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Two-Stage High Efficiency Air Conditioners: Laboratory Ratings vs. Residential Installation Performance

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Two-Stage High Efficiency Air Conditioners:

Laboratory Ratings vs. Residential Installation Performance

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ABSTRACT

The increased installation of high Seasonal Energy Efficiency Ratio (SEER) air conditioners along with utility program rebates for these units prompted a study of the measured performance of these systems. This project assessed the performance of these systems in the climate zones found in the mid-Atlantic region of the U.S. Similar studies in hot dry climates have indicated that laboratory SEER ratings may not properly predict the actual impact of these systems.

This project monitored four high SEER air conditioners with dual-stage compressors, TXV metering devices, and high efficiency air handlers with ECM fan motors. One system with a single-stage compressor was also monitored. Data included capacity, power consumption, EER, indoor/outdoor temperature and relative humidity. The data were analyzed to assess the relationship between laboratory testing and real world performance.

This study found causes for concern including: actual seasonal energy efficiency ratios between 59% and 84% of the rated SEERs, constant fan operation substantially degrading seasonal efficiencies and reducing dehumidification, latent loads that exceed Manual J estimates, and sensible loads substantially lower than Manual J estimates. In addition there may be an energy and peak load penalty if dual-stage air conditioners are downsized to near the buildings' actual loads.

The study illustrates the intricacy of the whole building system. The air conditioners in the two leakiest building shells were unable to adequately control the indoor humidity.

Introduction

Utilities and contractors across the United States have successfully promoted the installation of high SEER central air conditioning systems. High SEER systems are often equipped with dual-stage compressors and variable speed blowers. This study addressed the impact of these systems on the electric grid in DOE Climate Zones 4 and 5. The test houses were located in New Jersey and New York at the Northern edge of Climate Zone 4 and the Southern edge of Climate Zone 5. Both climate zones are considered "moist", where outdoor air often contributes to the latent load on the air conditioner.

A continuing question in the energy-efficiency community is how closely SEER ratings represent the actual seasonal efficiency of air conditioners in various climates. Given the increasing frequency of these installations and emphasis of utility programs on installing higher efficiency SEER equipment, it is important to quantify the impact of these high efficiency systems.

The purpose of this study was to compare the operating characteristics (peak demand, EER, and kWh) of high efficiency residential central air conditioning systems to projections from ratings including SEER and EER.

One of the objectives of this study was to compare dual-stage units against a high efficiency single speed unit. This study monitored four recently installed dual-stage air conditioners and one high efficiency single speed AC system.

Monitoring System

Each air conditioner was monitored by a data acquisition system (DAS). The DAS has the flexibility to perform many data acquisition functions and is capable of being downloaded or reprogrammed via modem. The temperature probes were bare wire 36 gauge type T thermocouples, RTDs, or thermistors. Condensate flow from the indoor coil was measured with the use of a tipping bucket gauge attached to the termination of the condensate drain. Data points are summarized in Table 1.

Table 1. Monitored Points

Measurement	Sensor Type	Sensor Location
Supply Air Dry Bulb Temperature	4 Point RTD Grid	After Coil In Supply Plenum
Supply Air Dry Bulb Temperature	Thermister	After Coil In Supply Plenum
Supply Air Dry Bulb Temperature	Thermocouple	After Coil In Supply Plenum
Supply Air Dry Bulb Temperature	Thermocouple	Far Supply Register
Supply Air Dry Bulb Temperature	Thermocouple	Near Supply Register
Supply Air Relative Humidity	Humidity Transmitter	With Supply Air Thermister
Return Air Dry Bulb Temperature	Thermister	Return Plenum Before Furnace
Return Air Dry Bulb Temperature	Thermocouple	Return Plenum Before Furnace
Return Air Dry Bulb Temperature	Thermocouple	Return Grill
Return Air Relative Humidity	Humidity Transmitter	With Return Air Thermister
Buffer Space Dry Bulb Temperature	Thermocouple	Buffer Space
Outside Air Temperature	Thermister (Shielded)	Outside Near Condensing Unit
Outside Air Relative Humidity	Humidity Transmitter	With Outside Air Thermister
Indoor Air Temp	Thermister	Near Thermostat
Compressor Discharge Temperature	RTD	Surface Mounted To Compressor Gas Discharge Line (Insulated)
Refrigerant Liquid Line Temperature	RTD	Surface Mounted To Liquid Line After Condenser Coil (Insulated)
Condenser Saturation Temperature	2 Thermocouples	Surface Mounted to Condenser Refrigerant Circuit
Evaporator Saturation Temperature	2 Thermocouples	Surface Mounted to Evaporator Refrigerant Circuit
Vapor Suction Line Temperature	RTD	Surface Mounted To Suction Line Before Compressor (Insulated)
Evaporator Condensate Flow	Tipping Bucket	Evaporator Condensate Line
Compressor Current	Current Transducer	Compressor Power lines
Furnace Blower Current	Current Transducer	Blower Power Lines
Furnace Gas Valve Current	Current Transducer	Gas Valve Control Lines
Condensing Unit Power	Watt Transducer	Electrical Supply To Unit
Furnace Blower Power	Watt Transducer	Electrical Supply To Furnace Unit

