Memorandum

To: Jonathon Jackson, AIC, and Jennifer Morris, ICC
From: The Opinion Dynamics Evaluation Team
Date: February 28, 2018
Re: Residential HVAC Metering Study Results

This memorandum presents an overview of the evaluation team's yearlong metering study of Ameren Illinois Company (AIC) 2015 and 2016 HVAC Program participants installing one of the following measures:

- Variable-capacity central air conditioner (CAC) (n=12)
- Variable-capacity central air-source heat pump (ASHP) (n=17)
- High-efficiency blower motor (electronically commutated motor, commonly called simply 'ECM') (n=29)
- Single-speed (typical) CAC with a standard-efficiency permanent split capacity (PSC) fan motor (n=28)

The purpose of the study was to identify any needed updates to the Illinois Statewide Technical Reference Manual Version 6.0 (IL-TRM V6.0) to improve the savings estimation accuracy for these measures:

- Variable-capacity ASHPs (IL-TRM V6.0 Section 5.3.1 – Air Source Heat Pump)
- Variable-capacity CACs (IL-TRM V6.0 Section 5.3.3 – Central Air Conditioning)
- ECM fan motors (IL-TRM V6.0 Section 5.3.5 – Furnace Blower Motor)

Summary

Based on the findings and analysis described in this memo, we recommend the following updates to the TRM.

Table 1. Recommended TRM Changes

Measure	Value	TRM Value	Recommended Update	Reference: IL-TRM V6.0 Page #
All ECMs	Energy: Shoulder Season Savings	51 kWh	Change name to "Circulation Mode Savings". Savings vary with location and system type and are greater than TRM value.	91
	Demand: Shoulder Season Savings	0 kW	0.019 kW	91 & 92
ECM with CAC (CAC does not receive incentive)	Energy: Cooling Savings	263 kWh	$\Delta kWh_{ecm,cool} = EFLH_C \times Capacity_C \times 9.2 \frac{kWh}{MMBtu}$	91
ECM with no CAC	Energy: Cooling Savings	175 kWh	0 kWh	91

Measure	Value	TRM Value	Recommended Update	Reference: IL-TRM V6.0 Page #
ECM with CAC	Energy: Cooling Savings	263 kWh	0 kWh	91
ECM with ASHP (ASHP does not receive incentive)	Energy: Cooling Savings	N/A	$\Delta kWh_{ecm,cool} = EFLH_C \times Capacity_C \times 9.2 \frac{kWh}{MMBtu}$	N/A
ECM with gas furnace	Energy: Heating Savings	418 kWh	$\Delta kWh_{ecm,heat} = heat\ load\ (therms) \times \frac{0.1\ MMbtu}{therm} \times 7.1 \frac{kWh}{MMBtu}$	91
ECM with ASHP or electric furnace	Energy: Heating Savings	N/A	$\Delta kWh_{ecm,heat} = EFLH_H \times Capacity_H \times 4.2 \frac{kWh}{MMBtu}$	N/A
ECM with CAC (CAC does not receive incentive)	Demand: Summer Peak Savings	Varies with EFLH and higher EFLH = lower savings; this is counterintuitive.	$(\Delta kW_{SSP}) = 0.0118 \times cooling\ capacity\ (kBTUh)$	91 & 92
ECM with ASHP (ASHP does not receive incentive)	Demand: PJM Savings	N/A	$(\Delta kW_{PJM}) = 0.0079 \times cooling\ capacity\ (kBTUh)$	N/A
CAC Early Replacement	Baseline SEER	10 SEER or reported value	9.3 SEER	71
	Baseline EER	Algorithm (9.2 EER)	7.5 EER	Page 75, footnote 212, referencing "formula above" (on page 65 in ASHP section)
CAC Time of Sale (TOS)	Baseline EER	11.2 EER	10.5 EER	75
ASHP Early Replacement	Baseline SEER	Replaces ASHP: 9.12 Replaces CAC: 8.6	9.3 SEER	63
ASHP Early Replacement	Baseline EER	Algorithm	7.5 EER	Page 65
ASHP TOS	Baseline EER	11.8 EER	11 EER	66
CAC and ASHP	Installed SEER	AHRI[1] nameplate	$\%\ SEER_{adj} = 0.805 \times \left(\frac{EER_{ee}}{SEER_{ee}}\right) + 0.367$	ASHP: 63 CAC: 74
ASHP	Installed HSPF	AHRI nameplate	$\%\ HSPF_{adj} = \left(\frac{17\ °F\ Capacity}{47\ °F\ Capacity}\right) \times 0.158 + 0.899$	64

Background

The IL-TRM V6.0 Section 5.3.1 estimates savings of ASHPs replacing *in situ* equipment using equations that rely on assumed values for the seasonal energy efficiency ratio (SEER) and heating seasonal performance factor (HSPF) of the baseline equipment. The IL-TRM V6.0 Section 5.3.3 estimates cooling energy savings of CACs in the same way. The metering study provides SEER and energy efficiency ratio (EER) baseline values as well as new equipment SEER adjustments based on Illinois-specific empirical data.

The TRM includes an algorithm to convert SEER to EER used for estimating demand savings. There is no federal standard minimum (baseline) EER value, so the TRM includes a time of sale baseline value (11.8 EER)

[1] AHRI: Air-Conditioning, Heating, and Refrigeration Institute

that is derived from a formula. A footnote in the TRM explains the formula "is appropriate for single speed units only." The HVAC systems in AIC's program have historically included a mix of single-speed, dual-stage, and variable-capacity units.[2] Table 2 lists the percentage and average efficiency of each unit type in AIC's program in 2016. As Table 2 shows, the TRM formula underestimates EER for the single-speed units and overestimates EER for the variable-capacity units.

Table 2. Comparing AIC 2016 ASHP Nameplate EER to TRM-Derived EER Values for Different Compressor Types

System Type	Percent of Total Systems	Actual Average Nameplate SEER	Actual Average Nameplate EER	EER Derived from TRM Formula: $EER = -0.02 \times SEER^2 + 1.12 \times SEER$
Single-speed compressor	46%	16.0	13.0	12.8
Variable-capacity compressor (dual stage)	31%	16.6	12.5	13.1
Variable-capacity compressor (inverter-driven)	23%	18.4	12.4	13.8

A growing consensus[3] in the HVAC industry holds that parameters used to estimate savings (e.g., SEER, EER, and HSPF) may not accurately reflect the actual seasonal efficiency of variable-capacity systems. Relatively new to the residential HVAC market, the actual performance of variable-capacity systems has not been well documented or extensively researched. The actual miles per gallon achieved by a car is a good analogy for the actual performance of a variable-capacity HVAC system. The same car operating under different conditions may achieve different miles-per-gallon rates for various reasons (e.g. frequent stops, high speed operation). For all types of residential CAC and ASHP systems, the performance can be affected by many factors:

- For a cooling season (SEER): performance depends on the indoor conditions and climate in which the system operates.
- For a heating season (HSPF): performance depends on the indoor conditions and the climate in which the system operates (ASHPs only).
- (HSPF): performance depends on the proportion of heat delivered by inefficient backup electric resistance heat to heat delivered by the compressor (ASHPs only).
- (HSPF): performance depends on the energy consumption of defrost mode cycles.

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[2] Variable-capacity systems can change the heating and/or cooling capacity delivered by controlling the flow of refrigerant through the indoor and outdoor refrigerant coils. The most common variable-capacity systems use a multispeed compressor (often called "dual-speed" or "dual-stage" compressor), a dual compressor configuration (two separate compressors), or an AC/DC inverter (often called "inverter-driven" or "variable-speed").

[3] Examples: 1) VCHP [Variable Capacity Heat Pump] Task Group which includes members from EPA, DOE, BPA, PGE, CSA Group, VEIC, numerous Canadian utilities and IESA Canada. This group was set up specifically to develop a new HSPF testing protocol. Members of the group discussed and agreed that the AHRI tests should be updated – especially for systems operating at temperatures below 17F. 2) IEPEC Small Commercial HVAC Working Group (Cadmus, Navigant, DNV). HVAC technical experts from each consultancy discussed and agreed SEER and HSPF should be carefully considered when used in algorithms, especially for northern US region.

- Performance may be impacted by installation issues including insufficient airflow, incorrect refrigerant charge, or duct leakage.

The performance of a variable-capacity HVAC system may be impacted by these additional factors:

- How a homeowner operates a system (e.g., significant changes to thermostat settings); efficiency is variable even as indoor and outdoor conditions remain constant, changing with changes to the compressor speed and capacity output
- The logic of the system's controller
- How the system is sized (e.g., an undersized system will operate at higher average capacity and effectively lower efficiency)

The evaluation team conducted an *in situ* metering study to determine the operational characteristics and energy savings of single-speed CACs, variable-capacity CACs, and variable-capacity ASHPs.

The team also metered the power, runtime, and energy consumption of both "high-efficiency" (ECM) and "standard-efficiency" (PSC) furnace blower motors in all modes of operation (heating, cooling, and circulation mode). A thermostat typically has two modes of operation for the fan: "On" or "Auto" mode. If the thermostat is set to fan "On", the fan runs continuously, circulating air even if the system is not heating or cooling. Fan "On" mode is also referred to as "continuous" or "circulation" mode. If the fan is set to "Auto" mode, the fan only runs when the heating or cooling system operates. Compared to ECM fans, standard-efficiency fans use significantly more power when operating in circulation mode.

The TRM savings protocol for ECM fans is based on data from a Wisconsin field study[4] (conducted in 2002) and survey[5] (conducted in 2004). The Wisconsin survey found some homeowners who switched from a standard-efficiency to an ECM fan, changed how they operated their fan, operating the ECM in circulation mode more than the previous, standard-efficiency fan. The Illinois TRM explains such a behavioral change can result in low or negative savings. The team did not meter baseline (pre-measure) fan use to determine savings. Rather, the team interviewed ECM metering participants to identify instances of behavioral change that increase fan runtime and used this information to adjust the circulation mode savings estimate. The interview guide used to explore this issue can be found in Appendix C.

Findings and Recommendations

This section summarizes the team's findings and recommendations for improving the TRM and for AIC's consideration when planning its residential Heating and Cooling Program measures.

The team's research revealed information about ECMs and variable-capacity HVAC systems that may be used to improve the assumptions in the TRM. This section includes three recommendations related to the High Efficiency Blower Measure (henceforth the ECM measure) and four recommendations related to the ASHP and

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[4] Energy Center of Wisconsin. *Residential Programs Electricity Use by New Furnaces: A Wisconsin Field Study.* Prepared for State of Wisconsin Department of Administration Division of Energy. October 2003. https://www.proctoreng.com/dnld/WIDOE2013.pdf

[5] Glacier Consulting Group, LLC. *Focus on Energy Evaluation: ECM Furnace Impact Assessment Report.* Prepared for the State of Wisconsin Public Service Commission of Wisconsin. Final Report: January 12, 2009. https://www.focusonenergy.com/sites/default/files/emcfurnaceimpactassessment_evaluationreport.pdf

CAC measures in the TRM. To explicitly follow the team's recommendations, one would need to identify the following for HVAC measures:

- Whether an ECM is included with the installation of a heating or cooling system
- CAC compressor type: single-speed vs variable-capacity
- ASHP compressor type: single-speed vs variable-capacity
- ASHP AHRI heating capacity at 47°F and 17°F
- ASHP and CAC AHRI cooling capacity
- Gas furnace valve type: single-stage vs dual-stage or modulating

The appendix describes methods to determine the compressor type and gas furnace valve type (see Identifying Variable-Capacity Systems in Appendix A of this document). AIC should record whether an ECM is included as part of an ASHP installation. The impacts may warrant this additional effort. For example, many ASHPs installed through AIC's program have an ECM fan, yet savings in circulation mode are not claimed. Appendix B includes the team's recommended TRM measure updates, incorporating the findings from this study.

ECMs

Finding 1: Both standard-efficiency and ECM systems operated in circulation mode (i.e., the fan circulates air while the system is not heating or cooling). The TRM names the savings in this mode "Shoulder Season Savings" and cautions that savings could be negative if a "resident runs the [ECM blower] continuously because it is a more efficient motor and [the resident] would not run a non-[ECM blower] in that way". While the TRM classification of shoulder season savings implies that circulation mode occurs during the shoulder seasons, meter data showed fans operated in circulation mode throughout the year. Meter data also showed that various homeowners operated their HVAC fan in circulation mode continuously, occasionally, or never. Table 3 summarizes the circulation mode usage as observed from the meter data for both standard-efficiency and ECM fan types. The results listed in the table indicate circulation mode occurs throughout the year, not just during the shoulder seasons. The table also shows a significantly higher proportion of ECM fans had high circulation mode runtimes.

Table 3. Summary of Circulation Mode Use

System Type	% of Units by Type of Circulation Mode Usage*			% of Operation During Each Season		
	High Usage	Used Occasionally	Rarely or Never Used	Heating Season (Dec-March)	Cooling Season (June – Sept)	Shoulder Seasons (Oct-Nov) (April-May)
Standard-Efficiency Furnace Fan	8%	21%	71%	21%	11%	18%
ECM Furnace Fan	45%	31%	24%	40%	34%	38%

* For the possible circulation mode hours (when system is not heating or cooling):
High Usage: A fan operating more than 60% of the possible circulation mode hours
Used Occasionally: Operating 15% to 60% of the time
Rarely or Never: Operating less than 15% of the time

Table 4 shows significant differences in runtime and average power when comparing standard-efficiency and ECM fan meter data.

Table 4. Summary of Metered Results: Circulation Mode

System Type	Average Circulation Mode Runtime (Hours)	Average Hours During Metering Period*	% Circulation Mode Usage	Average Circulation Mode Power (kW)
Standard-Efficiency Furnace Fan	1,195	7,028	17%	0.470
ECM Furnace Fan	2,162	5,957	36%	0.114

*Time that circulation mode is possible (i.e. does not include hours the system is in heating or cooling mode)

Assuming equivalent baseline (standard-efficiency) and efficient (ECM) fan annual circulation mode hours, one may estimate savings by multiplying the power difference (0.356 kW) by total annual hours in circulation mode. However, as Table 4 indicates, circulation mode hours were quite different, with ECM fans running longer. Some studies (Wisconsin ECM Furnace Study[6]) found participants who installed ECM fans changed their behavior and chose to use the new fan more than their previous. If this occurs, ECM savings are:

$$Baseline\ Hours_{circ}\ \times\ 0.356\ kW - behavior\ change\ penalty$$

The behavior change penalty is the energy consumption due to additional hours of operation of the ECM fan:

$$(ECM\ Hours_{circ} - \ Baseline\ Hours_{circ})\ \times\ 0.114\ kW$$

We acknowledge the possibility of a *change* in behavior but also recognize the possibility of motivational *differences* in behavior between the standard-efficiency and ECM groups in this study. Circulation mode savings can be significant. Therefore, one might expect a homeowner who prefers to run their fan in circulation mode to have higher motivation to install an ECM than one who never runs the fan in circulation mode. Consequently, the team concluded that neither the standard-efficiency runtime (17%) nor ECM circulation mode runtime (36%) could be used to determine ECM savings. This study did not monitor fan runtime prior to installation of an ECM, so the team surveyed the ECM metering study participants to determine behavioral changes, specifically in circulation mode. Table 5 shows results of survey respondents circulation run time and whether or not they had changed behavior from before the ECM purchase. Since the 20 customers out of 29 that responded to the survey had very close to the same average circulation time (37% vs. 36%), it appears the sample is a good representation of the ECM participants.

