# Exactly What Is a Full Load Cooling Hour and Does Size Really Matter?

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**ABSTRACT**

To calculate cooling savings, Technical Reference Manuals (TRMs), engineering savings calculations, and evaluation reports frequently rely on full-load cooling hours. Surprisingly, the derivation of those values, their applicability, and the sizing assumptions underlying them are not well known or understood. How air conditioners are sized not only interacts with these calculations, but beliefs around sizing lead utilities to incentivize contractors around how they size and install air conditioners. Unfortunately, many of these programs may be getting it wrong.

This paper uses actual metering data from 60 homes in a similar climate to address sizing and full-load cooling hours by examining how the air conditioners are actually used and how they actually run. It will show how well air conditioners are sized based on their operation, how often they cycle, and how they are used to control space temperatures. The paper will also address the commonly held belief that even moderate "oversizing" can lead to humidity problems.

## Introduction

Utilities and state energy offices publish equations for calculating savings from the installation of high-efficiency heating and cooling equipment such as air conditioners and heat pumps. An equivalent full-load hour (EFLH) value is a common variable in the energy savings equation. Savings are directly proportional to the EFLH value; as such, the derivation and accuracy of EFLH values should be generally understood by those who estimate weather-related heating and cooling energy savings. Once EFLH values are understood, one can better understand why published EFLH values, when used in energy savings equations, might overestimate energy savings for a typical residential population. The presentation and analysis of metered energy consumption and outdoor and indoor temperature and humidity recorded during a cooling season of a controlled sample of residential heating, ventilation, and air-conditioning (HVAC) systems will help illustrate the main reasons that published EFLH values are overstated. These data will also show the impact of oversizing on indoor temperature set point and humidity control, which are conditions that may cause a homeowner to use more cooling energy than necessary.

## Equivalent Full-Load Cooling Hours—Explained

Equivalent full-load cooling hours ($EFLH_C$) are the number of hours an air conditioner would have to operate at full load to equal the amount of cooling delivered by the system at a constant thermostat setting over a cooling season. Equivalent full load heating ($EFLH_H$) hours are analogous to $EFLH_C$; this paper focuses only on cooling.[1]

The product of the $EFLH_C$ and the system's actual capacity is the amount of cooling the system delivers in a cooling season, as shown in Equation 1.

Equation 1:

$$System\ Capacity\ \left(\frac{Btu}{hour}\right) \times EFLH_C \left(\frac{hours}{season}\right) = Cooling\ Delivered\ \left(\frac{Btu}{season}\right)$$

Utilities and state energy offices publish equations for calculating savings from the installation of an air conditioner with higher efficiency than a baseline system. Most of the equations are in the form of Equation 2, which is essentially Equation 1 times the difference of the reciprocals of the Seasonal Energy Efficiency Rating (SEER) values of a baseline unit and the replacement unit.

Equation 2:

$$\Delta kWh_{COOL} = System\ Capacity\ \times \mathrm{EFLH}_C \times \left(\frac{1}{SEER_{BASE}} - \frac{1}{SEER_{EE}}\right)$$

Where:

$\Delta kWh_{COOL} = Reduction\ in\ annual\ kWh\ consumption\ of\ cooling\ equipment$

$System\ Capacity = capacity\ of\ cooling\ equipment\ kBTU/hr$

$SEER_{BASE} = Seasonal\ energy\ efficiency\ rating\ of\ baseline\ cooling\ equipment$

$SEER_{EE} = Seasonal\ energy\ efficiency\ rating\ of\ high\ efficiency\ cooling\ equipment$

$\mathrm{EFLH}_C = Equivalent\ Full\ Load\ Hours$

There are several ways to calculate EFLH from metering data for a set of systems. Equation 3 is the simplest and only requires that unit energy consumption be logged for the cooling season.

Equation 3:

$$\mathrm{EFLH}_C = \frac{Total\ Seasonal\ Cooling\ Energy\ Consumption\ [kWh]}{Nameplate\ Rated\ Peak\ Demand\ [kW]}$$

If the cooling delivered is known (a quantity much trickier to meter than total power), Equation 4 yields a similar result. The advantage of this equation is that it is based on cooling capacity, which is typically how it is used (see Equation 2).

Equation 4:

$$\mathrm{EFLH}_C = \frac{Total\ Seasonal\ Cooling\ Delivered\ [BTU]}{Nameplate\ Rated\ Capacity\ [BTU/hr]}$$

Where metering data are not directly available or where the savings of a population of air conditioners across a region are desired, more general $EFLH_C$ values are often used. $EFLH_C$ has historically been published in a number of locations, including on the ENERGY STAR® site as part of their calculators (EPA 2016), in the Code of Federal Regulations (USFTC 2013), and in various TRMs. None of these sources describes the methodology used to derive EFLH values.