Site Descriptions

The characteristics of the homes and air conditioners in the project convenience sample are listed in Table 2.

Table 2. Site Characteristics

House Specifications					
	Site P[1]	**Site S**	**Site T**	**Site N**	**Site W**
House Size (square feet)	1620	2375	1900	3500	2300
Year Built	2002	2000	1940s	1960s	1970s
Manual J7 Cooling Load (Btuh) at 90/75/63 no window coverings	22118	36342	27513	56462	29549
Manual J7 Cooling Load (Btuh) at 90/75/63 with drapes	18180	29471	22851	44106	23864
House Tightness: Air Changes per Hour (ACH50)	3.9	2.8	5.5	7.4	12.7
Air to Air Heat Exchanger Flow (% of airflow)	17%	12%	none	none	none
% of Time Indoor RH >60%	2%	13%	0%	40%	35%
Air Conditioner Specifications					
Rated SEER	14.25	14	15	14	14
High Speed Rated EER at 95/80/67	10.3	10.7	9.4	10.8	11.5
Low Speed Rated EER at 95/80/67	None	12.2	11.7	12.4	12.6
High Speed Rated Capacity at 95/80/67 (Btuh)	34900	46080	34300	48230	48230
Low Speed Rated Capacity at 95/80/67 (Btuh)	None	25260	24700	27180	27180
Number of Compressor Speeds	1	2	2	2	2
Metering Device	Fixed	TXV	TXV	TXV	TXV
Fan Motor Hp	1/2 Hp	1 Hp	1 Hp	1 Hp	1 Hp
Fan Motor Type	ECM	ECM	ECM	ECM	ECM
Fan Mode (TD = Time Delay IO = Instant Off)	TD	Const.	TD	IO	Const.

1. ARI ratings are not available for the Site P system combination (using a third party evaporator coil). The estimated rated SEER, EER, power, and capacities are for a manufacturer's combination with the same nominal capacity.

House Characteristics

All the sites were 2 story homes with furnaces and ducts in the basement. Site P is a new modular home and Site S is a new Energy Star home.

The homes were tested for air leakage using a single point (50 pascals) blower door test. There was a large variation in the measured air leakage (2.8 ACH50 to 12.7 ACH50). Sites N and W were the leakiest homes and the dual-stage/variable airflow air conditioners were unable to adequately control the inside relative humidity.

The cooling loads were calculated using Manual J7 at an inside/outside temperature difference of 15 °F, with an infiltration rate estimated from the blower door test, with latent and sensible loads calculated independently, with the swing factor set to 1, and with no window coverings. Site N has a Manual J 7 estimated load that exceeds the nominal tonnage of the installed air conditioner – an unusual situation.

Aggressive Manual J7 load estimates were also calculated based on interior drapes on all the windows.

Air Conditioner Specifications

The air conditioners represented a narrow band of efficiencies from 14 to 15. The units were three-ton and four-ton units. There were three different types of furnace fan operation observed in these units: Constant on – this produces the minimum latent capacity since moisture on the coil at the end of the compressor cycle is evaporated back into the house air; Time delay – this is the most common fan control which is designed to maximize the SEER of the unit; and Instant off – the fan control which should produce the most latent cooling (moisture removal). Two homes used a constant fan. One home used the constant fan to circulate air around musical instruments, pianos, cellos, etc. The other home began using a constant fan part-way through the summer for an unknown reason.

Results

Cooling Season results are tabulated by 5ºF temperature bins and compared to the manufacturer's ratings. Tables 3 and 4 display the results for the 80ºF to 85ºF temperature bin. Complete results are available in Proctor, Cohn & Conant 2006.