[6] Glacier Consulting Group, LLC. *Focus on Energy Evaluation: ECM Furnace Impact Assessment Report.* Prepared for the State of Wisconsin Public Service Commission of Wisconsin. Final Report: January 12, 2009.
https://www.focusonenergy.com/sites/default/files/emcfurnaceimpactassessment_evaluationreport.pdf

Table 5. Baseline Circulation Mode Hours

ECM Participant Group	n	Run Time in Circulation Mode
Survey Result: Change in behavior	6	68% (See next table for additional details)
Survey Result: No change in behavior	14	23%
Baseline Survey Completed	**20**	**37%**
Total ECM Sample	**29**	**36%**

Fourteen of 20 (70%) of surveyed participants said they did not change the way they operate their new fan. The circulation mode runtime for these participants (average of 23%) represents the assumed operating time of both the previous and new systems, since the higher use for the other six surveyed participants who changed behavior is accounted for in the penalty calculation. Table 6 summarizes the team's findings for the ECM participants who increased the fan runtime in circulation mode.

Table 6. ECM Participants Who Changed Behavior

Metered Circulation Mode Runtime	Did Previous System Have ECM?	Did circulation mode hours increase after installing ECM?	How did you operate previous fan in circulation mode?	Reason(s) for Increasing fan use
85%	Don't know	Yes	Did not use	Bought an air filtration system, filters the air in winter, too much humidity, so doesn't use fan in summer. Contractor/installer did say fan costs only pennies to run
17%	No	Yes	Did not use	Personal choice (not a recommendation from contractor)
73%	No	Yes	Did not use	Contractor suggestion and wanted to filter air
86%	No	Yes	"Rarely"	Contractor suggestion and wanted to filter air for asthma/allergies
96%	No	Yes	Did not use	Personal choice (not a recommendation from contractor)
54%	No	Yes	Occasionally used to circulate and filter air when smoking indoors	Contractor recommendation

Four of the six participants indicated they did not operate their previous fan in circulation mode while the other two may have some baseline circulation mode hours of operation. Since hours are unknown but likely low, the team assumed the baseline hours of operation in circulation mode for these participants was zero. The resulting penalty is:

$$30\%\ of\ ECM\ participants\ \times\ 68\%\ Hours_{circ}\ \times\ 0.114\ kW = \mathbf{0.02326\ kW}\ \times\ Possible\ Circulation\ Mode\ Hours$$

The heating and cooling hours in the TRM vary with location, so the total hours a fan can operate in circulation mode also varies with location. The TRM includes location-specific heating and cooling equivalent full-load hour (EFLH)[7] values. Table 7 lists the TRM values, including the home heating values that represent the annual

[7] EFLH are the number of hours a unit would have to operate at full capacity to equal the amount of heating or cooling by the system during a typical year. If a unit operates below its full (rated) capacity, its actual runtime hours would be higher than its EFLH.

heat load (in therms) of a home with a gas furnace.[8] To estimate EFLH based on the therm heat load values, the team determined the average full-load capacity of gas furnaces from AIC's 2015 program tracking data.

Table 7. TRM Full-Load Hour and Home Heat Load Values for Single-Family Homes

Location	EFLH Cooling	EFLH Heating (ASHPs Only)	Gas Furnace Heating Therms	Approximate* Gas Furnace EFLH
Rockford	512	1,969	873	1,166
Chicago	570	1,840	834	1,113
Springfield	730	1,754	714	953
Belleville	1035	1,266	551	736
Marion	629	1,821	561	749

*The TRM includes annual heat load (in therms). It does not include an EFLH value. The team used AIC's tracking data to estimate the average full-load capacity of furnaces in Illinois (74,900 Btuh output) to approximate EFLH values.

The team compared the TRM values to the heating and cooling EFLH determined through metering. The difference between the metered estimates and TRM values was not statistically significant, so the team is not proposing any adjustments to the values in the TRM. However, actual heating and cooling runtime for HVAC systems with variable capacity is inherently different from the heating and cooling EFLH, so the team calculated adjustment factors to estimate actual heating and cooling mode runtime from the TRM EFLH values (See Table 8).

[8] Heat load is equal to annual heating therm consumption multiplied by furnace efficiency.

Table 8. Estimating Runtime from EFLH

System Type	TRM Parameter	TRM Parameter Adjustment Factor	Source	Explanation
Single-Speed CAC	$EFLH_C$	0.74	Meter data analysis	EFLH are based on nameplate peak (95F°) capacity and total capacity delivered in a season. Capacity of single-speed systems decreases as temperature increases. As expected, runtime is less than EFLH. Note: When EFLH are calculated from total seasonal kWh and peak kW, runtime would typically be greater than EFLH.
Single-Speed ASHP	$EFLH_H$	1.24	Not observed	Though not metered in this study, we assume single-speed ASHP heating mode runtime is similar to variable-capacity ASHPs. Although single-speed ASHPs have lower capacity at colder temperatures—which would require longer runtime, the use of backup electric strip heat would effectively decrease runtime.
Single-Stage Gas Furnace	Gas Furnace Heating Load (therms)	1.34	Derived from AIC average furnace output capacity	Therm to Btu conversion based on average nameplate capacity (74,900 Btuh): 1.34 = 100,000 Btu/therm / 74,900 Btuh
Variable-Capacity Gas Furnace	Gas Furnace Heating Load (therms)	2.1	Meter data analysis	The team found the actual average Btuh output capacity for variable-capacity furnaces was 49,500 Btuh. The full-load (high-stage) nameplate capacity of these furnaces was 78,900 Btuh, meaning the furnaces ran at lower capacity for more hours than an equivalent single-stage furnace. 78,900/49,500 x 1.34 conversion = 2.1
Variable-Capacity CAC and ASHP	$EFLH_C$	1.2	Meter data analysis	
Variable-Capacity ASHP	$EFLH_H$	1.24	Meter data analysis	

HVAC system heating and cooling runtime and a homeowner's decision to operate a fan in circulation mode affect the fan runtime and potential energy savings. The power difference between a standard-efficiency and ECM fan in each mode of operation (heating, cooling, circulation) is unique, so the team used meter study observations to develop a new TRM method to calculate both the power difference and operating hours to determine ECM energy savings.

Recommendation 1: The team recommends using the assumption that a fan operates in circulation mode for 23% of time when a system is neither heating nor cooling. To estimate circulation mode savings for any ECM fan installation, use the following algorithms with parameters listed in Table 4 through Table 8. A savings penalty corrects for the increase in energy use due to the runtime increase that the team determined through participant interviews.

For example, for a single-speed CAC with variable-capacity gas furnace with ECM in Belleville, Illinois, first estimate the total non-heating and non-cooling hours and actual circulation mode runtime as shown:

$$hours_{no\ heat,cool} = 8{,}760 - cooling\ runtime - heating\ runtime$$

$$hours_{no\ heat,cool} = 8{,}760 - 1{,}035 \times 0.74 - 551 \times 2.1 = 6{,}837\ hours$$

6,837 hours is the total number of possible circulation mode hours for this example. We then calculate the total circulation mode runtime:

$$hours_{circ} = \%\ circulation\ mode \times hours_{no\ heat,cool} = 23\% \times 6{,}837 = 1{,}573\ hours$$

Use the following algorithm to estimate ECM circulation mode energy savings (without penalty):

$$\Delta kWh_{ecm,circ} = 0.356\ kW \times hours_{circ}$$

$$\Delta kWh_{ecm,circ} = 0.356\ kW \times 1{,}573 = 560\ kWh$$

Note: 0.356 kW is the difference in the circulation mode power of standard-efficiency and ECM fans (See Table 4). Finally, use the following algorithm to estimate the ECM circulation mode energy savings penalty due to increased hours of use:

$$\Delta kWh_{ecm,circ,penalty} = 0.02326\ kW \times 6{,}837 = -159\ kWh$$

$$\Delta kWh_{ecm,circ} = 560\ kWh - 159\ kWh = 401\ kWh$$

To estimate peak demand savings from ECMs operating in circulation mode, the team recommends using the coincidence factor assumptions in the TRM. A CAC with peak demand coincidence factor of 68% would have ECM peak demand savings of approximately:

$$\Delta kW_{ecm,circ} = \frac{(1 - CF) \times \Delta kWh_{ecm,circ}}{hours_{no\ heat,cool}}$$

Using the example above:

$$\Delta kW_{ecm,circ} = \frac{(1 - 0.68) \times 401}{6{,}837} = 0.019\ kW$$

Finding 2: The team calculated heating and cooling capacity delivered by each system and recorded the energy consumption of the furnace fan motor. The values in Table 9 represent the metered average fan energy consumption normalized by the heating and cooling capacity delivered by the system, and are described in terms of kWh per MMBtu of delivered capacity. The team chose to use this parameter rather than runtime and average power because the system capacity (HVAC unit size) impacts fan energy use, and because runtimes for variable-capacity systems are different from single-speed systems. This parameter simplifies the energy-savings calculations.

Table 9. Average Fan Energy Consumption per MMBtu

System Type	Fan in Cooling Mode kWh/MMBtu	Fan in Heating Mode kWh/MMBtu
Standard-Efficiency Fan	20.1	13.8**
ECM Fan (CAC)	10.9*	6.7
ECM Fan (ASHP)		9.6
Difference	**9.2**	**CAC w/ Furnace: 7.1** **ASHP: 4.2**

*CAC kWh/MMBTU was 11.9 and ASHP kWh/MMBTU was 10.4 but the difference between these results was not statistically significant at the 90% confidence interval.

**The team did not meter standard-efficiency fans in baseline ASHP systems. Therefore, this estimate is conservative because a gas furnace can achieve a significantly higher temperature differential, effectively delivering a higher rate of capacity than an ASHP.

An ECM fan installation generates savings when installed as a new component of an existing CAC or ASHP system. (For measure description, see section 5.3.5 in TRM). The TRM Furnace Blower Motor measure does not include key criteria for common installations that do not save energy. The TRM savings algorithm is not appropriate for ECMs installed with new CACs or ASHPs because the efficiency (SEER, EER, and HSPF) of these systems typically already includes the gain in efficiency from the ECM furnace fan. Consequently, an ECM installed in AIC's ASHP retrofit program does not generate any additional heating or cooling savings.

Recommendation 2: The TRM furnace fan measure should include the kWh/MMBtu values in Table 9. MMBtu is the location- and measure-specific heating and cooling load of a home. This value is estimated from nameplate capacity and TRM parameter values (EFLH and gas furnace heat load, see Table 7). The values in Table 9 should be used to estimate ECM heating and cooling mode savings, which vary with the heating and cooling requirements of a home.

For an ECM fan installed with a CAC and furnace, estimate energy savings with the following algorithms:

$$\Delta kWh_{ecm,cool} = EFLH_C \times Capacity_C \times 9.2 \frac{kWh}{MMBtu}$$

$$\Delta kWh_{ecm,heat} = heat\ load\ (therms) \times \frac{0.1\ MMbtu}{therm} \times 7.1 \frac{kWh}{MMBtu}$$

For an ECM fan installed with an ASHP or electric furnace, use these algorithms:

$$\Delta kWh_{ecm,cool} = EFLH_C \times Capacity_C \times 9.2 \frac{kWh}{MMBtu}$$

$$\Delta kWh_{ecm,heat} = EFLH_H \times Capacity_H \times 4.2 \frac{kWh}{MMBtu}$$

The TRM should clearly explain the type(s) of savings realized from the most common ECM installations, and should explain the scenarios that do not generate savings. Table 10 lists these scenarios with the applicable equation number from the list above in parentheses. The equations to estimate circulation mode peak demand savings are described in Recommendation 1 above. The cooling-related peak demand savings equations are described in Finding 3 below.

Table 10. Summary of ECM Savings

Installation Type	Claim Heating Energy Savings?	Claim Cooling Energy Savings?	Claim Circulation Mode Energy Savings?	Claim Circulation Mode Peak Demand Savings?	Claim Cooling-Related Peak Demand Savings?
ECM with ASHP installed through program and ASHP savings calculated using HSPF and SEER	No	No	Yes	Yes	No
ECM with CAC installed through program and CAC savings calculated using SEER	Yes (2)	No	Yes	Yes	No
ECM with non-qualifying or existing CAC	Yes (2)	Yes (1)	Yes	Yes	Yes
ECM with non-qualifying or existing ASHP or electric furnace	Yes (4)	Yes (3)	Yes	Yes	Yes
ECM with any furnace, no cooling system	Yes (2)	No	Yes	Yes	No

Note: The team metered ECM power and energy consumption in high-efficiency gas furnaces. All furnaces had either dual-stage or modulating burners. We did not meter ECM power and energy consumption in single-stage gas furnaces; such a combination in AIC's program was uncommon. The recommended savings presume the baseline is a standard-efficiency fan with one cooling and one heating fan speed setting.

Finding 3: The demand savings algorithm in the TRM does not produce consistent, practical values. There are two issues with the TRM demand savings methodology for ECMs:

- The TRM $EFLH_C$ vary by location, but ECM cooling mode savings (263 kWh) do not. One would expect ECM cooling use and savings to correlate with either cooling EFLH or system size.
- The ECM demand savings algorithm divides the cooling mode savings value (263 kWh for all locations) by the location-specific $EFLH_C$ value. This means that locations with lower seasonal cooling use have higher ECM demand savings, as shown in the following example:

$$Belleville\colon \frac{263\ kWh\ saved\ by\ ECM\ in\ cooling\ season}{1{,}035\ EFLH} = 0.254\ kW$$

$$Rockford\colon \frac{263\ kWh\ saved\ by\ ECM\ in\ cooling\ season}{512\ EFLH} = 0.514\ kW$$

Comparing hourly energy use of standard-efficiency and ECM fans for the peak hour and for the PJM hours of interest, the team found a summer system peak demand difference of 0.0118 kW per kBTUh of cooling capacity and a summer PJM peak demand difference of 0.0079 kW per kBTUh of cooling capacity. These savings estimates include the interactive effects of waste heat reduction for a system operating at 11 EER.

Recommendation 3: Update the TRM algorithm to incorporate the peak demand savings values that the team found. Remove the demand savings algorithms and replace with the following equations:

$$summer\ system\ peak\ demand\ savings\ (\Delta kW_{SSP}) = 0.0118\ \times\ cooling\ capacity\ (kBTUh)$$

$$PJM\ summer\ system\ peak\ demand\ savings\ (\Delta kW_{PJM}) = 0.0079\ \times\ cooling\ capacity\ (kBTUh)$$

Total summer peak demand savings for an ECM installed into an existing or non-program qualifying 36,000 Btuh CAC with gas furnace in Belleville, Illinois are estimated as follows:

$$\Delta kW_{SSP} = \Delta kW_{cool} + \Delta kW_{ecm,circ}$$

$$\Delta kW_{SSP} = 0.425 + 0.019 = 0.444\ kW$$

Variable-Capacity Air-Source Heat Pumps and Central Air Conditioners

Finding 4: The team found several minor issues with the eligibility requirements of AIC's early retirement ASHP program and with the TRM efficiency assumptions for early retirement CACs and ASHPs.