[1] Most TRMs use a modified form of Equation 2 to determine heating savings for a heat pump. When a TRM equation is used, one should consider that the heating seasonal performance factor (HSPF) includes an efficiency decrement because of assumed use of backup electric resistance heat. If the heat pump does not use electric resistance heat, a modification to both the HSPF and EFLH heating value should be considered.

For several geographic locations with published $EFLH_C$, we investigated whether early published values were based on simple weather data rather than detailed calculations or modeling. We assumed that $EFLH_C$ was based on the following equation:

Equation 5:

$$\mathrm{EFLH}_C = \frac{CDD\ x\ 24\ [degree\ hours]}{design\ temperature\ °F - 65°F}$$

Where:

$CDD = \text{summation of average daily temperature} - 65°\text{F}$
$Design\ temperature = 97.5\%\ \text{or}\ 99\%\ \text{peak design temperature}$

We found a similar equation in the previous version of the Arkansas TRM (APSC 2013) that appeared to confirm our belief. Table 1 shows heating and cooling full-load hour values for four different cities in Arkansas published in the Arkansas TRM.

Table 1. Equivalent full-load cooling/heating hours

| Weather Zone | Location | $EFLH_C$ | $EFLH_H$ |
|---|---|---|---|
| 9 | Fayetteville | 1,233 | 1,923 |
| 8 | Fort Smith | 1,493 | 1,793 |
| 7 | Little Rock | 1,669 | 1,682 |
| 6 | El Dorado | 1,647 | 1,474 |

To check the assumption, we derived the Little Rock $EFLH_C$ value using normal cooling degree days (CDD) and two different design temperatures.

| | |
|---|---|
| $T_{99}$ | = 95°F design temperature (ACCA 1986) |
| $T_{97.5}$ | = 96°F design temperature (ASHRAE 1985) |
| CDD | = 2,107 (years 1948–1990) (ACCA 2006) |
| $EFLH_C$ | = 2,107 x 24/ (95 – 65) = 1,686 |
| $EFLH_C$ | = 2,107 x 24/ (96 – 65) = 1,631 |

The derived values are just 17 hours higher and 38 hours lower than the published value, representing differences of about 1–2%. We again followed this methodology for other cities and found similar results. This close agreement is consistent with our belief that the derivation of traditionally published $EFLH_C$ is analytically derived from two general weather parameters. The small variance could result from a different period of averaging for CDD.

Results from metering studies (KEMA and Cadmus 2010; Navigant 2010; Navigant and Cadmus 2014; Walczyk et al. 2014) of air conditioners in various locations across the United States show that the actual mean $EFLH_C$ was, on average, 60–70% of the original published values. The following are likely reasons for this difference:

- *Variable usage due to vacancy.* A population of air conditioners will include those that are heavily, moderately, and lightly used. The $EFLH_C$ equation is based on an air conditioner used throughout the summer and operated to maintain a constant indoor temperature.
- *Variable weather patterns and manual thermostat control.* The $EFLH_C$ equation is based on the air conditioner running every day that the average daily temperature is above 65°F. Homeowners may wait for days after the first CDD are generated and shut their units off before the last CDD are generated in the fall.
- *Sizing practices.* The $EFLH_C$ equation is based on the air conditioner running at full capacity when the outside air temperature reaches the design temperature. Most air conditioners, as we will show, do not operate at full capacity when the outside temperature reaches the Air Conditioning Contractors of America or American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) design temperature. An HVAC contractor may purposely oversize a system simply because it is better to have excessive cooling capacity than insufficient cooling capacity. Or, an HVAC contractor may not correctly estimate the equipment size because of the complexity of estimating cooling load for a particular home (e.g., the thermal characteristic of the home is difficult to quantify). Additionally, proper sizing practice (ACCA 2006) assumes an indoor temperature setpoint of 75°F. Any thermostat setback or deviation from that temperature results in a change to the cooling load of the home.
- *In a heating-dominated climate, a heat pump is sized to meet the heating load.* If a home requires greater heating capacity than cooling capacity, a heat pump could be sized correctly to meet the heating load, but oversized for the cooling load.

The following sections provide examples of submetered data to show why the $EFLH_C$ is, on average, lower than published values. We also provide a summary of data from a controlled group of air conditioners to explore the impact that oversizing has on controlling indoor humidity.

**Evidence of Variable Usage on $EFLH_C$**

The EFLH equation is based on the assumption that an air conditioner operates to condition a space at a constant indoor temperature every day that the average daily temperature is above 65°F. Some homeowners do leave their thermostat at a constant temperature throughout the cooling season. Figure 2 shows outdoor temperature (blue line) and power (red line) of a ductless mini-split heat pump in the Northeast for the months of June, July, and August. For the majority of the cooling season, the homeowner maintained a constant indoor temperature setpoint of 75°F. The mini-split provided cooling for just over 600 hours during the three-month period.