End of Cycle Performance

The End of Cycle (EOC) data are obtained by taking the sensor measurements from the last full minute of compressor operation for each cycle. This point is used to compare performance to the manufacturers' steady state ratings because it is the point in the cycle closest to steady state operation. The results are displayed in Tables 3 and 4.

Low Speed. Sites P and T do not have low speed data because the former is a single speed machine and the latter dual speed machine that ran only on high speed. At low speed the other three units achieved 89% or better of the rated capacity and their input power exceeded the rated values by 10% or more. The unit at Site N approached the rated capacity in this and other temperature bins.

Table 3. Low Speed Performance

	Site P	Site S	Site T	Site N	Site W
Average Outside Temperature (deg F)	NA	82.3	NA	82.3	82.1
Average Return Drybulb Temperature (deg F)	NA	70.2	NA	71.1	73.7
Average Return Wetbulb Temperature (deg F)	NA	63.8	NA	61.9	64.9
Average Cycle Length (min)	NA	22.8	NA	62.1	64.7
Number of Cycles	NA	414	NA	26	184
Capacity					
End of Cycle Net Capacity (Btuh)	NA	21529	NA	29230	24972
Mfr. Steady State (SS) Net Capacity (Btuh)	NA	24333	NA	26855	26754
% of Mfr. Steady State Net Capacity	NA	88%	NA	109%	93%
End of Cycle Net Sensible Capacity (Btuh)	NA	13483	NA	17225	17131
Mfr. SS Net Sensible Capacity (Btuh)	NA	15080	NA	18415	18837
% of Mfr. SS Net Sensible Capacity (Btuh)	NA	89%	NA	94%	91%
Input Power					
Total End of Cycle Input Power (W)	NA	2011	NA	1943	2037
Mfr. Steady State Input Power (W)	NA	1792	NA	1692	1860
% of Mfr. Steady State Input Power	NA	112%	NA	115%	110%
EER					
End of Cycle EER	NA	10.72	NA	15.09	12.27
Mfr. Steady State EER	NA	13.59	NA	15.88	14.39
% of Mfr. Steady State EER	NA	79%	NA	95%	85%

High Speed. Site P, the single speed machine has performance inconsistent with other single speed ACs field monitored by the authors (Proctor 1998, 5). Field monitored efficiencies for single speed air conditioners with proper charge and airflow generally perform within a few percent of the manufacturer's published data. The EER of this unit was only 64% of the rating in this temperature bin. We found no explanation for this discrepancy during the study.

Sites N and W performed well at high speed with capacities and watt draws close to expectations based on manufacturers' published performance data.

Table 4. High Speed Performance

	Site P	Site S	Site T	Site N	Site W
Average Outside Temperature (deg F)	82.4	82.3	82.6	82.3	82.1
Average Return Drybulb Temperature (deg F)	72.5	70.2	71.9	71.1	73.7
Average Return Wetbulb Temperature (deg F)	64.7	63.8	63.7	61.9	64.9
Average Cycle Length (min)	14.5	9.2	128	10.3	29.6
Number of Cycles	366	186	9	9	52
Capacity					
End of Cycle Net Capacity (Btuh)	23560	32933	27445	46822	46161
Mfr. Steady State (SS) Net Capacity (Btuh)	33685	40964	32488	44382	42436
% of Mfr. Steady State Net Capacity	70%	80%	84%	105%	109%
End of Cycle Net Sensible Capacity (Btuh)	17018	19560	22777	28005	30823
Mfr. SS Net Sensible Capacity (Btuh)	20964	23900	24954	27692	27471
% of Mfr. SS Net Sensible Capacity (Btuh)	81%	82%	91%	101%	112%
Input Power					
Total End of Cycle Input Power (W)	2902	4375	2627	3909	4127
Mfr. Steady State Input Power (W)	2647	4423	2719	4164	3988
% of Mfr. Steady State Input Power	110%	99%	97%	94%	103%
EER					
End of Cycle EER	8.13	7.54	10.45	12.05	11.19
Mfr. Steady State EER	12.74	9.27	11.95	10.66	10.64
% of Mfr. Steady State EER	64%	81%	87%	113%	105%

Cycle Performance

Two types of analysis are necessary to evaluate cycle performance. For units with a furnace fan that stops with the compressor or after some time delay, the cycling performance is simply the capacity from fan-on to fan-off divided by the power expended during that time. This applies to Sites P, T, and N.