To be eligible for AIC's ASHP early retirement incentive, an ASHP must replace a working CAC or ASHP of 10 SEER or lower efficiency. The deemed remaining lifetime of an existing system in the TRM is 12 years. The federal minimum efficiency standard in 2006 was 13 SEER, so beginning in 2018, AIC's measure eligibility precludes 12-year old systems.

According to the early retirement ASHP measure in the TRM, the assumed baseline of an ASHP replacing an ASHP is 9.12 SEER, and 8.6 SEER if an ASHP replaces a CAC. The early retirement CAC measure baseline is 10 SEER. The team reviewed the data sources and conducted research to update the baseline SEER values. Using the assumptions and data sources listed in Table 11 and PY8 and PY9 tracking data, the team independently estimated a SEER value for an early retirement system.

Table 11. Assumptions and Data Sources for Early Retirement Baseline Efficiency Estimate

Assumption	Data Source
Annual historic ASHP and CAC shipment volume estimates	http://www.ahrinet.org/Resources/Statistics/Historical-Data/Central-Air-Conditioners-and-Air-Source-Heat-Pumps.aspx
Historic ENERGY STAR (high efficiency) system proportional estimates	https://www.energystar.gov/ia/partners/downloads/unit_shipment_data/2015_USD_Summary_Report.pdf?52f9-67a6
A reasonable and conservative estimate for efficiency degradation of a typical residential HVAC system is 1% annually.	1-3%: https://www.nrel.gov/docs/fy06osti/38238.pdf 1.1%:http://www.lincusenergy.com/blog/2012/01/hvac-cooling-and-heating-efficiency-degradation/ 1.4%:https://www.efis.psc.mo.gov/mpsc/commoncomponents/viewdocument.asp?DocId=935842419
Mortality curve for residential appliances is a reasonable approach to determine the yearly annual share of functional units	http://aceee.org/files/proceedings/2010/data/papers/1977.pdf
A reasonable range in age of replaced units is 10 to 21 years.	See analysis file: "Res_HVAC_Stock_Efficiency_Illinois2018.xlsx" Note the average weighted age calculated from the annual probability of mortality for this age range is 16.8 years.

Though the proportion of higher efficiency (ENERGY STAR share) was slightly higher for ASHPs than for CACs, information prior to 2004 was unavailable, so the team determined a single SEER value for both system types. Figure 1 shows the team's 2018 program year SEER and HSPF estimates for CACs and ASHPs. The SEER estimates include an efficiency degradation factor of 1% per year. The team used the TRM algorithm[9] to estimate HSPF from SEER, but imposed restrictions in degradation because HSPF includes the efficiency of both the heat pump outdoor unit and supplemental electric resistance heat and electric resistance heating efficiency does not degrade over time.

[9] TRM formulas: HSPF = SEER x 0.596.

Figure 1. 2018 Average Market SEER and HSPF Estimates by Year of Installation

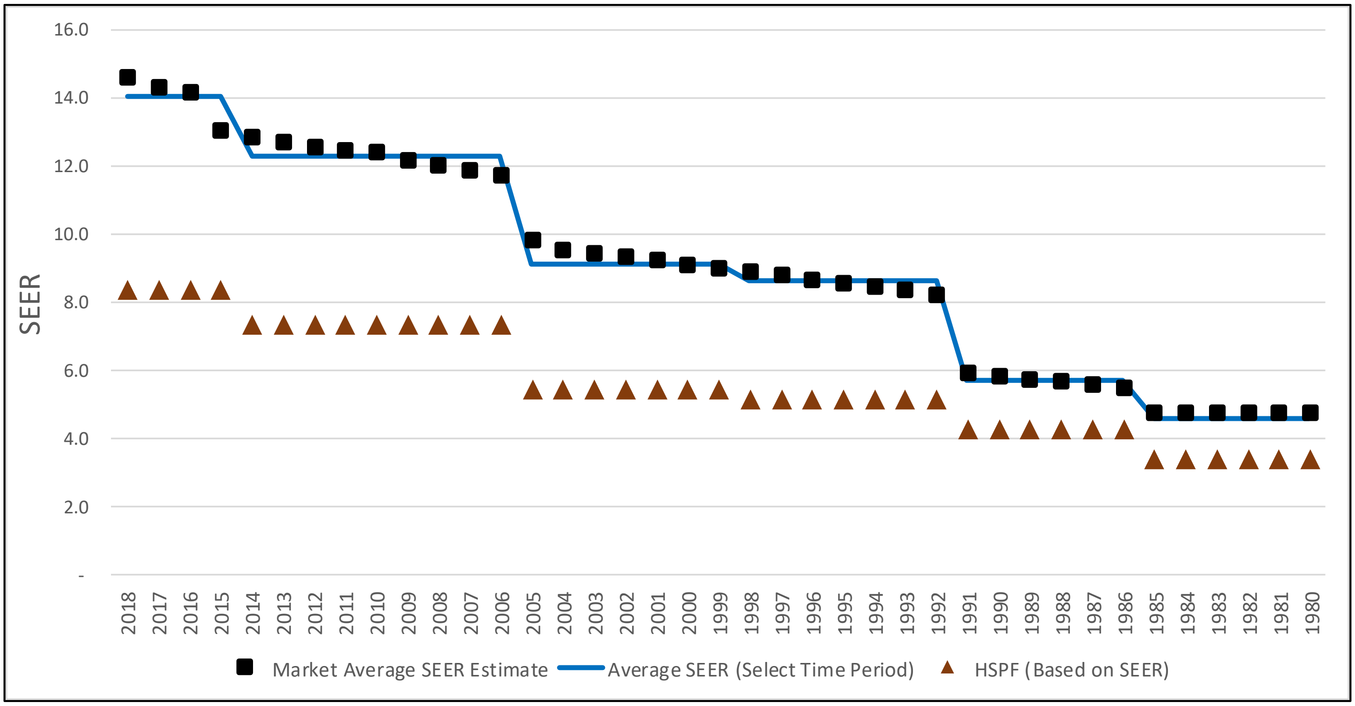

Table 12 lists average efficiency values for time periods with unique minimum SEER requirements. The estimates for "unknown age" are based on the team's mortality curve analysis, which estimates the expected operating efficiency of units that are eligible for replacement (see Table 11).

Table 12. Average Efficiency Estimates by Installation Year

Year of Installation	SEER Estimate	EER Estimate	HSPF Estimate
2015 or later	14.04	11.37	8.37
2006 - 2014	12.32	9.98	7.34
1999 - 2005	9.14	7.41	5.45
1992 - 1998	8.64	6.99	5.15
1986 - 1991	5.72	4.63	4.28
1978 - 1985	4.59	3.72	3.41
Unknown Age	**9.30**	**7.50**	**5.54**

Recommendation 4a: The CAC and ASHP TRM measures should have a consistent baseline SEER value for early replacement systems. The team recommends 9.3 SEER, 7.5 EER, and 5.54 HSPF[10] for program year[11] 2018 if unit age and SEER are unknown. If unit age is known or can be approximated within a few years, but SEER is unknown, the efficiency estimates in Table 12 should be used. If nameplate efficiency and system age are known, an efficiency degradation factor of 1% per year should be applied. The team recommends removing the SEER to EER formula in the TRM. This formula is especially problematic for variable-speed systems (see Table 2).

Recommendation 4b: AIC should consider changing the measure eligibility requirement (currently 10 SEER) and instead use an age requirement (e.g., 10–12 years or greater). While SEER of existing equipment is difficult or impossible to determine, system age is usually verifiable because it may be determined from the condenser serial number. HVAC contractors should report the serial number of each retired condenser and the condenser's age.

Finding 5: The EER baseline value in the TRM for time of sale CACs and ASHPs (11.8 EER) is too high. The team's review of 13 SEER CACs and 14 SEER ASHPs in the AHRI database revealed efficiency as low as 9.0 EER for both system types. More than 5% of the 16+ SEER ASHPs in 2016 had EER of 10.75 or less, and just more than 12% of ASHPs had EER's lower than 11.8, the TRM's recommended baseline efficiency. There is no federal standard EER in Illinois.

Recommendation 5a: Although the team found EER of 9.0 for some systems, this is not necessarily an appropriate baseline value because 13/14 SEER systems with 9 EER typically have variable-speed compressors (e.g., see AHRI Certificate # 10461080). Presumably, the manufacturer sacrifices efficiency for increased peak capacity. Variable-capacity systems are more expensive than single-speed systems, and relatively low SEER, variable-capacity systems are not frequently sold. The team reviewed active AHRI certificates for 13 SEER CACs and found a large quantity (3,000 tested combinations) had efficiency of 10.5 EER. The team performed the same review of federal minimum efficiency ASHPs (14 SEER, 8.2 HSPF) and found the highest quantity had efficiency of 11 EER. Future evaluation activities should include market research to determine an appropriate time of sale EER baseline value for 13 SEER CACs and 14 SEER ASHPs. Based on the team's review of active AHRI certificates, we recommend the following changes to TRM baseline EER values:

- CAC Time of Sale (TOS): change current value (11.0 EER) to 10.5 EER
- CAC early replacement: change current value (estimated with algorithm) to 7.5 EER
- ASHP TOS: change current value (11.8 EER) to 11.0 EER
- ASHP early replacement: change current value (estimated with algorithm) to 7.5 EER

Recommendation 5b: To increase peak demand savings, AIC should consider adding a minimum EER requirement to the new ASHP measure eligibility criteria. The minimum EER value should be greater than the

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[10] 7.5 EER estimated from 9.3 SEER by the ratio of 10.5 EER to 13 SEER for a baseline CAC unit.

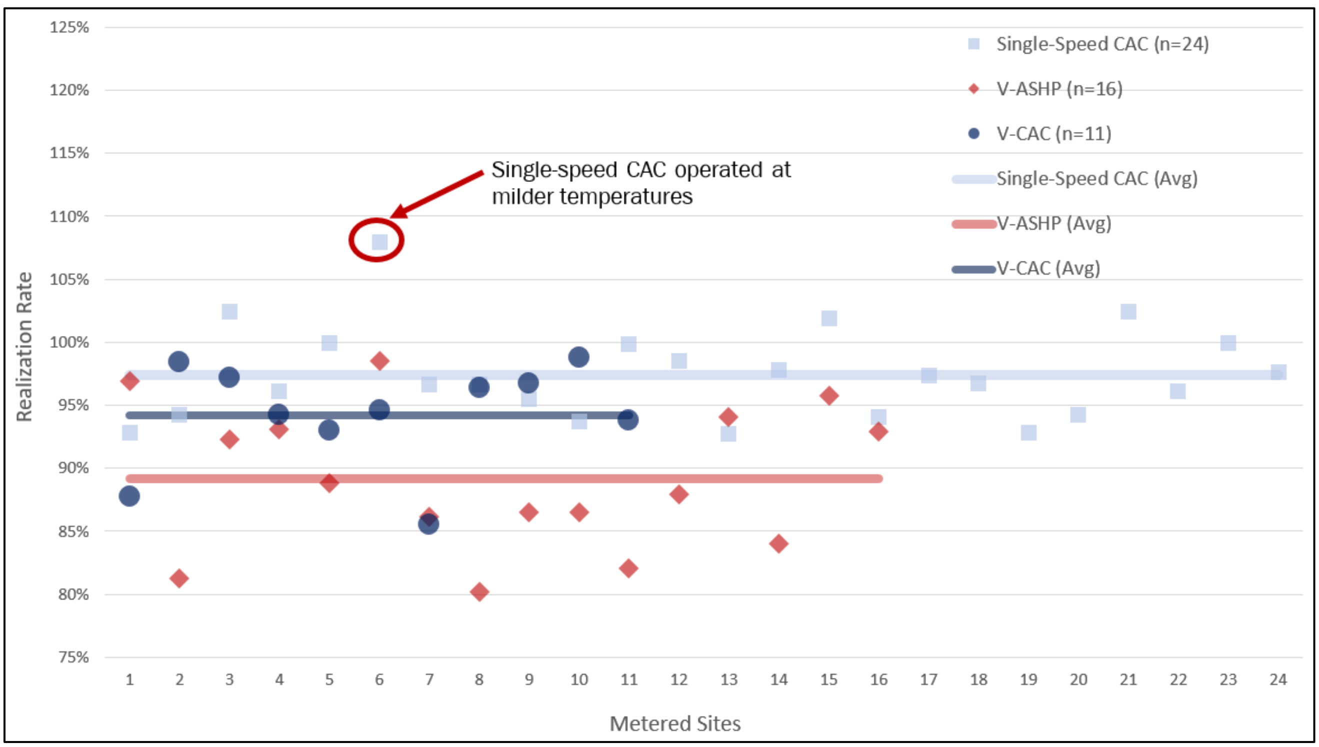
[11] Program year 2019 would increase to 9.5 SEER, and 2020 would increase to 9.7 SEER.

time of sale EER value in the TRM (subject to change per Recommendation 5a above). The team suggests the minimum value required for CACs and ASHPs to achieve ENERGY STAR certification: 12.5 EER[12].

Finding 6: Appendix A in this document describes the methodology that the team used to determine operating efficiency of variable-capacity systems based on equivalent operation of a single-speed system. Relative to single-speed systems operating in cooling mode, the team found that variable-speed systems operate less efficiently.

The light-blue points in Figure 2 represent each single-speed CAC system that the team metered. Ranging from 92% to 108% of nameplate efficiency (14.5 SEER), the single-speed CACs operated at an average of 98% of the AHRI Region IV nameplate SEER value. This indicates that the nameplate SEER of single-speed systems provides a reasonable estimate of the actual performance in Illinois.[13] Figure 2 shows that the variable-capacity CAC and variable-capacity ASHP systems operated below their nameplate SEER values (93% and 89% respectively). The difference between these is not statistically significant at the 90% confidence interval. The average of all variable-capacity systems was 91% of nameplate SEER.

Figure 2. Comparing Expected SEER to Actual SEER for Each System Type



[12] https://www.energystar.gov/products/heating_cooling/heat_pumps_air_source/key_product_criteria

[13] Illinois is in climactic Region IV in the United States.

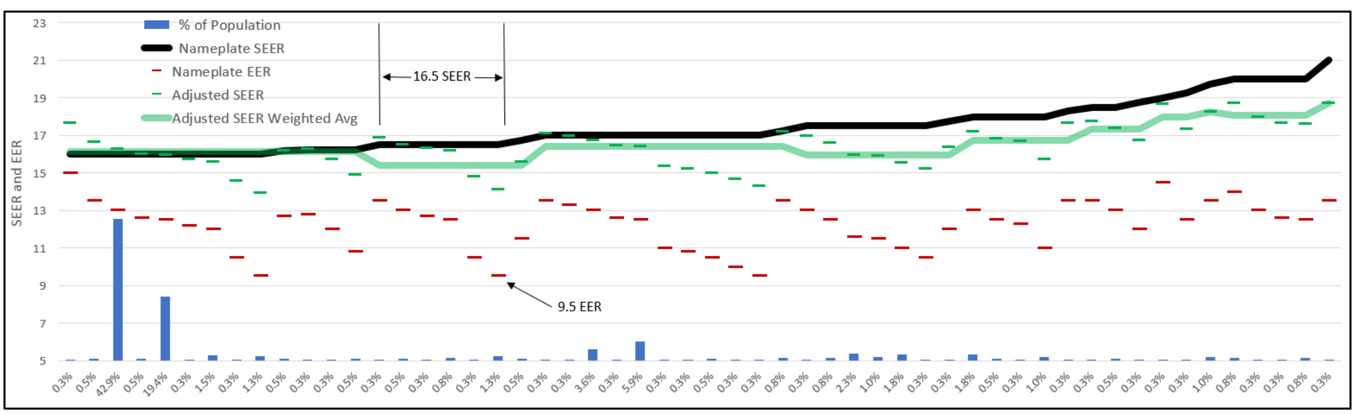
Two variable-capacity systems had lower-than-expected SEER due to improper control. In each case, the compressor operated at a fixed speed in cooling mode. Though the team was unable to fully investigate, the reason one system operated at fixed speed seemed obvious—the indoor furnace was not replaced. All other systems observed by the team were installed with new furnaces that included an integrated controls board.