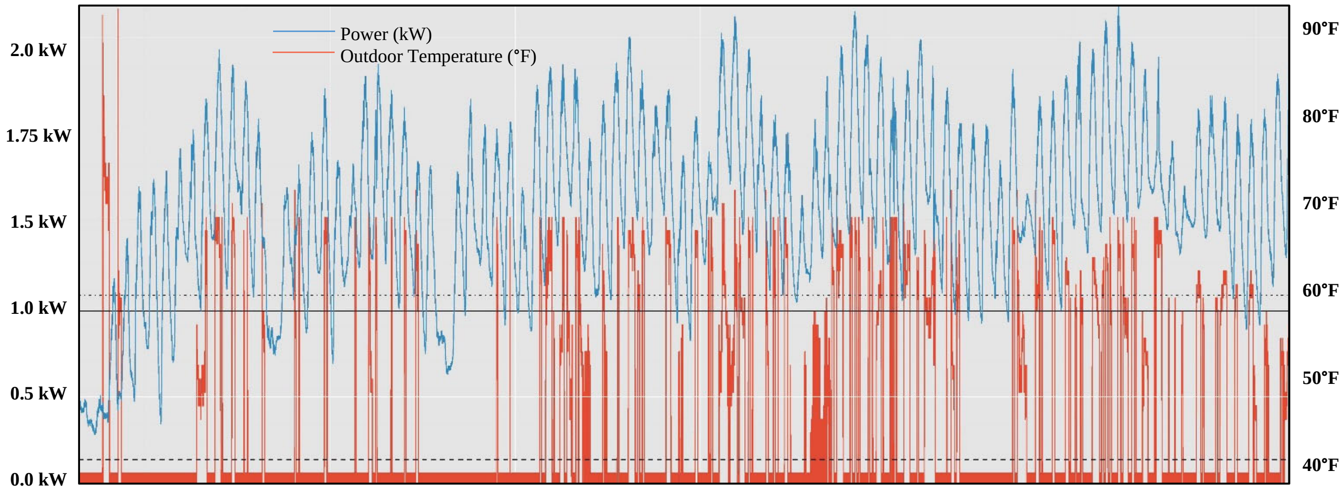


Figure 2. June–August operation of heat pump with nearly constant indoor temperature setpoint

Figure 3 shows another ductless mini-split heat pump at a different home from the same region metered during the same timeframe. During the three-month metering period, this system operated for a total of 130 hours.

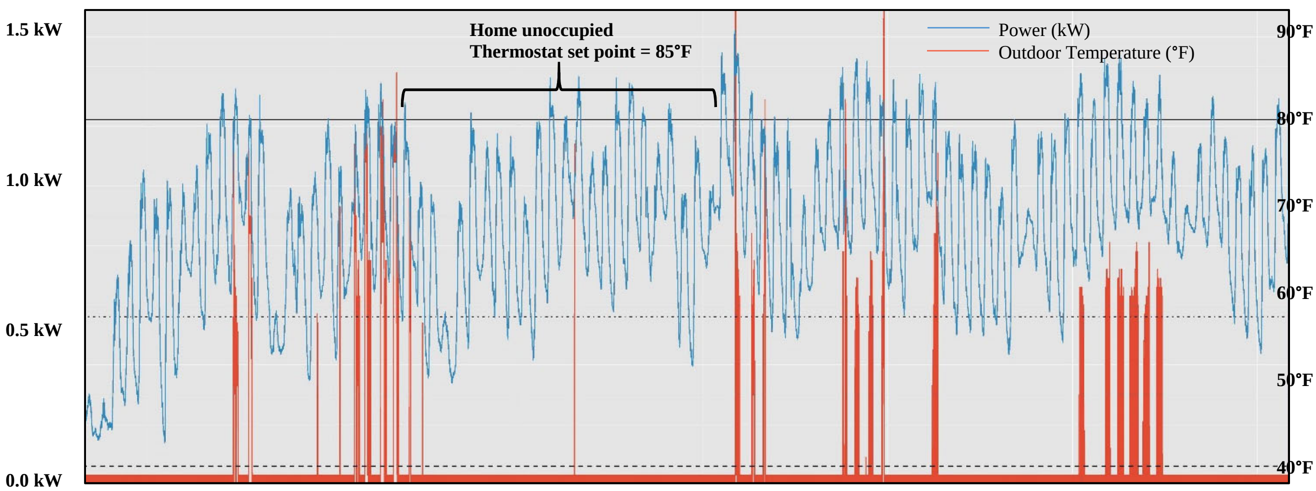


Figure 3. June–August operation of heat pump with variable usage

## Evidence of Oversizing

A general consensus about air conditioner sizing is summarized by a National Renewable Energy Laboratory report that states "typical design practice tends to result in oversizing (using a larger-than-needed unit). In general, the greater the oversizing, the fewer the operating hours, and the less efficiently a unit operates" (NREL 2000).

Numerous examples of submeter data indicate that units, on average, are oversized. This partially explains the reason that energy savings determined from submetering are consistently 60–70% of published values. The variable usage patterns described above are other probable reasons. We provide straightforward and obvious examples of both below.