For Sites S and W, the systems do not have off cycles because the occupants run the fan continuously during the cooling season. There are thus two sections to the cycle efficiency, the compressor section and the fan-only section. The fan-only section produces sensible capacity, but produces negligible net capacity. The fan-only sensible capacity is a result of the water being evaporated off of the coil, producing negative latent capacity. Compressor cycles are defined by the interval between compressor-on and compressor-off. This includes either or both compressor speeds. The cycle EER includes the power and capacity for both sections calculated as:

$$Cycle\ EER = \frac{Compressor\ Cycle\ Capacity \times \%\ of\ time\ on + Fan\ Cycle\ Capacity \times \%\ of\ time\ off}{Compressor\ Cycle\ Power \times \%\ of\ time\ on + Fan\ Cycle\ Power \times \%\ of\ time\ off}$$

Table 5. System Cycle Performance

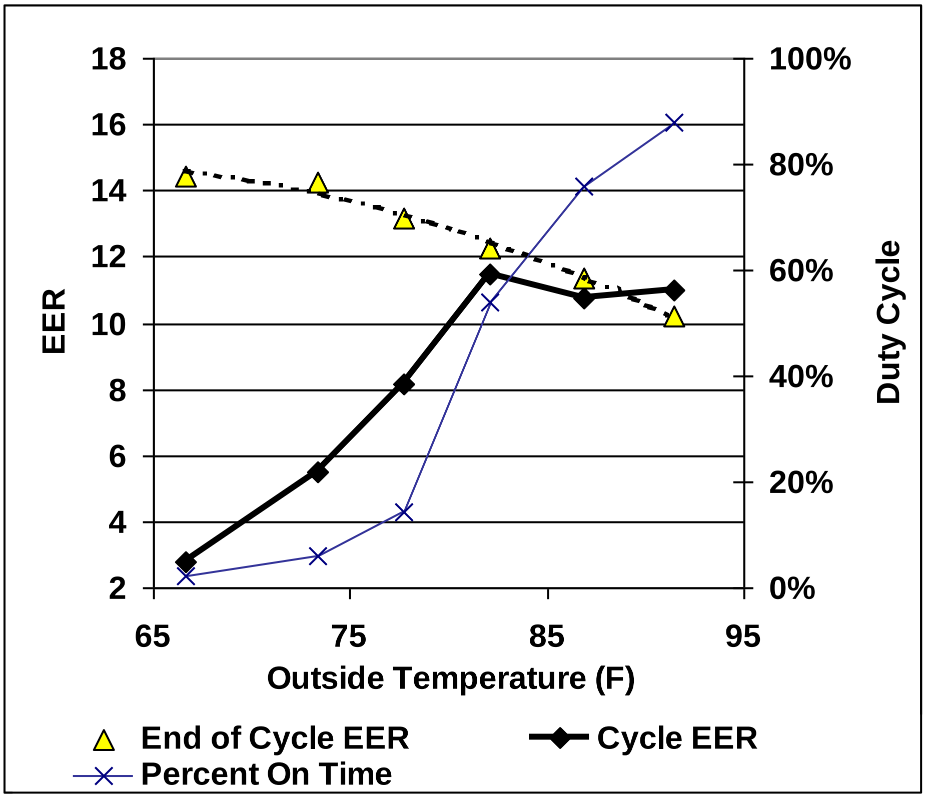
	Site P	Site S	Site T	Site N	Site W
Average Outside Temperature (deg F)	82.4	82.3	82.4	82.3	82.1
Mean Cycle Length (minutes)	14.5	19.9	162	112.8	97.4
Mean Off-Cycle Length (minutes)	16.7	13.8	214	70.7	82.5
% of Time on High Speed	Single S	15%	100%	5.40%	10%
Number of Cycles	366	610	15	23	145
Capacity					
Cycle Net Capacity (Btuh)	17550	23767	28586	29648	26751
Cycle Sensible Capacity (Btuh)		14005	22382	16460	17511
Fan Cycle Sensible Capacity (Btuh)		2157			2934
Input Power					
Mean Cycle Input Power (W)	2266	2628	2606	1986	2185
Fan Cycle Input Power (W)		136			176
EER					
End of Cycle EER	8.13	10.72	10.45	15.09	12.27
Cycle EER	7.75	8.88	10.97	14.98	11.46
% of EOC EER	95%	83%	105%	99%	93%

Table 5 shows while, in some cases, the steady state performance did not reach expectations, the cycling efficiency was very close to the steady state efficiency.