To calculate energy savings for variable-capacity systems, one must first determine whether the system has a variable-speed compressor, and then multiply the nameplate SEER value by the recommended adjustment of 91%. As the ratio of EER to SEER for a high-efficiency unit decreases below 0.8, the adjusted nameplate SEER value decreases. The team found a relationship between the ratio of EER and SEER and the actual operating efficiency of a variable-capacity system in the cooling season. The following algorithm approximates this relationship and can be used without need to determine system type:

$$\% \, SEER_{adj} = 0.805 \times \left(\frac{EER_{ee}}{SEER_{ee}}\right) + 0.367$$

Figure 3 summarizes all of AIC's PY9 ASHP systems by unique SEER and EER values. The x-axis shows the quantity (by percent of total). The green hash marks represent the adjusted SEER values for every unique system, and the green line represents the weighted average of adjusted SEER values at one-half SEER increments. As nameplate SEER (the black line) increases, adjusted SEER tends to deviate further from the nameplate value with one exception. The highest quantity of 16.5 SEER units had EER of 9.5 (as emphasized in Figure 3). The team confirmed these were variable-capacity units. As expected, the adjusted SEER is less than nameplate SEER for units of this type.

Figure 3. Overview of Nameplate and Adjusted SEER Values using Percent SEER_{adj} Formula from AIC's PY9 Participants



Recommendation 6: The EER to SEER ratio of baseline units is approximately 0.8 (e.g., 7.5 EER to 9.3 SEER, 11 EER to 14 SEER). The team found that variable-capacity systems operated at an efficiency of 9% less than the nameplate rating and developed an algorithm to adjust the SEER value. As the ratio of EER to SEER for a high-efficiency unit decreases below 0.8, the adjusted nameplate SEER value decreases. The TRM should use the following algorithm to estimate an adjusted SEER value:

$$\% \, SEER_{adj} = 0.805 \times \left(\frac{EER_{ee}}{SEER_{ee}}\right) + 0.367$$

Include this adjustment in the standard algorithm to determine cooling savings as follows:

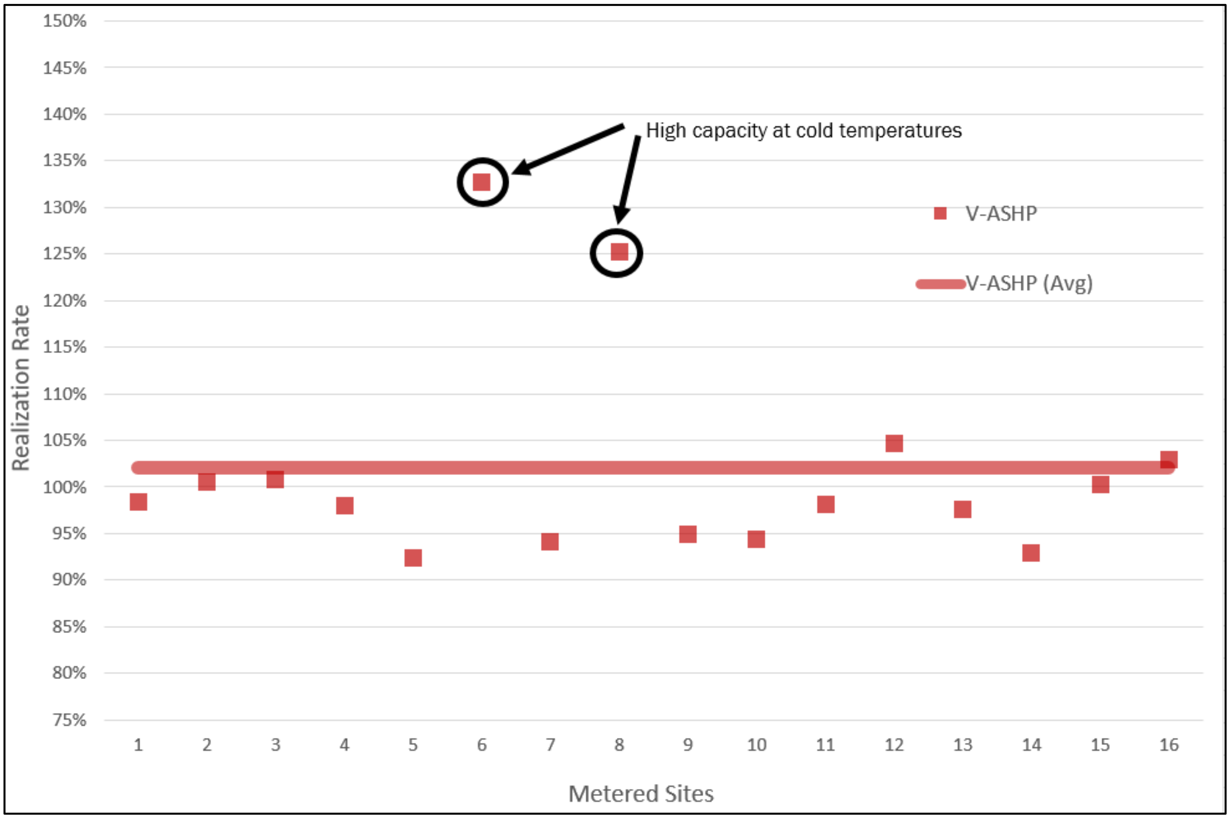
$$\Delta kWh_{cool} = EFLH \times Capacity_{cool} \times \frac{1\,kBtu}{1{,}000\,Btu} \times \left(\frac{1}{SEER_{base}} - \frac{1}{\%SEER_{adj} \times SEER_{ee}}\right)$$

Finding 7: Variable-capacity heat pumps often have higher cold temperature capacity ratios[14] than standard heat pumps. For each variable-capacity unit, the team chose a baseline unit of equivalent capacity at 47°F. The team followed the approach described above (see Finding 6) to determine an adjusted HSPF[15] value for variable-capacity ASHPs. Figure 4 shows two data points with an actual HSPF of greater than 125% of the operating efficiency of a single-speed system. In each case, this was primarily due to the operation of the variable-capacity system at cold conditions. A baseline ASHP of equivalent capacity at 47°F is unable to produce the equivalent capacity below a certain outdoor temperature. Consequently, the baseline ASHP requires more electric resistance (ER) heat. The team accounted for the increase in ER energy consumption using detailed capacity and coefficient of performance (COP) curves for both a baseline and the variable-capacity ASHP systems. For additional detail, see the Methodology Details: Heating Savings section in Appendix A of this document.

[14] The ratio of rated heating capacity at 17°F to the heating capacity at 47°F

[15] All AHRI HSPF rated values for split central ASHPs include impacts of defrost energy consumption and electric strip heat use. The team used only the fan and condenser energy consumption to determine adjusted HSPF, comparing equivalent baseline system performance to variable-capacity performance. Use of supplemental electric resistance heat by a variable-capacity system generated no savings.

Figure 4. Comparing Expected HSPF to Actual HSPF for Each System Type



The team found variable-capacity systems operated about 3% more efficiently than a baseline single-speed system would have operated. Table 13 shows the average capacity ratio for the variable-capacity ASHPs that the team metered, and the capacity ratio for a single-speed baseline ASHP.

Table 13. Summary of Variable-Capacity and Baseline System Capacity Ratios

System Type	HSPF Adjustment Factor	17° F Capacity	47° F Capacity	Capacity Ratio
Single-Speed	1.00	24,147	37,918	0.64
Variable-Capacity	1.03	31,497		0.83

Recommendation 7: To account for the increase in heating efficiency due in part to higher-capacity heat output at cold temperatures, the team recommends calculating an HSPF adjustment factor using the following equation:

$$\% \, HSPF_{adj} = \left(\frac{17\,°F\ Capacity}{47\,°F\ Capacity}\right) \times 0.158 + 0.899$$

The numerical constants in this equation (slope and intercept) are derived from the HSPF adjustment factors and capacity ratios in Table 13. When an ASHP's AHRI certificate is known, the heating capacity values at 17°F and 47°F are readily available from AHRI's database (see Figure 12 in Appendix A). AIC records each ASHP's AHRI certificate number, but does not explicitly report both heating capacity values. To do so should not require sufficient additional effort.

The team recommends revising the TRM to include the following algorithm adjustment to determine heating savings for ASHPs:

$$\Delta kWh_{heat} = EFLH \times Capacity_{heat,47} \times \frac{1\ kBtu}{1{,}000\ Btu} \times \left(\frac{1}{HSPF_{base}} - \frac{1}{\%HSPF_{adj} \times HSPF_{ee}}\right)$$

Evaluation Methodology

In conducting this research, the evaluation team performed the activities outlined below.

Data Review

The team reviewed AHRI[16] certificate information from AIC's tracking database to determine indoor (furnace) and outdoor equipment model numbers. Using the outdoor condensing unit model number, the team flagged all 2015 and 2016 HVAC systems with inverter-driven (variable-capacity) HVAC systems. The team also determined which 2015 CAC participants likely had a standard-efficiency furnace blower motor.[17]

TRM Review

The team reviewed the TRM measure description, savings algorithms, and inputs for the CAC, ASHP, and ECM fan motor measures. Specifically, the team sought to use the information gathered in this study to update the TRM when necessary, or to provide recommendations for the strategic advisory group's consideration to ensure that *ex ante* savings are calculated appropriately for HVAC equipment installations. The team also reviewed the *in situ* baseline efficiency assumptions for early replacement installations, and independently estimated baseline efficiency values.

Sampling

The team flagged all tracking data according to system types (e.g., variable-capacity CAC, ASHP, standard furnace fan) and randomly sampled a subset of each system type, targeting 15 variable-capacity CACs and 15 variable-capacity ASHPs with ECM fans. To establish baseline furnace fan energy consumption, the team targeted 30 CACs with standard-efficiency furnace fans.

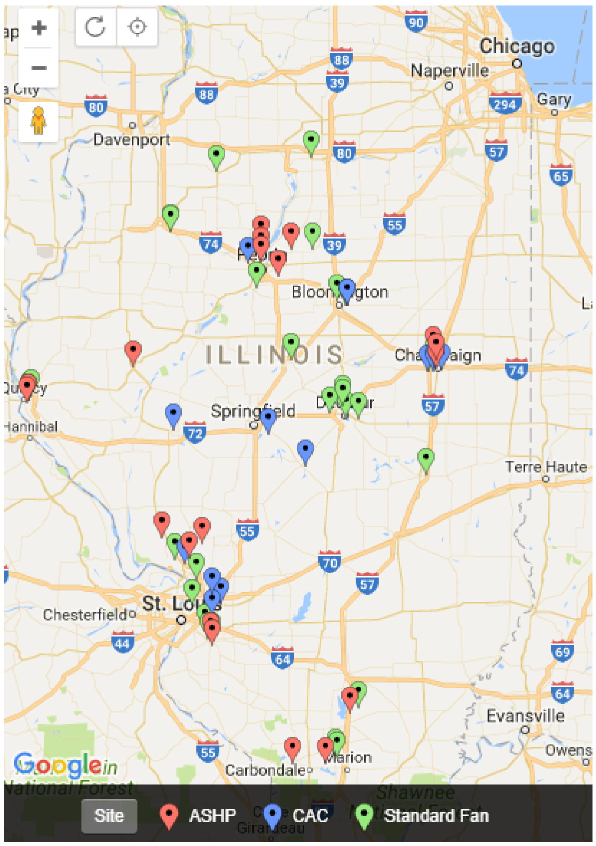
[16] AHRI information available online at https://www.ahridirectory.org/ahridirectory/pages/home.aspx

[17] The team used installed cost data, condenser model types, and AHRI certification report data to increase the likelihood that sampled CACs did not have an ECM.

Participant Recruitment

Participant recruitment for residential metering studies can prove challenging, given the need to enter participants' homes and leave equipment. The evaluation team sought to recruit a representative sample of AIC customers for this study (see the Non-Sampling Error section below). To achieve a high participation rate, the team provided incentives ($100 at the time of the first visit to install meters, and $75 at the time of the second visit to remove equipment). The team also included a $2 bill in recruitment letters mailed to each sampled participant. In total, the team sent 200 letters and recruited 62 metering participants—a relatively high participation rate for such studies. Figure 5 shows the locations of meter study participants.

Figure 5. Locations of Metering Study Participants by Equipment Type



Site Visits and Metering

The evaluation team conducted the initial site visits in February and March 2016, then returned to remove metering equipment in March 2017. For all systems, the team metered the following:

- Fan currents (using a current transformer), along with spot measurements of fan power factors and furnace voltages
- Furnace static pressure differentials across the entire furnace (i.e., upstream of the filter in return duct and downstream of the evaporator coil in supply duct)
- Supply and return temperatures and relative humidity

- Outdoor systems true power (i.e., total energy consumption in one-minute intervals) using either 20-amp or 50-amp current transformers (depending on systems' rated amperages)
- Outdoor temperatures and relative humidity

The following figures provide examples of meters installed by the team. Figure 6 shows power meters installed in the outdoor condenser of two variable-capacity systems.