The rationale that oversized systems result in less efficient operation compared to correctly sized systems is that oversized units have shorter cycle times. Shorter cycle time may decrease efficiency because

- the startup efficiency of an air conditioner is lower than the steady-state efficiency
- moisture is not removed from the air until condensed water physically drips from the indoor coil. Longer runtimes presumably result in more effective condensation and water removal. If indoor humidity is high, a homeowner might be more inclined to reduce the temperature setpoint to increase comfort; this would lead to increased energy usage.

Although conventional wisdom states that oversizing is a concern, some studies show mixed results when using metering data. A study of four homes in Florida where air-conditioning units were downsized and the units were metered pre- and post-installation saw no significant energy savings or humidity improvement across all four houses (Sonne, Parker, and Sirey 2006). According to Proctor (2010), "changes in the design of air conditioners, along with new research, call the traditional beliefs into question." Dual-speed, multi-speed, and variable refrigerant flow air-conditioning systems are able to change the speed of the compressor to reduce cooling capacity. For example, a 4-ton multi-speed central air conditioner can operate at low speed as if it is a 2-ton system. Lower capacity operation increases operating efficiency because of the effective increase in relative coil size (an increase in surface area increases the ability of the coil to reject and absorb heat). An oversized variable speed system might actually *increase* efficiency. Additionally, units operating in climates with low relative humidity are not constrained by a necessity to remove moisture.

To show the impact that system sizing has on humidity, we analyzed meter data of 60 air conditioners operating for an entire cooling season. This controlled sample includes only central air conditioners with single-speed compressors operating in the Midwest—a region with high temperatures and oftentimes high relative humidity. These systems were specifically selected because each maintained a nearly constant indoor temperature throughout the cooling season. The ASHRAE design temperature for the units selected for this analysis is 94°F and the weather-adjusted[2] EFLH value is 1,155 hours. Figure 4 shows the distribution of EFLH values by coincidence factor.[3] We determined a coincidence factor for each metered unit based on the amount of time it operated when the outdoor temperature was above 90°F. We used 90°F rather than the design temperature to ensure sufficient data were used.[4]

The coincidence factor is an indication of system sizing. A unit with a low coincidence factor at high outdoor temperatures is probably oversized for the space. Figure 4 compares the coincidence factor for all units when the outdoor temperature was above 90°F, with the $EFLH_C$ metered for each unit.

[2] Value weather-normalized by the ratio of hourly cooling degree days observed during the metering period to the hourly cooling degree days from 8,760 TMY3 data.

[3] Coincidence factor is the ratio of minutes that the system provided cooling to the total minutes that the local weather station observed each temperature.

[4] Only nine hours were observed at the design temperature during the summer. By including additional hours (90°F–~98°F), we gain nearly 10 times the metered data. This increases the reliability of estimates.

Average $EFLH_C$ for units with a +80% coincidence factor when outdoor temperature was above 90°F was 1,052 hours. Four units averaged coincident factors above 90% and these had an average EFLH value of 1,132 hours, which was nearly the same as the published ASHRAE hours for the region.