Continuous Fan. The units with the highest cycling losses are the two units that use a constant fan (Sites S and W). Low cycle efficiency is caused by the continuous fan energy consumption with little or no capacity. With a constant fan, the watt draw of the fan during the compressor off periods dominates the efficiency equation. As shown in Figure 1, only when the compressor on times (duty cycles) increase significantly do the cycle efficiencies begin to approach the low speed steady state efficiency.

It should be noted that continuous fan operation is often recommended by contractors to “mix the air” or for filtration. However it should not recommended – particularly in humid climates where it returns moisture from the coil to the home – a loss of latent capacity (Shirey, Henderson & Raustad 2006; Proctor & Pira 2005). In the average home with standard air distribution losses, the use of a continuous fan increases distribution losses substantially.

Figure 1. Cycle EER Degradation from Continuous Fan (Dual-Stage Machine)



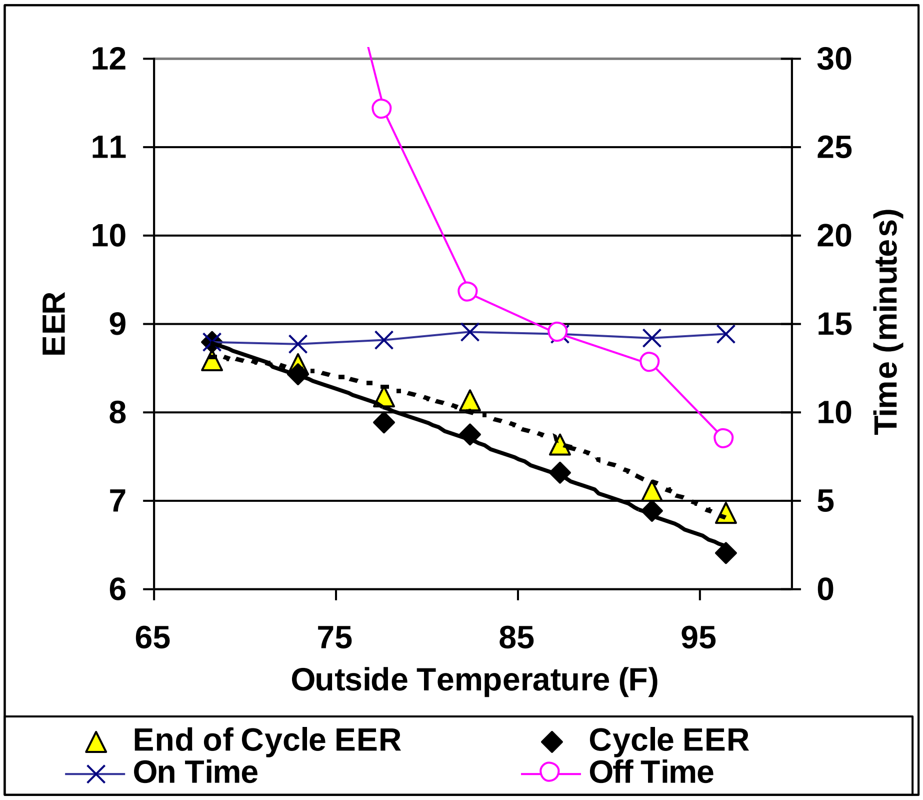
Dual-Stage Units. The two dual-stage units with fan-off at or near compressor off show little or no cycling degradation. The lack of degradation can be interpreted to indicate that there is little if any savings available for downsizing these dual-stage units. This is consistent with the long cycle times that minimize the startup losses and minimize the effect of the fan only "tail", which can provide a positive efficiency boost in dry climates. Downsizing the dual-stage machines would cause them to run more in the lower efficiency high-speed mode. One remaining question is the interaction of the dual-stage equipment efficiencies with the distribution system efficiencies that change when the equipment changes speed.

Single Speed Unit. For the single speed machine (Site P), the 5% efficiency drop may be an indication of a small potential savings from downsizing that unit. Figure 2 shows that the single speed (Site P) cycling performance conforms closely to its steady state performance over a range of temperatures.