Figure 6. Power Meters Installed in Two Variable-Capacity Systems

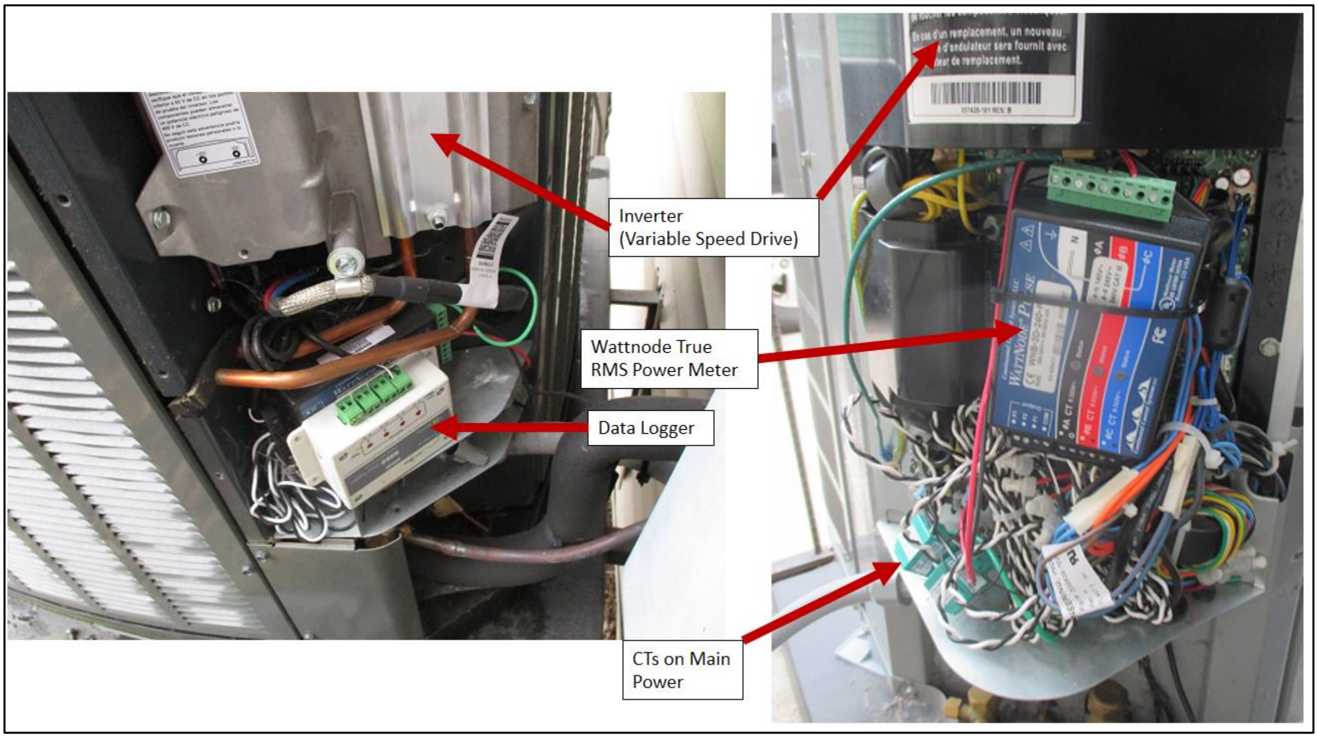


Figure 7 shows a power meter installed in a standard, single-speed air conditioner. Though the electrical compartments in Figure 6 and Figure 7 are normally covered, the team exposed them here to show the power meters. An AC/DC inverter is quite expensive (added cost of ~$2000) and large. The inverter and complex control board is housed in the electric compartment of the condenser. We found that the electric compartments of variable-capacity systems include the large inverter and have much less free space than single-speed systems, which makes meter installations difficult. Despite this challenge, the team successfully installed meters as planned in all systems.

Figure 7. Power Meter Installed in Single-Speed CAC

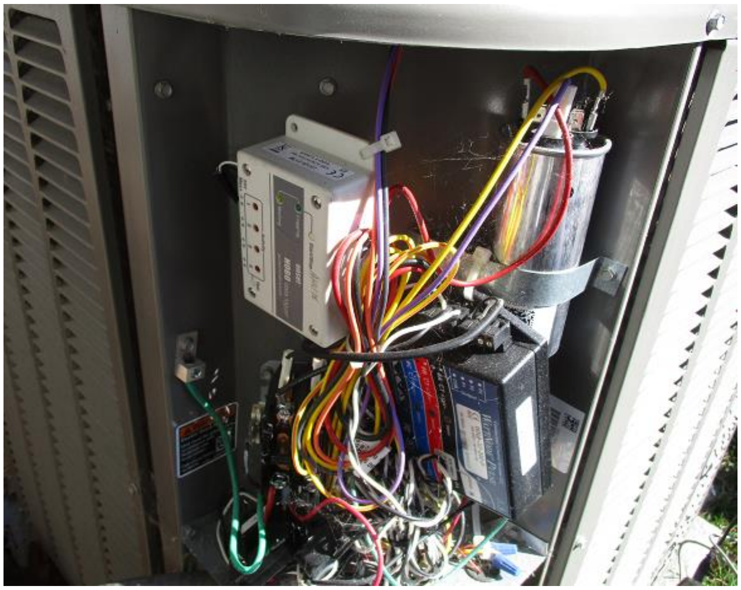
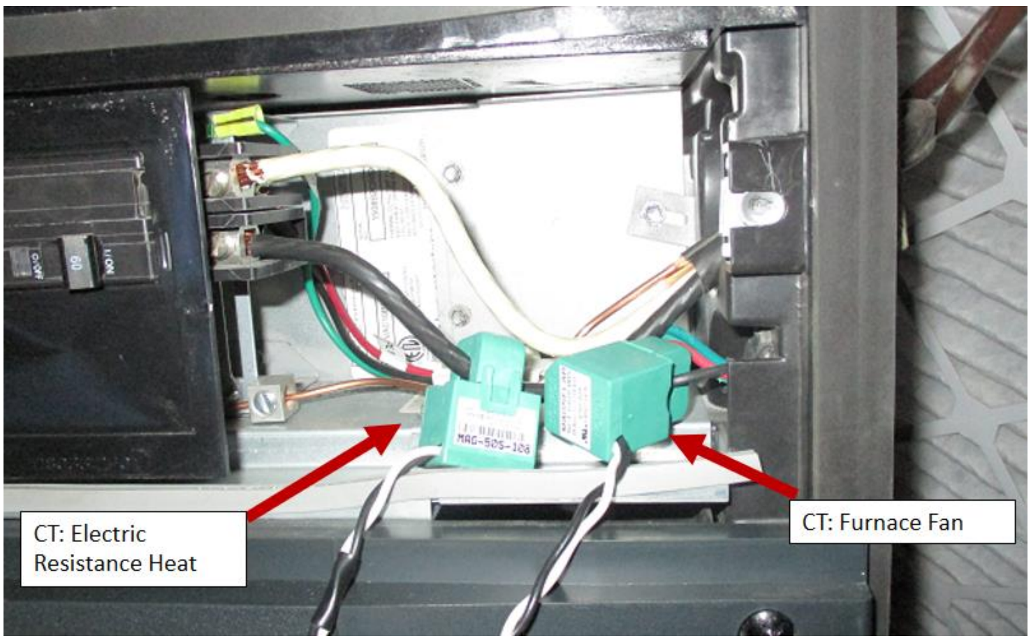


Figure 8 shows two current transformers installed in an ASHP furnace electrical compartment. One current transformer measured the furnace fan's amperage, and the other measured the electric strip heat circuit's amperage.

Figure 8. Current Transformers Installed in ASHP with Electric Resistance Strip Heat



Similarly, Figure 9 shows a current transformer installed in a CAC that uses an ECM. To isolate the fan blower amperage from other energy-consuming components (i.e., the furnace draft inducer blower or control board), the team identified the wire that carried continuous current to power the ECM. The figure also shows a temperature and relative humidity sensor installed to monitor the return air's condition.

Figure 9. Current Transformers Installed on Main Power to ECM and Temperature/Relative Humidity Sensor

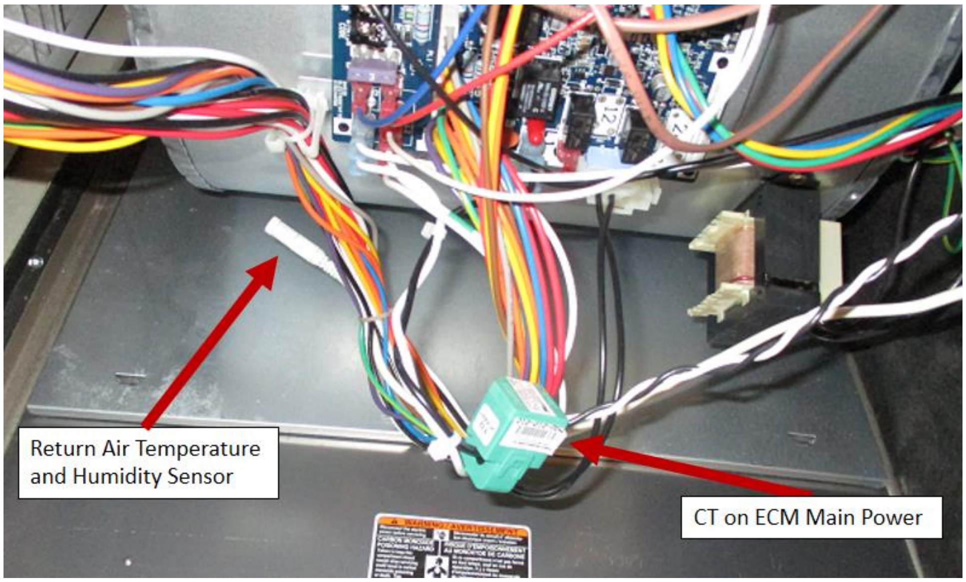
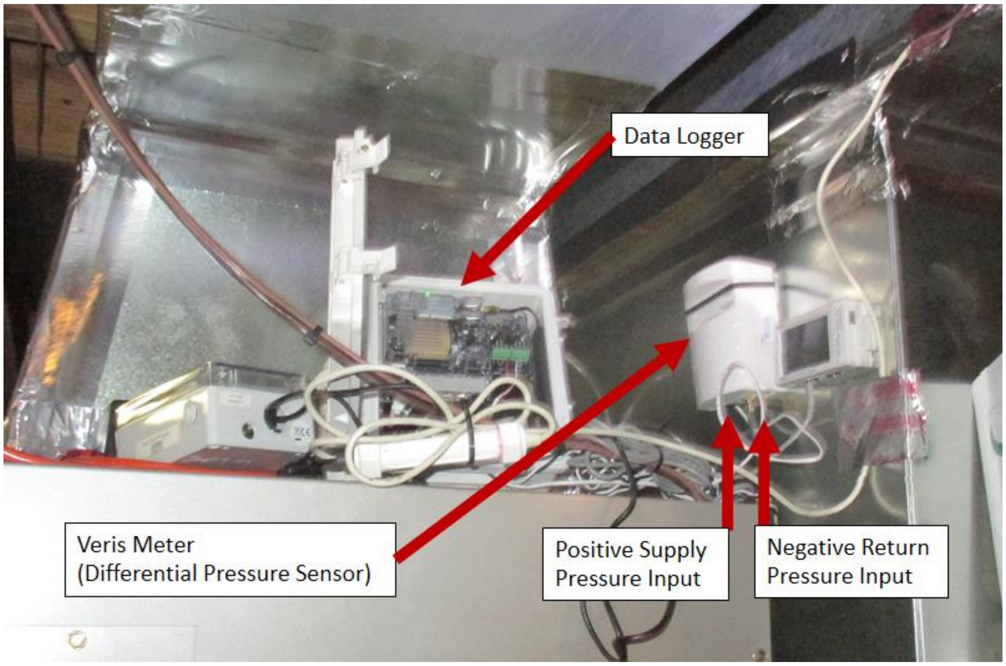


Figure 10 shows the data logger and pressure sensor that the team installed on each indoor unit (furnace). The team metered differential static pressure across the entire furnace (upstream of the air filter and downstream of the evaporator coil) rather than across the furnace blower.[18] The team collected these data points to compare variance in power and pressure for furnaces examined in the study, not to establish actual fan performance estimates.

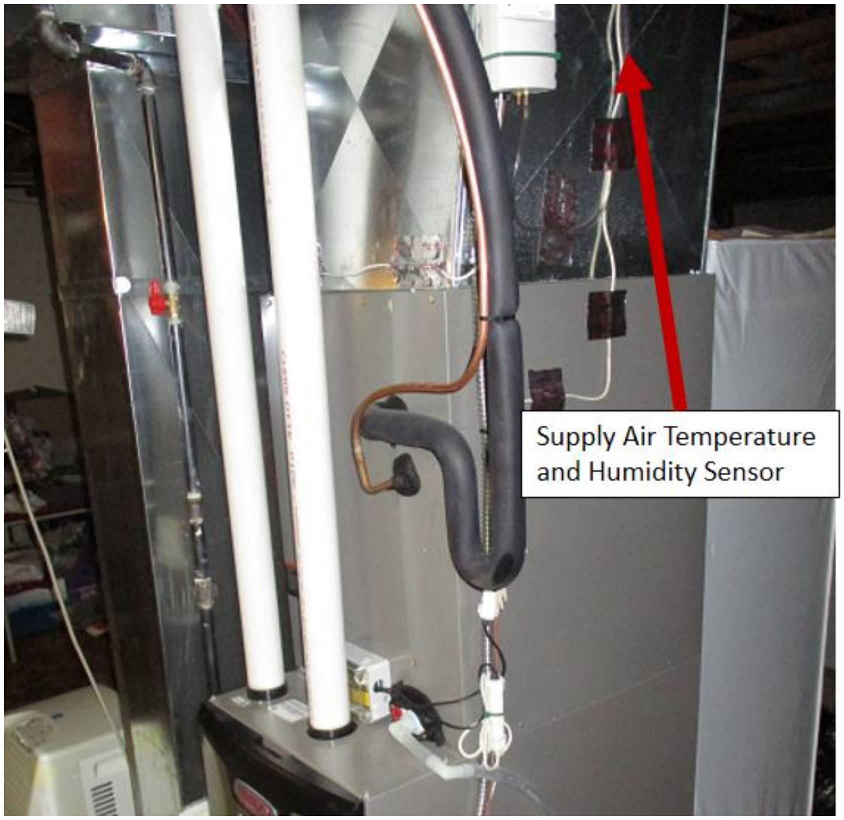
[18] Assessing actual fan efficiency (in terms of CFM per watt) requires continuous pressure, flow, and power measurements (which fall outside this study's scope). Typically, static pressure immediately downstream of the blower cannot be measured unless a service port is drilled into the furnace cabinet, which could void a manufacturer's or HVAC contractor's warranty.

Figure 10. Differential Pressure Sensor and Logging Equipment on Indoor Furnace



Lastly, the team installed a temperature and relative humidity sensor in the supply air stream (i.e., the furnace supply plenum), approximately one foot downstream from the evaporator coil (shown in Figure 11). Typically, the team installed the supply static pressure hose in the same location.

Figure 11. Supply Air Temperature and Relative Humidity Sensor



Analysis of Meter Data

The evaluation team reviewed all meter data, primarily to remove any erroneous data and to ensure all data estimates fell within the expected measurement range. The team developed analytical procedures and tools to inspect data quality and to perform analysis calculations. A senior-level quality control review applied to all data analyses ensured the accuracy of calculations and confirmed the consistency of findings with general expectations. Appendix A includes explanations of the challenges inherent in using sub-meter data to determine performance and estimate savings of variable-capacity HVAC systems.

Sources of Error in this Study

Sampling Error

The evaluation team installed meters on 62 HVAC systems and determined the precision of key parameters at the 90% confidence level. Precision varied for each key parameter (e.g., HSPF, SEER, demand savings, ECM heating, cooling, and circulation mode savings) from approximately ±3% to ±20% at the 90% confidence level.

Non-Sampling Error

Self-Selection Bias

As individual homeowners volunteered all sites used in the meter study, the results may be subject to self-selection bias. This happens if people who agreed to participate in a study differ from those who refused to participate in a way that correlates with the study findings. Given the difficulty of recruiting participants for such studies, a random selection of customers usually produces a low response rate, opening the study to such bias. Though this is an inherently difficult bias to control, no obvious reasons emerged to expect a relationship between willingness to participate and normal HVAC use. In addition, the team provided incentives for each participant ($175) to encourage participation. This could effectively reduce bias by encouraging the participation of some customers who did not express interest in the study or its results.

Participant Operational Use Bias (Hawthorne Effect)

In any human subject study, a possibility exists that participants will change their behaviors due to the study itself (in this case, they would use their cooling equipment differently than when they were not part of the study). Social psychology literature refers to this bias as the Hawthorne Effect. The team corrected for the effect by instructing all study participants not to change their equipment-use habits due to the study. The team expected compliance with this instruction would be reasonably high, especially as minor initial behavior changes would tend to fade over the yearlong period that the meters remained in place.