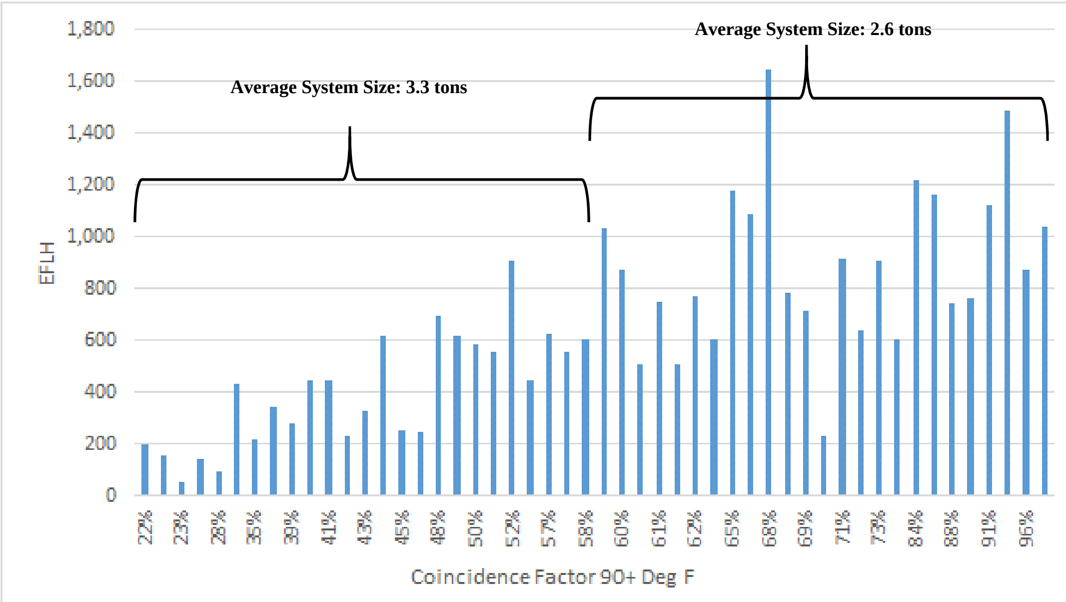


Figure 4. Coincidence factor and $EFLH_C$ values for 60 central air conditioners

The average $EFLH_C$ for all units was only 656 hours, 57% of the ASHRAE value. From Figure 4, it is evident that a large number of units operated less than one might expect when the outdoor temperature was above 90°F. From this, one might conclude that these units are oversized. The actual average system size further supports that assumption. Figure 4 shows that the average system size for units below a 60% coincidence factor was 3.3 tons. Units with coincidence factors higher than 60% averaged only 2.6 tons. We did not perform detailed load calculations of the homes, but the average conditioned square footage of each group was similar.[5]

## Oversizing Analysis: Indoor Humidity and Temperature Control

We might expect to see a lower indoor temperature for oversized systems because a homeowner might be inclined to decrease the indoor temperature to improve comfort (whereas, with better humidity control, one may tolerate a higher indoor temperature). To investigate this trend, we compared the average indoor temperature of all units to the coincidence factor when the outdoor temperature was above 90°F. We were unable to conclude whether homeowners decreased the indoor temperature because of a lack of humidity control. On the contrary, we see an increase in indoor temperature (see Figure 5). This increase, however, may simply show the impact that indoor temperature settings have on unit operation (coincidence factor). If contractors sized all units in exactly the same way, one would expect that a lower indoor temperature would result in longer runtimes.

[5] 1,950 $ft^2$ for group below 60% coincidence factor and 1,890 $ft^2$ for group above 60% coincidence factor

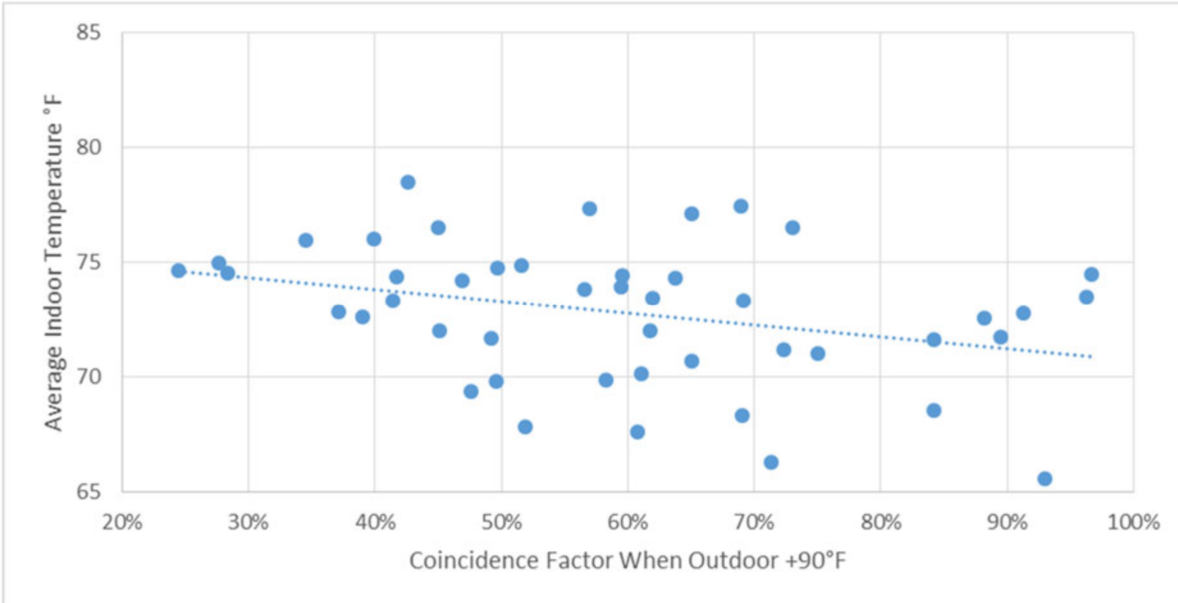


Figure 5. Average indoor temperature versus coincidence factor when outdoor temperature is +90°F

We also investigated indoor humidity specifically. Because the saturation temperature of air changes with temperature,[6] we grouped units by 2°F average indoor increments. Figure 6 shows the variance in relative humidity (orange columns) with increasing coincidence factor (blue columns). No obvious trends occur; therefore, we cannot definitively conclude from these that humidity control is impacted by system size.

[6] In other words, relative humidity alone is insufficient. For example, relative humidity of 55% at 75°F is very different (much more "muggy" feeling) from the same or even higher relative humidity at 70°F.