The results of oversizing single speed units are increased electrical peak and, in some cases, insufficient dehumidification and increased energy consumption. Using a combined thermostat, air conditioner, and building simulation model, one study estimated that an AC system oversized by 50% would use 9% more energy than a properly sized system (Henderson 1992). A regression model based on data from 308 of the Florida field test of 368 homes estimated energy penalties of 3.7% and 9.3% respectively for units 20% and 50% oversized. In addition, homes with systems greater than 120% of Manual J averaged 13% greater peak cooling electrical load than homes without oversized systems (James et al. 1997). An empirical analysis of closely monitored units produced similar energy savings

(3.1% and 8.7% respectively for units downsized 23% and 47%) and a 12% peak reduction for a 31% downsizing (Proctor & Pira 2005). With the exception of the Florida field test, the estimates are based on models that were developed from a limited number of laboratory or field tests.

Figure 3. Full Cycle EER vs. End of Cycle EER (Single Speed Machine)



Cooling Loads

Table 6 compares the Manual J estimated cooling loads to the measured cooling loads (sensible and latent energy removed from the duct system by the unit at design conditions). Total loads in were calculated based on the change in enthalpy between the return plenum and the supply plenum using the formula:

Total Capacity (Btuh) = 4.5 x CFM x ΔBTU/LB

Sensible loads were calculated based on the change in dry bulb temperature between the return plenum and the supply plenum sensor grid using the formula:

Sensible Capacity (Btuh) = 1.08 x CFM x ΔT (ºF)

Both sensible and total loads were binned in 5 degree outdoor temperature bins. Least squared error second order equations were fit to the data and the equations solved for design conditions.

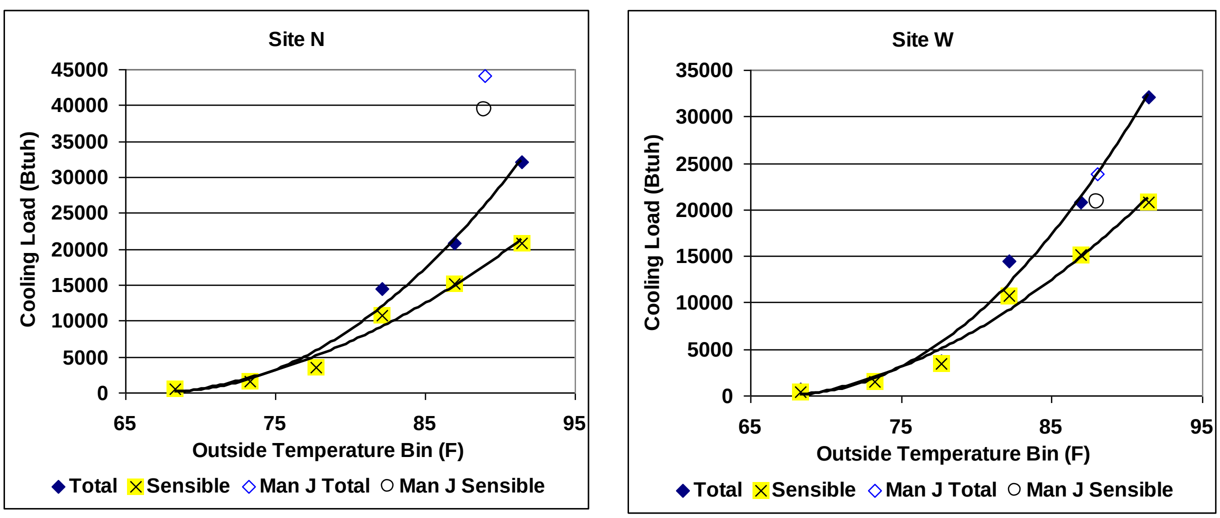
Table 6. Cooling Loads

	Site P	Site S	Site T	Site N	Site W
Design Cooling Loads					
Manual J7 Cooling Load (Btuh) at 90/75/63 no window coverings	22,118	36,342	27,513	56,462	29,549
"Aggressive" Manual J7 Cooling Load (Btuh) at 90/75/63 with drapes	18,180	29,471	22,851	44,106	23,864
Measured Total (Btuh)	8,381	15,702	15,026	34,565	23,660
Actual/Aggressive Total MJ7(%)	46%	53%	66%	78%	99%
Manual J7 Estimated Sensible (Btuh) no window coverings	19,298	24,044	30,862	51,631	27,010
"Aggressive" Manual J7 Sensible Cooling Load (Btuh) with drapes	15,360	19,382	23,992	39,275	20,841
Measured Sensible (Btuh)	6,687	10,648	12,395	15,767	16,244
Actual/Aggressive Sensible MJ7 (%)	44%	55%	52%	40%	78%

