.04	5.57	0.00	1.13	1.25
hr15	hour = 3 pm	1.23	0.04	6.45	0.00	1.17	1.29
hr16	hour = 4 pm	1.24	0.04	6.52	0.00	1.17	1.30
hr17	hour = 5 pm	1.26	0.04	6.93	0.00	1.19	1.33
hr18	hour = 6 pm	1.24	0.04	6.32	0.00	1.17	1.31
hr19	hour = 7 pm	1.22	0.04	5.96	0.00	1.16	1.30
hr20	hour = 8 pm	1.18	0.04	5.01	0.00	1.12	1.24
hr21	hour = 9 pm	1.09	0.03	3.01	0.00	1.04	1.14
hr22	hour = 10 pm	1.05	0.03	1.67	0.10	1.00	1.10
hr23	hour = 11 pm	1.05	0.03	1.60	0.11	1.00	1.10
weekend	1 if weekend day	0.98	0.12	-0.20	0.84	0.80	1.19
decade90s	1 if home built 1990 or later	1.23	0.15	1.67	0.09	1.00	1.51
decade80s	1 if home built 1980-1989	0.80	0.13	-1.32	0.19	0.61	1.06
decadepre50s	1 if home built prior to 1950	0.84	0.12	-1.20	0.23	0.65	1.07
bed	# of bedrooms	1.05	0.07	0.71	0.48	0.94	1.17
townhouse	1 if townhouse	1.50	0.55	1.11	0.27	0.82	2.75
mobilehome	1 if mobile home	1.23	0.45	0.56	0.57	0.67	2.24
multismall	1 if 2-4 unit multifamily	1.56	0.45	1.53	0.13	0.97	2.51
multilarge	1 if 5+ unit multifamily	1.74	0.60	1.60	0.11	0.98	3.08
hhsize1	1 if single-person household	0.51	0.11	-3.05	0.00	0.35	0.73
hhsize2	1 if 2-person household	0.69	0.12	-2.14	0.03	0.52	0.92
hhsize6plus	1 if 6+ person household	0.95	0.33	-0.15	0.88	0.54	1.69
kids	1 if kids in household	0.80	0.14	-1.24	0.22	0.60	1.08
seniors	1 if senior(s) in household	0.96	0.13	-0.26	0.79	0.77	1.21
own1	1 if home is owned	1.49	0.36	1.65	0.10	1.00	2.22
edulow	1 if < college education level	0.88	0.11	-1.06	0.29	0.72	1.07

eduhigh	1 if advanced college degree	0.71	0.11	-2.17	0.03	0.54	0.92
lowincome	1 if < $20k yearly income	0.76	0.19	-1.13	0.26	0.51	1.14
highincome	1 if $75k+ yearly income	1.34	0.17	2.23	0.03	1.08	1.66

APPENDIX C — MEASURED SYSTEM OPERATING CHARACTERISTICS (2007 FOCUS STANDARD PROTOCOL SITES)

		Measured		Measured per-ton	
	SEER	Mean	Std. dev.	Mean	Std. dev.
Nominal tons	≤10 (n=23)	2.3	0.5		
	13 (n=8)	2.3	0.3		
	14 (n=30)	2.5	0.6		
Outdoor temperature at time of test	≤10	78.4	8.9		
	13	74.7	9.9		
	14	77.9	7.8		
Airflow (cfm)	≤10	861	208	379	73
	13	869	60	391	56
	14	1,007	211	420	96
Air handler watts	≤10	440	174	194	71
	13	437	163	196	71
	14	405	168	173	86
Air handler watts per 1,000 cfm	≤10	503	139	230	88
	13	503	201	226	86
	14	405	160	177	90
Compressor watts (as measured)	≤10	2,181	710	953	237
	13	1,598	288	709	88
	14	1,710	459	692	80
(normalized to 82F)[a]	≤10	2,212	635	975	220
	13	1,676	266	743	56
	14	1,751	434	711	73
(normalized to 95F)[a]	≤10	2,425	705	1,068	242
	13	1,832	292	812	64
	14	1,918	478	778	80
Total watts (as measured)	≤10	2,616	732	1,144	227
	13	2,034	378	905	137
	14	2,121	456	868	93
(normalized to 82F)[a]	≤10	2,648	657	1,168	216
	13	2,113	321	939	93
	14	2,156	439	884	99
(normalized to 95F)[a]	≤10	2,860	723	1,261	237
	13	2,268	348	1,008	101
	14	2,322	480	952	101
Sensible Btu/hr	≤10	13,167	4,261	5,784	1,438
	13	13,672	4,822	5,993	1,805
	14	17,510	4,341	7,189	1,204
Latent Btu/hr	≤10	6,160	3,047	2,679	1,129
	13	6,364	2,293	2,828	962
	14	6,717	2,508	2,720	848

Total Btu/hr	≤10	19,327	6,055	8,462	1,948
	13	20,036	5,939	8,820	2,201
	14	24,227	5,967	9,909	1,473
EER (as measured)	≤10	7.5	2.0		
	13	9.8	2.6		
	14	11.5	2.1		
(normalized to 82F)[b]	≤10	6.9	1.8		
	13	8.7	2.4		
	14	10.9	2.0		
(normalized to 95F)[b]	≤10	5.0	1.8		
	13	6.8	2.4		
	14	9.0	2.0		
				Normalized to 1,000 cfm[c]	
Air handler external static (in. H20)	≤10	0.73	0.21	0.68	0.22
	13	0.75	0.21	0.70	0.20
	14	0.75	0.24	0.75	0.27
Evap. coil pressure drop (in. H20)	≤10	0.24	0.10	0.22	0.10
	13	0.26	0.06	0.25	0.06
	14	0.17	0.12	0.17	0.12
Filter pressure drop (in. H20)	≤10	0.26	0.18	0.24	0.15
	13	0.13	0.06	0.12	0.06
	14	0.28	0.20	0.28	0.20

[a]Based on estimated 0.75% increase in compressor power per 1 F° increase in temperature.

[b]Based on estimated 0.15 decrease in EER per 1 F° increase in temperature.

[c]Based on sqrt(1,000/measured cfm)