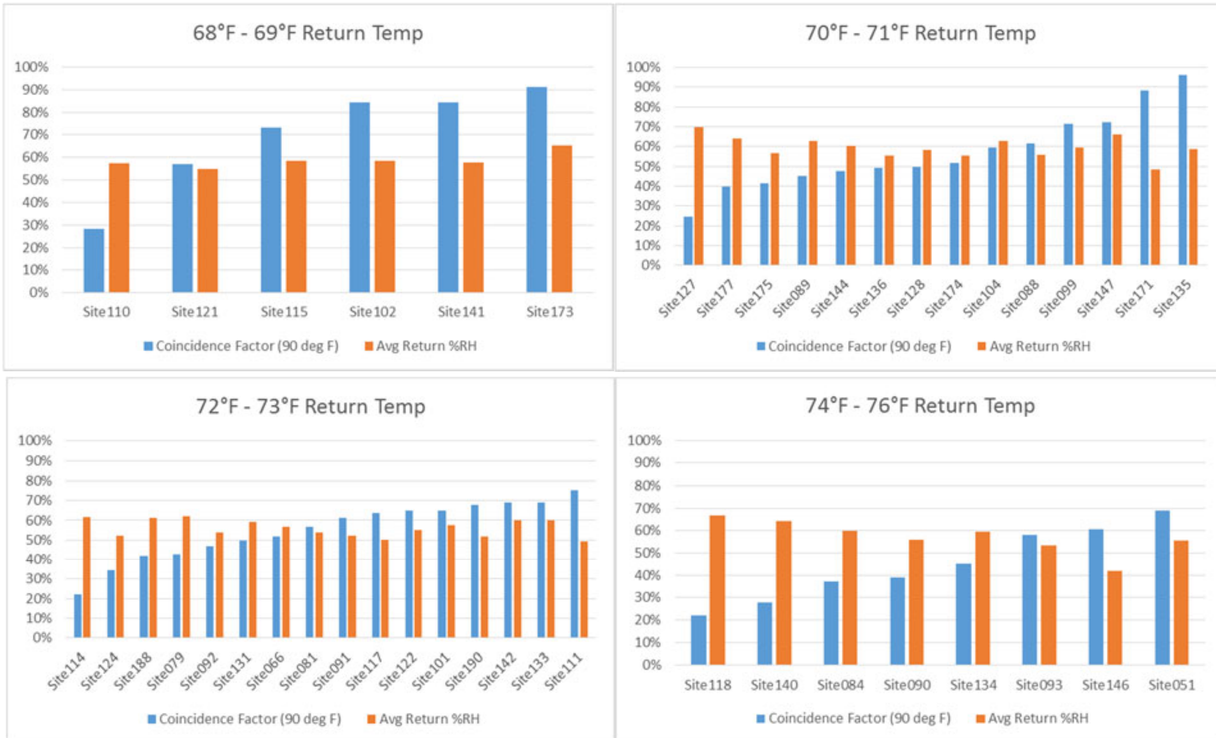


Figure 6. Variance of relative humidity with coincidence factor at different indoor temperatures

Figures 7 and 8 show the coincidence factor (blue dots) and cooling run time (red dots) for 2°F outdoor temperature bins for HVAC systems in two different homes.

The average indoor temperature of the homes was 69°F and 73°F in Figure 7 and Figure 8, respectively. At the design temperature (94°F), the system in Figure 7 had a coincidence factor of 93% (it ran almost continuously). The system in Figure 8 had a coincidence factor of about 55 at the design temperature.

The system in Figure 7 had a higher EFLH value because of the lower indoor temperature and because it was sized to operate near 100% at the design temperature. The site shown in Figure 8 had a higher indoor temperature setpoint and was sized to operate at about 50% when the outdoor conditions reached the design temperature. Consequently, there was nearly a factor of 3 difference in EFLH.

It is important to review the impact that a change in indoor temperature might have on the coincidence factor and EFLH. Both figures show relatively linear correlations of outdoor temperature with the coincidence factor[7]. This relationship is expected, assuming that heat transfer through the shell of a home is a simple function of temperature differential between inside and outside. We reviewed the slope of the lines and found that if the indoor temperature were raised to 75°F for the HVAC system in Figure 7, this system may use approximately 18% fewer EFLH.

[7] A small, unexpected drop occurs in the coincidence factor in Figure 7. Review of data indicates this is due to the decreasing number of hours, and consequential decreasing reliability of the metered data (i.e., fewer sampled intervals increases volatility of the average of the measurements).