Table 6 shows that the latent loads were significantly higher than the Manual J estimates in the three leakiest homes even though the infiltration estimates were based on blower door tests[1]. In all cases the sensible loads of all four homes were substantially less than the Manual J7 estimates. The differences between the Manual J estimates of sensible loads and the actual sensible loads are higher than those found in previous studies in hot dry climates (Proctor 1998, Proctor et al. 1996). The site that most closely approached the Manual J7 total load estimate was Site W, which had a constant fan, high house air leakage, and wall cavity returns in virtually every room.

Figure 4 shows the loads and Manual J estimates for Sites N and W, the two sites where the estimates most closely approached the measured total loads.

Figure 4. Sites N & W Loads vs. Aggressive Manual J7



[1] Estimated infiltration rates based on blower door tests have been presumed to be more accurate than estimates based on a subjective rating system of "loose, average, or tight".

Peak Energy Use

Peak watt draws between 4:00-5:00 pm and 5:00-6:00 pm are tabulated in Table 7 from the 10 cooling days with the highest power draw as well as the single day with the maximum power draw. In this table, the input power is the average power in the given hour. The input power is less than the full connected load if the unit cycles during the hour or if it runs on low speed for some part of the hour. The worst-case scenario for utility peak demand is if the unit runs at connected load for the hour. This adverse condition normally occurs when the delivered capacity of the air conditioner is less than the load or when the occupants' change the thermostat setting to a lower temperature. Large scale studies have shown that, at peak, there are a group of homes (14% to 36%) running at full connected load, another group of homes with the air conditioner cycling (44% to 85%) and a third group of homes with the air conditioners not running at all (1% to 21%). The local mix of these groups determine the diversified peak performance of air conditioners and the efficacy of measures designed to reduce peak (Peterson and Proctor 1998).

Table 7. Summer Peak Demand

	Site P	**Site S**	**Site T**	**Site N**	**Site W**
Peak kW (10 highest)					
4 to 5 PM Average Power (Wh/hr)	1592	2791	2631	2243	3439
4 to 5 PM Duty Cycle	66%	84%	100%	100%	100%
5 to 6 PM Average Power (Wh/hr)	1714	2517	2528	1058	3122
5 to 6 PM Duty Cycle	70%	79%	99%	53%	100%
Peak kW (single highest)					
4 to 5 PM Average Power (Wh/hr)	1749	3490	2851	2674	3959
4 to 5 PM Duty Cycle	67%	80%	100%	100%	100%
5 to 6 PM Average Power (Wh/hr)	2709	2973	2647	2036	3855
5 to 6 PM Duty Cycle	97%	100%	100%	95%	100%
Connected Load (W at High Speed)	3225	4679	2820	4200	4390

One of the sites (Site T) fell into the full connected load group in the highest days. This was caused by thermostat manipulation by the occupants. A second site (Site P) approached full connected load only in the single highest peak day. These two units illustrate the potential peaking problem with oversized single speed machines.

All the units that were operating as true dual-stage machines (S, N, and W) operated at less than full connected load, switching between high and low speed, during the peak hours.

Seasonal Efficiency

Seasonal Energy Efficiency and consumption were calculated using TMY-2 temperature bins for locations selected for proximity, similar latitude, and similar distance inland. The capacities and input powers were averaged for all cycles in each temperature bin for Seasonal Efficiency calculations. Each site is compared to average central air conditioned homes in the Middle Atlantic region by AC Energy Intensity [kWh/sq.ft.] (EIA 2001). The results of that analysis are shown in Table 8.

Table 8. Seasonal Energy Efficiency and Consumption

Site	Site P[1]	Site S	Site T	Site N	Site W
Seasonal Performance					
Rated SEER	14.25	14	15	14	14
Measured Seasonal BTU/Wh	7.92	8.6	11.5	11.7	8.25
Seasonal BTU/Wh without constant fan		9.9			12.2
Seasonal kWh	1445	2351	2777	2168	1870
Seasonal kWh without constant fan		1986			1299
Average Seasonal kWh/sq.ft. in Region	0.63	0.63	0.63	0.63	0.63
Site Seasonal kWh/sq.ft.	0.89	0.99	1.46	0.62	0.81

1. ARI ratings are not available for the system combination (using a third party evaporator coil). The estimated rated SEER, EER, power, and capacities are for a manufacturer's combination with the same nominal capacity.