Physical Measurement Error

Outliers

Occasionally, field metering produces unexpected data or numbers simply out of the normal range observed for other similar metered data. To identify and address possible outliers, the evaluation team divided questionable data into two categories:

- Physically unexplainable data
- Data falling outside the range of most other data

Calibration

To minimize measurement errors from energy and temperature sensors, field monitoring staff checked to determine that all sensors used in the field operated properly. Staff took parallel measurements with temperature sensors to ensure variability less than 1°F, and they compared energy and amperage sensor readings to spot measurements from handheld amperage/power meters.

Data Recording

To ensure the recording of realistic data, the team monitored indoor conditions to compare heating and cooling energy use. To ensure the simultaneous recording of data such as energy consumption and temperature, field staff used consistent measurement intervals, synchronized for all metering equipment at each site. This consistency ensured that data from multiple sites could be compared across a uniform time period.

Appendix A. Additional Information

The team used sub-metered HVAC energy consumption data at discrete intervals to estimate savings for a typical year with the equation:

$$\Delta kWh = \sum_{i=T_L}^{T_H} \left(kWh_{METERED} \times \frac{EER_{EE}}{EER_{BASE}} - kWh_{METERED} \right) * \frac{Hours_{TMY}}{Hours_{ACTUAL}}$$

Where:

$kWh_{METERED}$	=	energy consumption of an outdoor unit and indoor fan
EER_{EE}	=	Instantaneous efficiency of the installed equipment
EER_{BASE}	=	Instantaneous efficiency of the baseline equipment

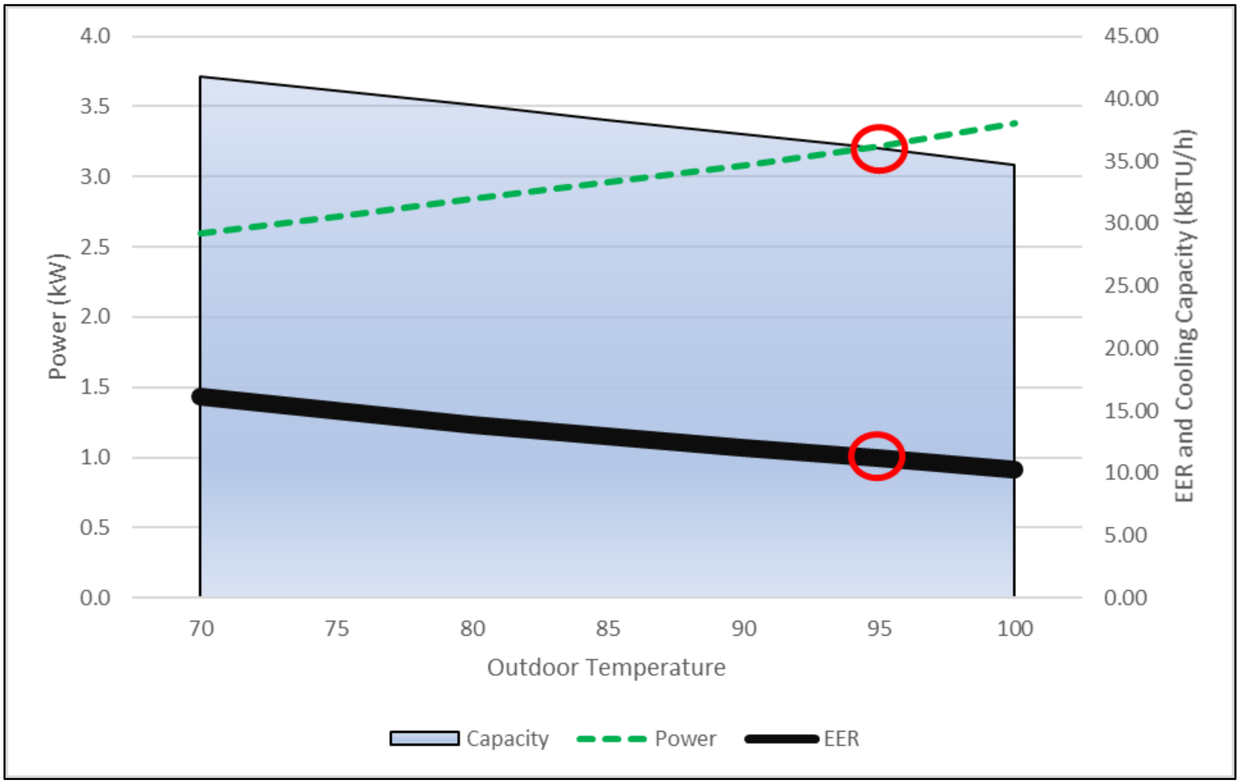
Manufacturers typically provide power and capacity specification tables. Efficiency, a function of outdoor temperature, can be calculated using these tables. The team compiled manufacturers' specification data from 10 different, baseline (14 SEER), single-speed ASHP systems to develop generic functions of capacity, power, and efficiency. Figure 12 shows the AHRI-certified efficiency and capacity values tracked when an AHRI certificate number is known.

Figure 12. AHRI Certification Example for ASHP

Cooling Capacity (Btuh):	35000
EER Rating (Cooling):	13.00
SEER Rating (Cooling):	18.00
Heating Capacity(Btuh) @ 47 F:	32200
Region IV HSPF Rating (Heating):	10.00
Heating Capacity(Btuh) @ 17 F:	25200

Figure 13 shows the cooling functions that the team developed for a single-speed system, with AHRI-rated efficiency and capacity circled.

Figure 13. Cooling Operation: Efficiency, Power, and Capacity Functions of Baseline (14 SEER) System

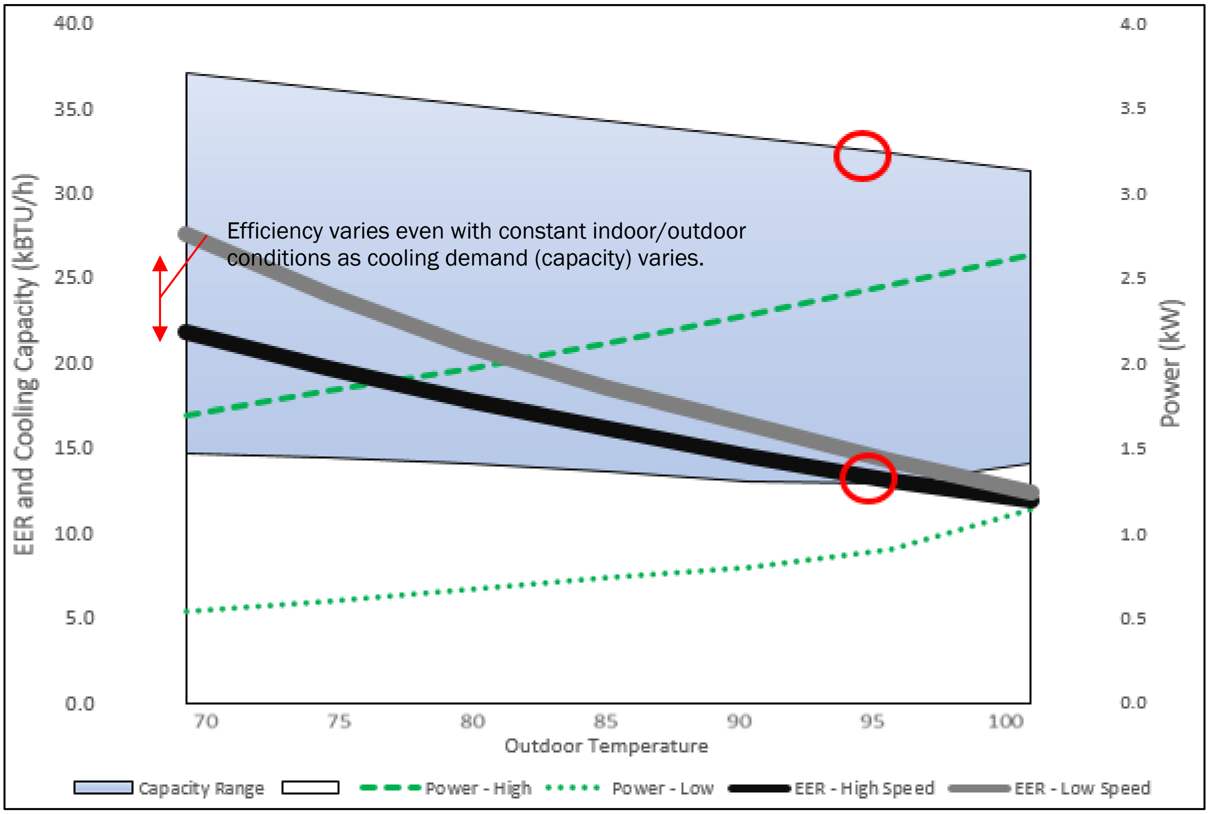


The EER is the system performance with the compressor operating at maximum (rated) speed at an outdoor temperature of 95°F. EER is a single, instantaneous measurement of efficiency calculated in the same way for all system types. The SEER calculation for single-speed systems is relatively straightforward, while the SEER calculation for variable-speed systems assumes partial-load operation (i.e., lower speed, higher efficiency) for nearly all hours of the cooling season.

An HVAC system's nameplate-rated HSPF and SEER is calculated by following the AHRI 210/240 protocol, which requires laboratory measurements of instantaneous steady-state efficiency at several outdoor temperatures and fixed compressor speeds. HSPF and SEER are then calculated from bin temperature analysis of a specified region (usually U.S. Region IV). HVAC systems installed in residential homes may operate at outdoor conditions that differ from the AHRI test conditions. Though a single-speed HVAC system may perform as expected, actual heating and cooling average efficiencies may differ from AHRI SEER due to conditions at which the system operates. Actual HSPF may differ for the same reason, but the use of electric resistance supplemental heat and energy consumption in defrost mode also impact the actual heating efficiency in the winter.

Figure 14 shows the variance of capacity and efficiency for a variable-speed system. The only certified AHRI efficiency values for cooling are the SEER and the EER of the "rated" speed at 95°F. As shown in this example, EER varies significantly. At 70°F, if operating at low speed, this system may provide cooling with an efficiency between 22 EER and 28 EER. At 100°F, the system operates at about 12 EER. The operation range for the baseline system (Figure 13) was not nearly as broad, ranging from 17 EER to 10 EER.

Figure 14. Cooling Operation: Efficiency, Power, and Capacity Functions of Variable Capacity (20 SEER) System



The team used metered energy consumption and indoor/outdoor conditions to estimate the system's provided capacity for all metered intervals throughout the heating and cooling season. The team employed a linear interpolation of from low to high power at each temperature, and input the actual observed (metered) power to estimate percent load. The team then assumed actual capacity followed the same relationship and calculated the capacity for each metered interval. Assuming a static relationship between the baseline efficiency and outdoor temperature, the team could calculate savings for every metered interval. For the metered period, the team calculated savings using the following equation:

$$\Delta kWh = \sum_{i=T_L}^{T_H} \left(kWh_{METERED} \times \frac{EER_{EE}}{EER_{BASE}} - kWh_{METERED} \right)$$

Similarly, the team calculated cooling seasonal efficiency (SEER) using the following equation:

$$\text{SEER} = \frac{\sum_{i=T_L}^{T_H}(kWh_{METERED} \times EER_{EE})}{kWh_{METERED}}$$

Though the analysis of single-speed system operation indicated that a single-speed SEER baseline value may be reliable, the team did not simplify the analysis with such an assumption. Refer to the data point called out in Figure 2. This single-speed CAC operated at 108% of its nameplate SEER value because the home was vacant during the hottest parts of the summer. Consequently, the system operated at relatively mild conditions[19]. Rather than using the observed single-speed system operation (and resulting metered SEER value, 98% of 14.5, See Finding 6) as a baseline, the team developed baseline EER curves for each variable-capacity system to compare how a single-speed system would have operated in its place.

With metered capacity from the variable-capacity system for all temperatures, we calculated the expected baseline energy consumption and an adjusted baseline SEER for each system. The average adjustment for baseline SEER was small – about 101% of the performance value (i.e. 9.3 SEER baseline averaged 9.4 SEER). This means that, overall, the variable-capacity systems ran at slightly milder (more efficient) conditions than did the single-speed CAC systems (i.e., the 14.5 SEER systems that we metered had average SEER of 14.2).

Methodology Details: Heating Savings

The team followed the same approach to determine heating savings with one exception. The calculation of heating savings is complicated by the fact that many variable-capacity ASHP condensers can provide more heat than standard ASHPs at cold temperatures.

The team developed a model that could input the AHRI-rated capacity values (capacity at 17°F and 47°F) of any installed system to develop efficiency, capacity, and power curves for a generic baseline single-speed system. Figure 15 shows an example of functions that the team developed to establish the heating performance of a single-speed ASHP (the AHRI-rated capacities are circled in the figure).

[19] To further illustrate the example, if a 14.5 SEER, 11 EER unit only operates when the outdoor temperature is 95°F, it's actual seasonal efficiency would be 11 SEER.

Figure 15. Heating Operation: Efficiency, Power, and Capacity Functions of Baseline (8.2 HSPF) System

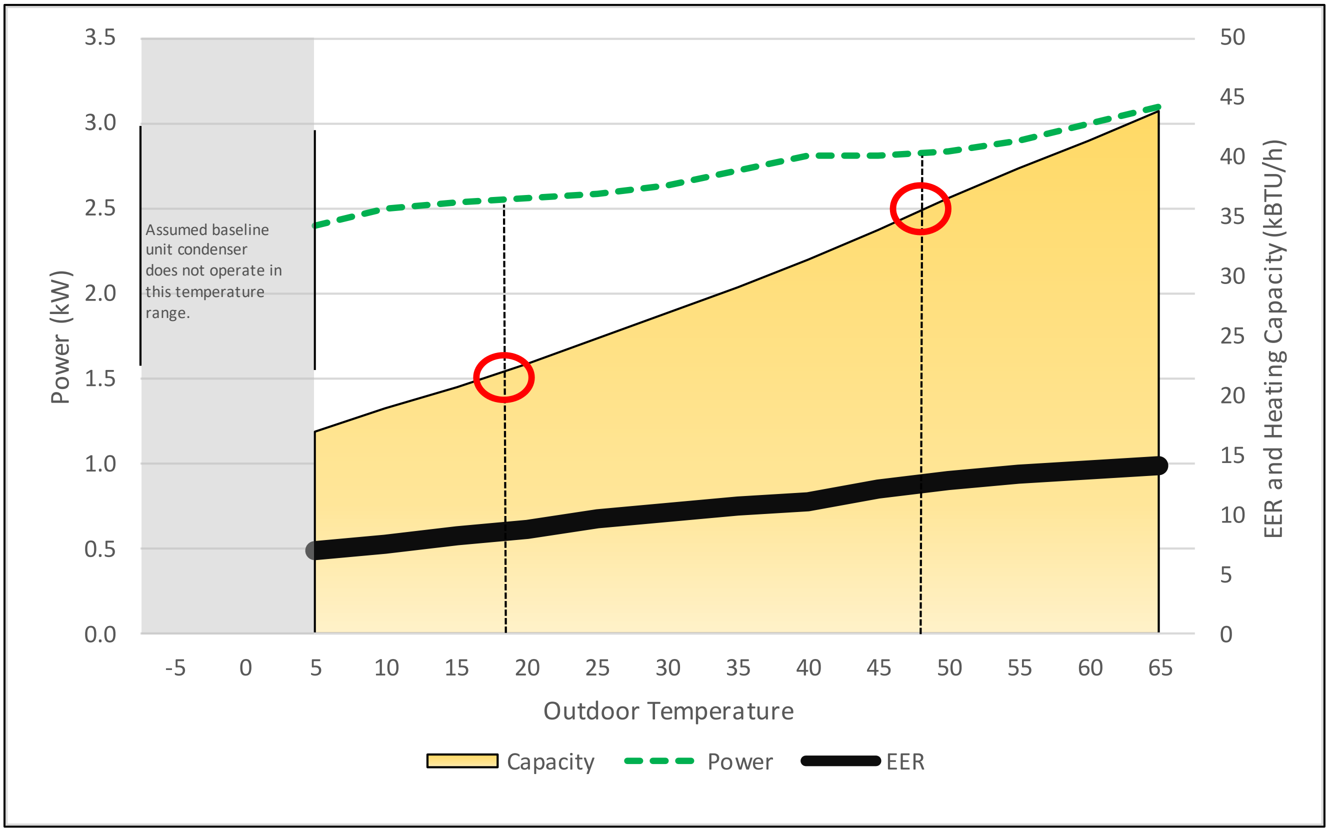


Figure 16 shows an example of manufacturer's performance data that the team used to develop capacity, power, and efficiency curves. Figure 17 shows the capacity curves for an example variable-capacity system and includes the single-speed capacity curve for the baseline system for this unit. As the figure shows, the team ensured equivalent capacity for both systems at 47°F. When the outdoor temperature drops below about 15°F, the baseline system is incapable of producing the required capacity (see the black line in Figure 17 which represents actual operating capacity of the metered outdoor unit). The team assumed the difference between operating capacity and baseline system capacity would have been provided by inefficient electric resistance strip heat. This effectively degrades the baseline efficiency and increases the savings.