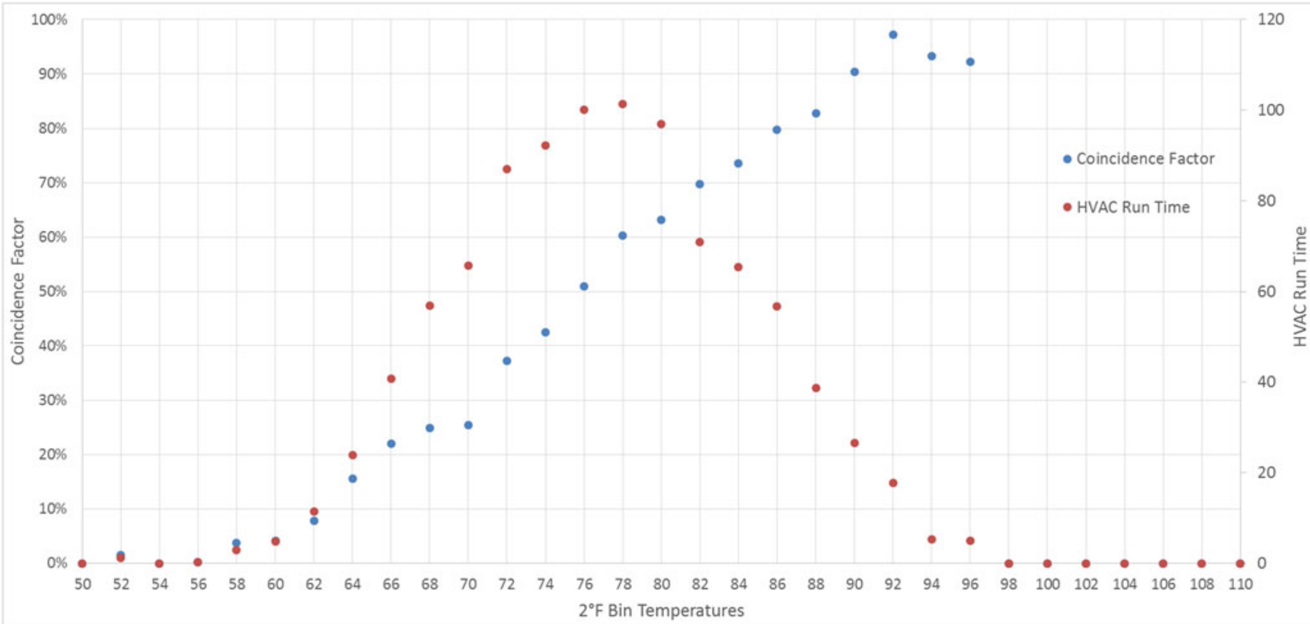


Figure 7. Site 102. Average indoor temperature: 69°F. $EFLH_C$: 1,218

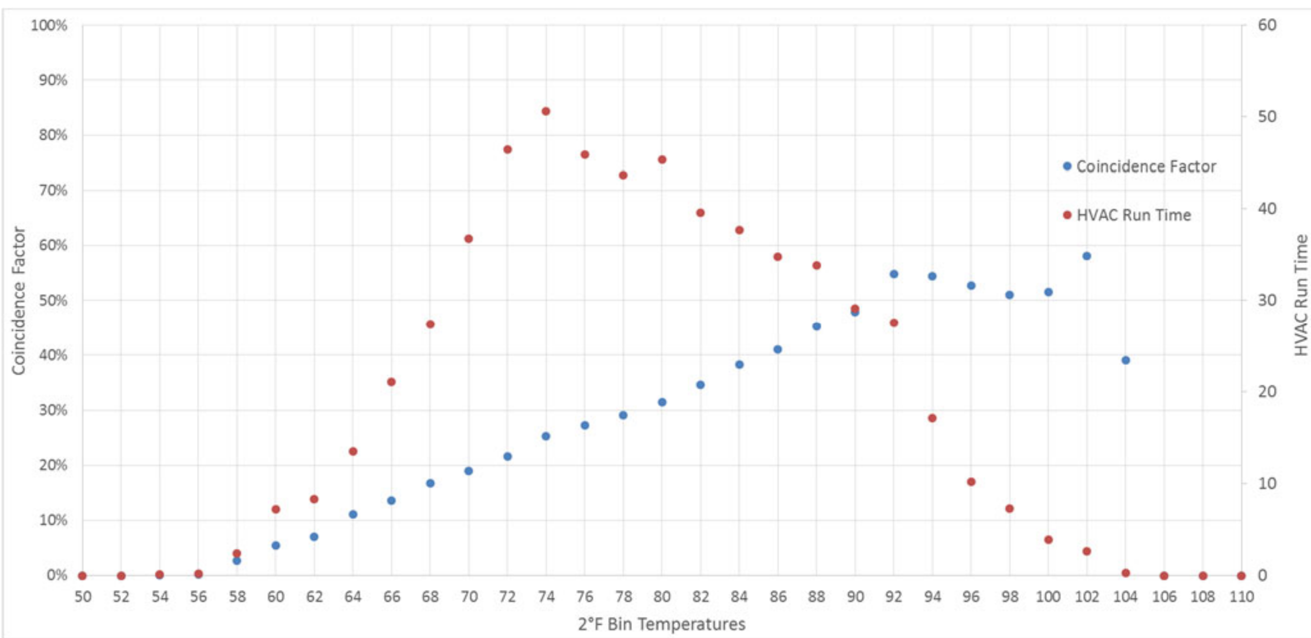


Figure 8: Site 174. Average indoor temperature: 73°F. $EFLH_C$: 448

## Summary

Many TRMs that establish savings that utility programs may claim use an equation of the form of Equation 2. The equation is typically applied to a population of air conditioners installed under a program where the units' capacity (size) and nameplate SEER are known. Historically, $EFLH_C$ values were high for the reasons described in this paper, but, in recent years, have been decreased based on modeling and metering efforts. The older values were correct or nearly correct for units sized just large enough to cool at the design temperature, but too large for a population of units that are sized larger than necessary to meet design loads or not used for the full season. As TRM values are dropped to better match previous metering efforts, they will match a population of mixed relative sizing and usage. However, if additional efforts are made to more carefully size air conditioners or to target programs to high users, these new, reduced $EFLH_C$ values may actually underestimate savings.