In all cases the seasonal efficiency was less than the rated SEER. Sites S and W would both be substantially more efficient without the continuous fan. All homes except Site N had an AC Energy Intensity (seasonal air conditioning kWh per conditioned square foot) that exceeded the average for the Mid Atlantic Region (EIA 2001).

Site P The calculated seasonal efficiency based on the TMY-2 data in combination with the monitored data is 7.92, which is 53% of the rated SEER of 14.25. One reason for the large discrepancy may be the use of an aftermarket evaporator coil with an inferred rating.

Uncertainty

Sources of uncertainty in the analysis include the airflow, air temperature measurement, humidity measurement, and electrical power measurement. Confidence in temperature rise and electrical power measurements is high. Humidity and airflow measurements are the largest sources of uncertainty.

Estimated supply and return air temperature measurement accuracy is ± 1°F. The supply/return air differential was calibrated to less than 1°F. Electrical power was measured with Ohio Semitronics watt-hour transducer rated as accurate to 2%. The relative humidity sensors have a rated accuracy of ± 2% for relative humidity between 10% and 90%. Relative humidity in the supply plenum often approaches 100% and is difficult to measure accurately. We estimate our supply humidity measurements to be accurate to ± 4%, not including the effect of response time. Airflow was measured with Energy Conservatory's True Flow® flow grid. The flow grid is rated as accurate to within 8%.

We estimate the confidence interval in the total capacity and efficiency statistics at about ±15%. We estimate the confidence interval in sensible capacity and efficiency statistics at about ±10%. We estimate the confidence interval in the energy consumption at about ± 2%.

Summary and Conclusions

Site N, the 1960s 3500 square foot home, was the best performing home from an energy and peak perspective. This home had a constant thermostat setting. It also had the second highest air leakage and, as a result, the properly sized, dual-stage, variable blower speed air conditioner with instant fan off was unable to adequately control the moisture in the home.

This home had the smallest capacity air conditioner per square foot and per measured cooling load. Nevertheless the air conditioner was still sufficiently oversized that it operated on a combination of high and low speed during peak hours. This site also had the most efficient unit turning in a cycle efficiency of 15 Btu/Wh in the 80ºF to 85ºF temperature bin. This cycle efficiency is 99% of the actual low speed EER and 95% of the manufacturer's rating.

Site P, the single speed machine has performance inconsistent with other single speed ACs field monitored by the authors (Proctor 1998, 5). Field monitored efficiencies for single speed air conditioners with proper charge and airflow generally perform within a few percent of the manufacturers' published data[2]. The EER of this unit was only 64% of the rating in the 80 ºF to 85 ºF temperature bin.

The two sites (S and T) with constant fan display a variety of problems expected with that type of fan control (Shirey, Henderson, & Raustad 2006). The seasonal energy efficiencies of these two units were substantially degraded, the cycling efficiencies were low compared to their end of cycle efficiencies, and the dehumidification was compromised.

The two sites (N and W) with the leakiest building shells had excessive indoor humidity that was not adequately controlled by the air conditioners. It is likely that Site W had significant duct leakage which, when combined with constant fan, exacerbated the humidity problems.

The dual-stage units with instant fan off or slightly delayed fan off showed little cycling degradation indicating that reduced AC sizing may have little effect on the seasonal energy consumption. In fact there is a concern that reducing the size of these units would cause them to run at their less efficient high speed more of the time and increase both energy consumption and peak watt draw.

The design conditions sensible loads at every site were between 40% and 78% of the Manual J estimates. On the other hand, the latent loads at the three leakiest buildings were higher than the Manual J estimates.

The dual-stage units were producing actual seasonal energy efficiency ratios between 59% and 84% of their rated SEERs. In addition, all the sites used a more cooling kWh per square foot than the EIA published average for the Middle Atlantic Region. The Energy Star rated home was the next to worst performing home of the group based on air conditioner electrical consumption intensity.

[2] When the manufacturers' published data are corrected for actual fan watt draws.

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