Figure 16. Example of Manufacturer's Performance Data for Variable Speed ASHP (Carrier 25VNA825)[20]

25VNA825 / FE4ANF005 HEATING EFFICIENCY MODE – OUTDOOR COIL ENTERING AIR TEMPERATURES °F (°C)

INDOOR AIR	−3 (−19.4)				7 (−13.9)				17 (−8.3)				27 (−2.8)			
EDB °F (°C)	ID SCFM	Capacity MBtuh Total	Capacity MBtuh Integ‡	Total Sys KW†	ID SCFM	Capacity MBtuh Total	Capacity MBtuh Integ‡	Total Sys KW†	ID SCFM	Capacity MBtuh Total	Capacity MBtuh Integ‡	Total Sys KW†	ID SCFM	Capacity MBtuh Total	Capacity MBtuh Integ‡	Total Sys KW†
STAGE 5																
65 (18.3)	N/A				N/A				825	20.11	18.34	2.16	825	22.21	19.73	2.08
70 (21.1)										19.90	18.14	2.26		21.97	19.52	2.17
75 (23.9)	N/A				N/A				825	19.71	17.97	2.36	825	21.73	19.30	2.27
STAGE 3																
65 (18.3)	N/A				340	9.32	8.57	1.42	500	11.26	10.27	1.36	650	13.17	11.70	1.22
70 (21.1)						9.21	8.46	1.48		11.11	10.13	1.42		12.99	11.54	1.29
75 (23.9)						9.10	8.36	1.54		10.96	10.00	1.48		12.82	11.39	1.35
STAGE 1																
65 (18.3)	N/A				N/A				N/A				N/A			
70 (21.1)																
75 (23.9)																

INDOOR AIR	37 (2.8)				47(8.3)				57 (13.9)			
EDB °F (°C)	ID SCFM	Capacity MBtuh Total	Capacity MBtuh Integ‡	Total Sys KW†	ID SCFM	Capacity MBtuh Total	Capacity MBtuh Integ‡	Total Sys KW†	ID SCFM	Capacity MBtuh Total	Capacity MBtuh Integ‡	Total Sys KW†
STAGE 5												
65 (18.3)	825	25.00	22.75	2.11	825	27.16	27.16	2.11	N/A			
70 (21.1)		24.69	22.46	2.21		26.80	26.80	2.21				
75 (23.9)		24.36	22.17	2.31		26.41	26.41	2.31				
STAGE 3												
65 (18.3)	650	15.11	13.75	1.25	650	17.04	17.04	1.25	650	19.10	19.10	1.27
70 (21.1)		14.89	13.55	1.32		16.77	16.77	1.32		18.78	18.78	1.35
75 (23.9)		14.67	13.35	1.39		16.51	16.51	1.40		18.44	18.44	1.42
STAGE 1												
65 (18.3)	N/A				585	7.58	7.58	0.44	585	9.05	9.05	0.42
70 (21.1)						7.40	7.40	0.48		8.83	8.83	0.47
75 (23.9)						7.22	7.22	0.52		8.62	8.62	0.52

See notes on page 44

High Capacity

Low Capacity

An ASHP may be sized for either cooling or heating load. If sized for heating, contractors often choose a specific temperature at which the outdoor unit can no longer provide all the heat required by a home. The team chose to assume equivalent capacity at 47°F because the tracking data indicated that assumption was more reasonable than assuming equivalent capacity at 17°F. As Table 14 shows, the 47°F to 95°F degree capacity ratios are similar for each unit type (1.02 to 1.06). If units were sized for peak cooling load, it would be reasonable to assume a single-speed unit and variable-capacity unit have the same heating capacity at 47°F. It would be illogical to assume a single-speed unit would have the same heating capacity as a variable-capacity unit at 17°F.

Table 14. Summary of Average Capacity by Compressor Type

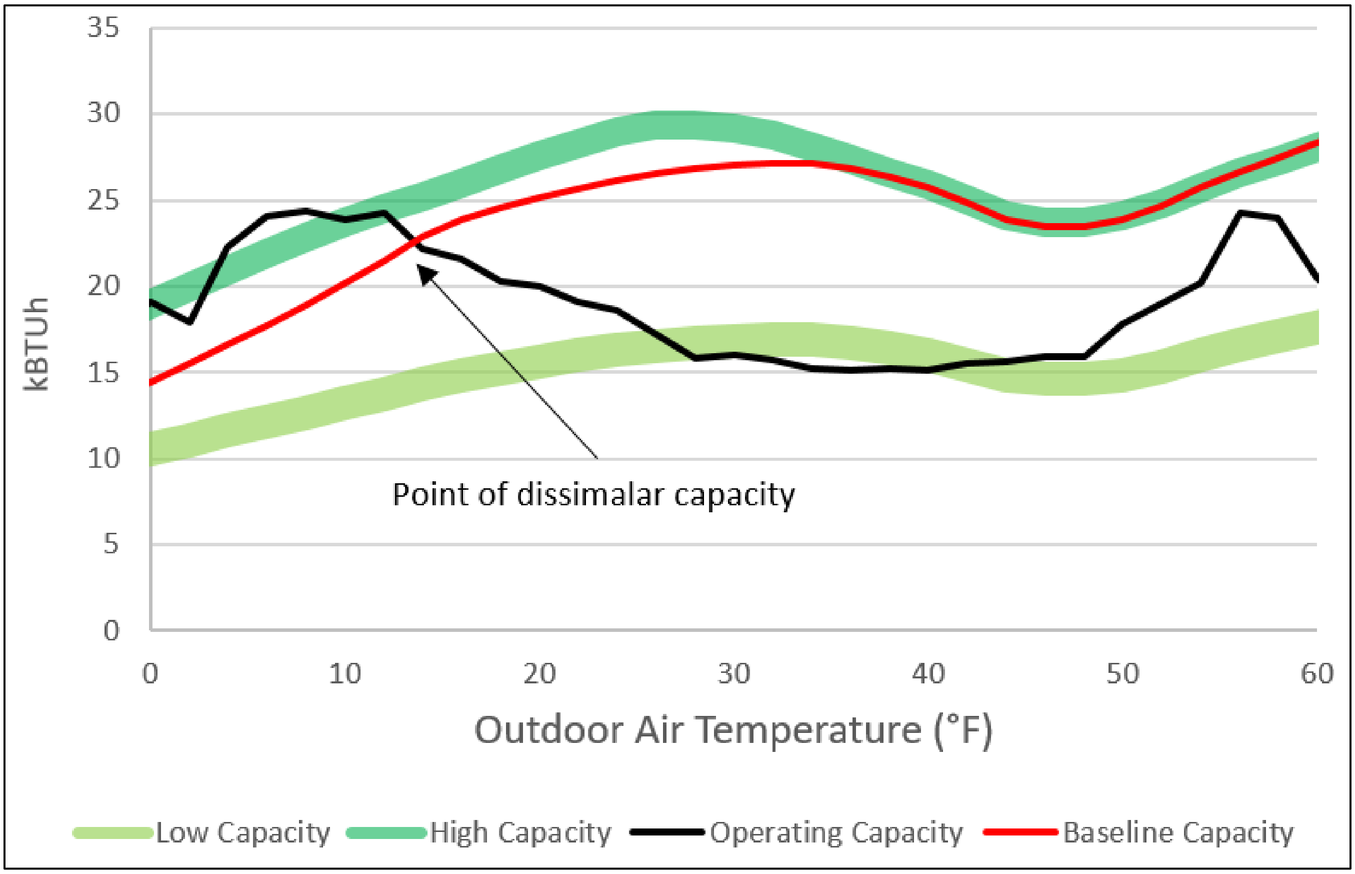
System Type	HSPF Adjustment Factor	47° F Capacity	17°F Capacity	Heating Capacity Ratio	95° F Cooling Capacity	95° F Cooling to 47°F Heating Capacity Ratio
Single-Speed	1.00	30,189	19,225	0.64	31,864	1.06
Two-Speed	1.00	34,348	21,686	0.63	35,551	1.04
Variable-Capacity	1.03	37,918	31,497	0.83	38,530	1.02

For homes in Illinois, peak heating load is generally greater than peak cooling load. The fact that single-speed ASHPs in AIC's program had average heating capacity at 17°F of only 19kBtu could indicate that these systems are either grossly oversized for cooling or that they are not sized to meet the home's heating load at 17°F. The

[20] Online: http://dms.hvacpartners.com/docs/1009/public/0e/25vna8-01pd.pdf

latter asserts the team's assumption (i.e., that a baseline unit would provide equivalent capacity at 47°F). This assumption can only be validated by assessing the precise balance point for each system type for the homes in Illinois, but this was not a research objective of the study.

Figure 17. Example of ASHP Capacity Curves for Variable-Capacity and Single-Speed (Baseline) Systems



The values in Table 15 further illustrate the team's approach. For every two-minute interval, the team used the actual metered power with low and high power and low and high capacity from published manufacturer's data (see Figure 16) to calculate an interpolated capacity output. We developed unit-specific capacity and power curves for each ASHP in the metering study, and we included the manufacturer's adjustments for specific indoor coil and furnace models.[21] We summarized results into temperature bins. Comparing initial baseline to final adjusted COP, some of the final adjusted COP values are lower because the baseline heat pump cannot provide as much capacity as the variable-capacity heat pump, so additional supplemental electric resistance heat would be required.

[21] Outdoor units can be matched with numerous combinations of indoor coils and furnaces. Manufacturers provide power and capacity correction factors that typically adjust capacity and power values within ±5%.

Table 15. Example Derivation of Baseline COP

Temp Bin	Published Performance Data for Variable-Capacity ASHP				Derived Performance Data for Baseline ASHP		Meter Data	Interpolated Value*	Btu/ (kW*3.412)	Δ Interpolated & Baseline Capacity	Btu/ (kW*3.412)	Final Baseline Value
	Low kW	High kW	Low Btu	High Btu	Baseline kW	Baseline Btu	Actual kW	Actual Btu	Actual COP	Capacity Required by Electric Resistance	Initial Baseline COP	Adjusted Baseline COP
4 - 6	1.47	2.81	12.2	21.9	N/A	N/A	**2.6**	20.4	**2.30**	20.4	N/A	**1.00**
6 - 8	1.49	2.85	12.7	22.8	2.78	17.0	**2.8**	22.4	**2.35**	5.39	1.79	**1.60**
8 - 10	1.50	2.89	13.2	23.7	2.81	18.2	**2.7**	22.3	**2.42**	4.11	1.89	**1.73**
10 - 12	1.52	2.92	13.8	24.5	2.84	19.4	**2.7**	22.8	**2.48**	3.45	2.00	**1.85**
12 - 14	1.53	2.94	14.4	25.3	2.87	20.6	**2.8**	24.2	**2.53**	3.60	2.10	**1.94**
14 - 16	1.54	2.96	14.8	26.0	2.88	21.5	**2.5**	22.4	**2.63**	0.94	2.19	**2.14**
16 - 18	1.54	2.97	15.3	26.8	2.88	22.1	**2.4**	22.1	**2.71**	0.00	2.25	**2.25**
18 - 20	1.53	2.98	15.7	27.6	2.86	22.6	**2.1**	20.3	**2.84**	0.00	2.31	**2.31**

*For illustration only. Btu values were calculated by interpolation for each two-minute interval. The team set limits on capacity and COP such that the final COP was within the range of the manufacturer's performance values for all temperatures.

Identifying Variable-Capacity Systems

Table 16 lists the most common variable-capacity ASHPs in AIC's program. The '**' in each model number replaces the kBtu nameplate capacity of the system. For example, the condenser model number of a three-ton (36,000 Btu) Carrier 25VNA series unit is '25VNA036'. Many model numbers include the letter 'V' to designate a variable-speed compressor.

Table 16. Most Common Variable-Capacity Condenser Model Numbers

Manufacturer	Percent of Total Systems
Carrier	24VNA0**
Carrier	25VNA0**
Carrier	25VNA8**
Bryant	280ANV0**
Bryant	288BNV0**
Trane	4TWV000**
Trane	4TWV80**
American Standard	4A6V00**
Lennox	XP20-0**
Lennox	XP25-0**
Lennox	SP25-0**

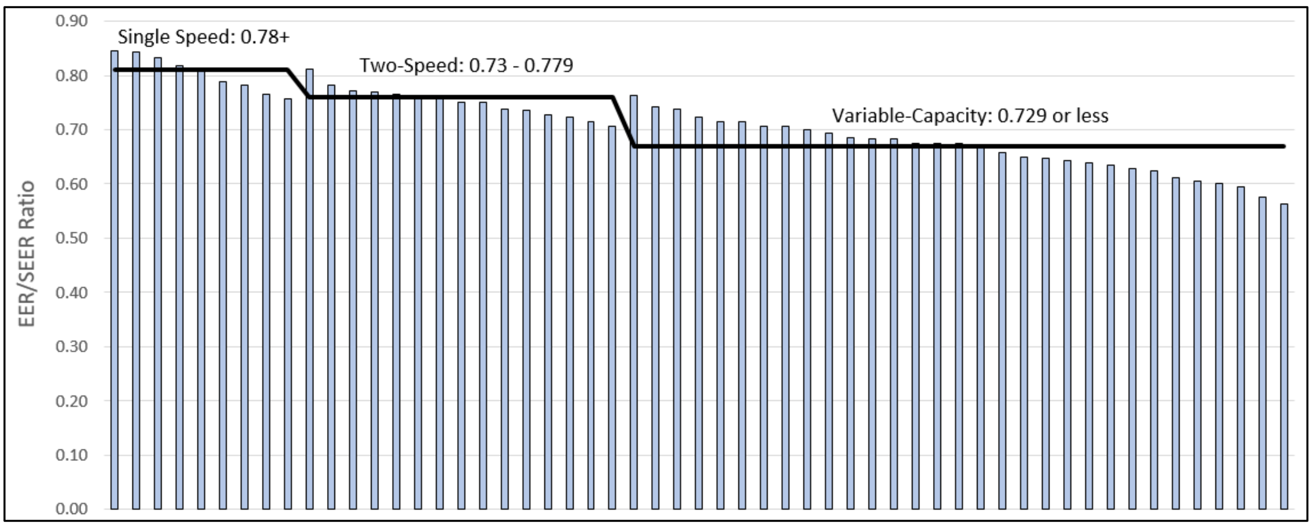
A simple analytical approach may also predict the units that have variable-capacity compressors with reasonable certainty.[22] Figure 18 shows a summary of all ASHPs in AIC's PY9 program. The black line represents the average EER to SEER ratio for these systems:

- Systems with single-speed compressors: 0.81
- Systems with two-speed compressors: 0.76
- Systems with variable-speed compressors: 0.67

Figure 18 includes the recommended ranges to determine the system type, as follows:

- If EER to SEER ratio is 0.78 or greater, assume single-speed
- If EER to SEER ratio is greater between 0.73 and 0.779, assume two-speed
- If EER to SEER ratio is less than 0.73, assume variable-capacity.