## Conclusions

Historically, published $EFLH_C$ values were derived from two climate factors: CDD and design temperature. The advantage of these values is that they are easily calculated and do not rely on HVAC system or home characteristics. The problems with these historical values, however, is that they mathematically imply that a unit is operating whenever CDD are generated—even during shoulder seasons—and that the units are designed to run continuously at 100% capacity at the outdoor design temperature.

We found that there was a large variety in the coincidence factors of continuously operated air conditioners above an outside temperature of 90°F, indicating varying and possibly inaccurate sizing methods. The size of the coincidence factor was directly correlated with the $EFLH_C$, and units varied by a factor of 3 in the range of the $EFLH_C$ values metered.

Conventional wisdom suggests that oversized air conditioners lead to indoor humidity problems. Using a population of 60 directly metered air conditioners, we compared indoor humidity to the operating coincidence factors, directly testing if we could see a difference in humidity in oversized units that ran at low frequencies (short cycle times) at high temperatures. We did not see any clear trend in increasing humidity with decreasing run frequency.

## References


Air Conditioning Contractors of America. (ACCA). 2006. *Manual J: Residential Load Calculation Eighth Edition*. http://www.acca.org/speedsheet/.

Air Conditioning Contractors of America. (ACCA). 1986. *Manual J: Seventh Edition, Residential Load Calculation*. Copyright 1986, Reprinted 2008. pp. 51 and 107.

American Society of Heating, Refrigerating and Air-Conditioning Engineers. (ASHRAE). 1985. *ASHRAE Handbook, 1985 Fundamentals*.

APSC (Arkansas Public Service Commission). 2013. *Arkansas Technical Reference Manual Version 3.0*. http://www.apscservices.info/EEInfo/TRM.pdf.

EPA (Environmental Protection Agency). 2016. *ENERGY STAR Air Conditioner, Central.* Savings Calculator. https://www.energystar.gov/sites/default/files/asset/document/CentralAC_Calculator.xls.

KEMA Inc. and Cadmus. 2010. *Evaluation Measurement and Verification of the California Public Utilities Commission HVAC High Impact Measures and Specialized Commercial Contract Group Programs. 2006–2008 Program Year*. http://www.calmac.org/startDownload.asp?Name=Vol%5F2%5FSpecialized%5FCommercial%5FReport%5FAPPENDICES%5F02%2D10%2D10%2Epdf&Size=1421KB.

National Renewable Energy Laboratory. (NREL). Right-Size Heating and Cooling Equipment. 2002. http://www.nrel.gov/docs/fy02osti/31318.pdf.

Navigant Consulting. 2010. *Final Evaluation Report: Central Air Conditioning Efficiency Services (CACES).* http://ilsag.org/yahoo_site_admin/assets/docs/ComEd_PY2_CACES_Evaluation_Report_2010-10-18.299122020.pdf

Navigant Consulting and Cadmus. 2014. *EmPOWER Maryland Final Evaluation Report Evaluation Year 4 (June 1, 2012–May 31, 2013) Residential HVAC Program.* http://www.neep.org/sites/default/files/resources/EmPOWER_EY4%20Res%20HVAC%20Impact%20Report_FINAL%2023Jun14_CLEAN.pdf.

Proctor, J. 2010. *Home Energy Magazine—Sizing Air Conditioners*. http://www.proctoreng.com/dnld/Home_Energy_Sizing_AC.pdf.

Sonne, J., D. Parker, and D. Sirey III. 2006. *Closing the Gap: Getting Full Performance from Residential Central Air Conditioners; Task 3.2: Benefits of Proper Sizing. STAC Solicitation #03-STAC-1*. http://www.fsec.ucf.edu/en/publications/pdf/FSEC-CR-1641-06.pdf.

United States Federal Trade Commission (USFTC). 2013. *Energy and Water Use Labeling Under the Energy Policy and Conservation Act.* 16 CFR Part 305. https://www.ftc.gov/policy/federal-register-notices/16-cfr-part-305-energy-and-water-use-labeling-under-energy-policy.

Walczyk, J., D. Korn, D. Bruchs, and S. Khawaja. 2014. *Ameren Missouri CoolSavers Impact and Process Evaluation: Program Year 2013*. https://www.efis.psc.mo.gov/mpsc/commoncomponents/viewdocument.asp?DocId=935842419.