Figure 18. Ratio of EER to SEER for all ASHPs in AIC's PY9 Program



To determine whether a furnace is single-stage, dual-stage, or modulating requires nameplate information that is not included in the AHRI certificate. If the furnace nameplate includes two or more heating output values, it has dual-stage or modulating ability and will operate for more hours in heating mode than a single-stage furnace.

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22 Identification of system type is required to adjust ECM circulation mode runtime and savings. Variable-capacity systems have slightly lower circulation mode energy-savings potential because these systems operate longer in heating and cooling mode. This approach will not identify all system types correctly, but the error in average circulation mode savings for a population of efficiency program participants should offset.

Additional Charts and Information

The next three figures show the aggregate energy use for all system types, normalized by tons of cooling capacity. One unexpected finding was the amount of energy consumed in standby mode by the variable-capacity systems. The black line in Figure 20 shows the average standby power. Some systems occasionally or continuously cycle the compressor to prevent freezing. Some use a sump heater up to an outdoor temperature of 80°F. Still others have an optional (accessory) crankcase heater. Though the team's tertiary data and observations of meter data did not definitively show the standby power is higher in variable-capacity systems, further research may be warranted. The standby power averaged about 50 kWh for variable-capacity CACs and 90 kWh annually for variable-capacity ASHPs.

Figure 19. Average Hourly Energy Use of Single-Speed CAC with Standard-Efficiency Fan

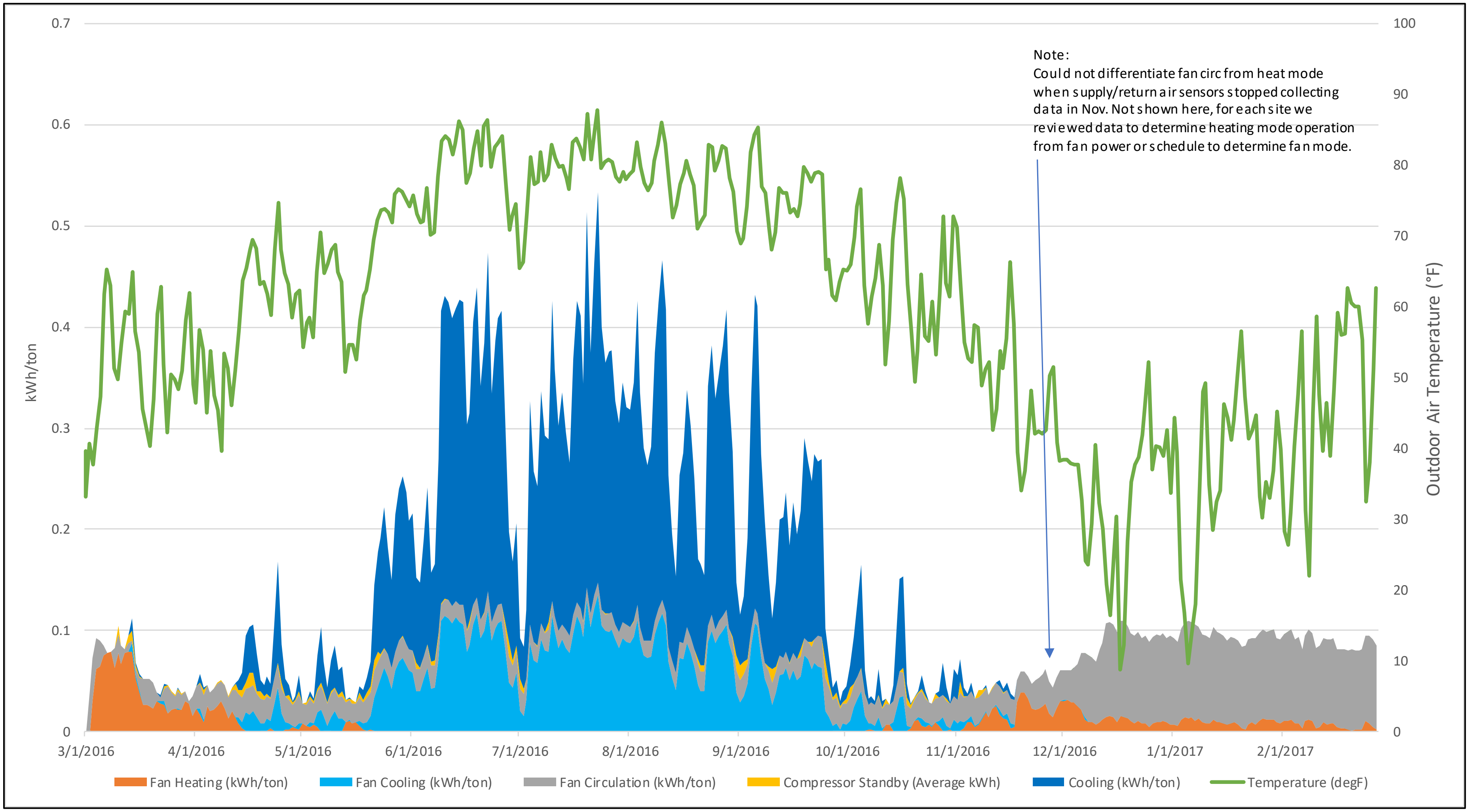


Figure 20. Variable-Capacity ASHP Average Hourly Energy Use

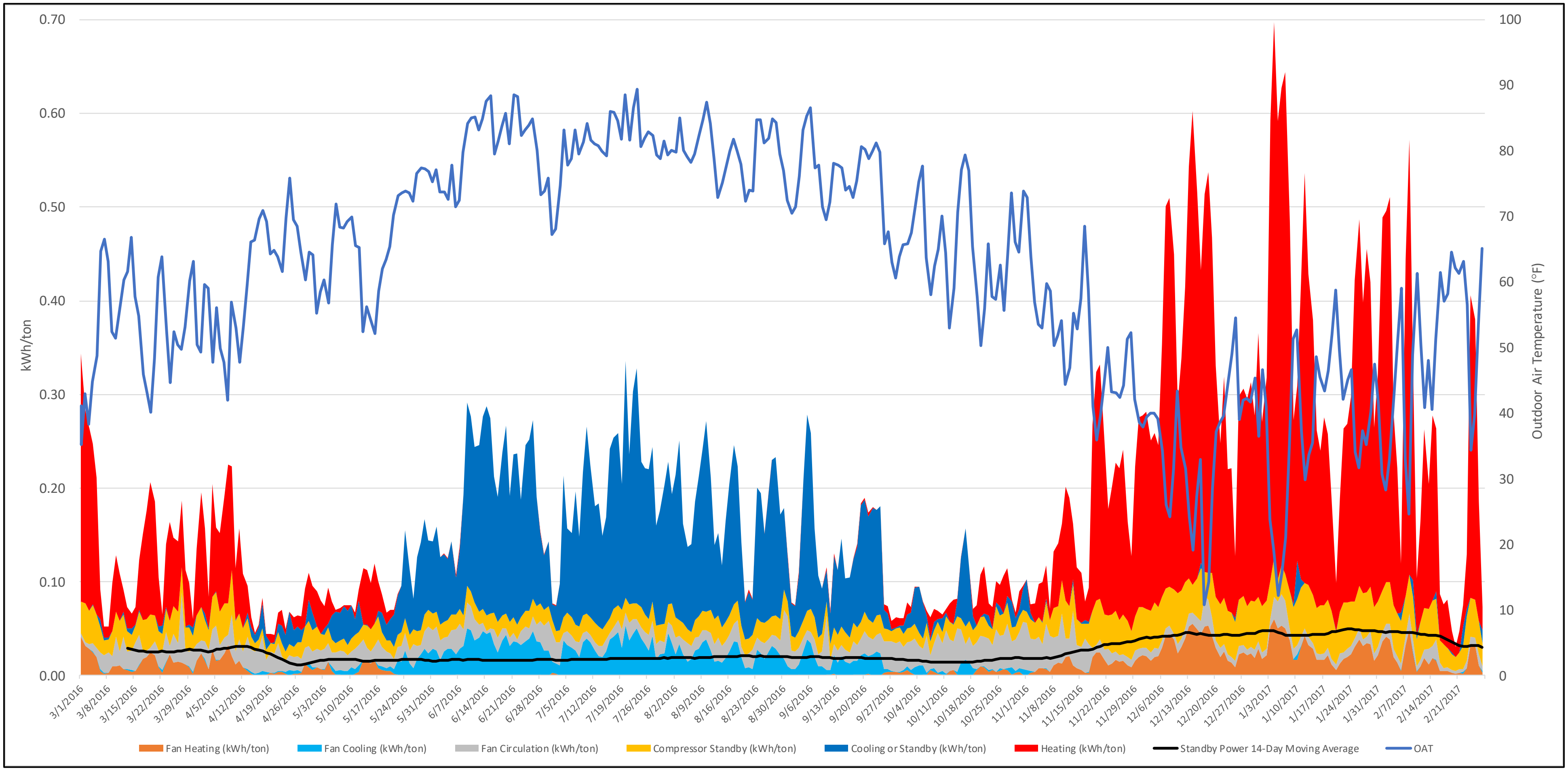
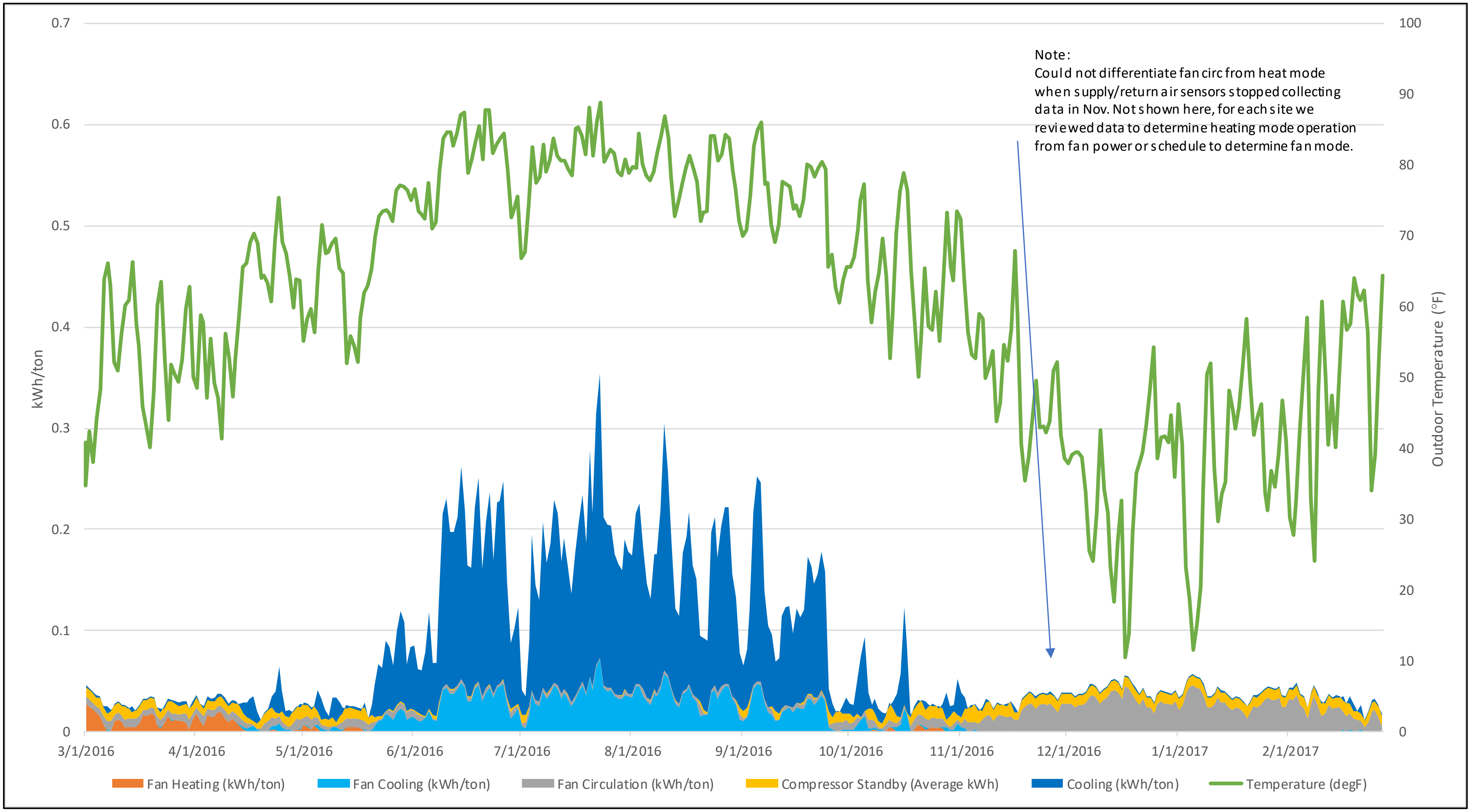


Figure 21. Variable-CAC ASHP Average Hourly Energy Use



Appendix B. TRM Updates

The embedded documents include all the recommended TRM updates from the findings in this study. Each word document is an extract from the relevant pages in IL-TRM V6.0 (effective January 1, 2018) and includes redline (tracked) changes.

5.3.1 Air Source Heat Pump

Air Source Heat Pump.docx

5.3.3 Central Air Conditioning

Central Air Conditioning.docx

5.3.5 Furnace Blower Motor

ECM savings may be calculated by following recommendation 1-3 in this report. This would require considerably more effort than required by the current TRM savings methodology for this measure. The team will work with stakeholders to determine the appropriate TRM updates for this measure.

Appendix C. ECM Baseline Hours Survey

The team interviewed ECM metering participants to determine the baseline hours of use for the fan in circulation mode. The embedded document below is the survey guide that that the team used to collect information to adjust the baseline hours of operation.

AIC_Metering_ECM_
Baseline_Survey.doc