FLORIDA SOLAR ENERGY CENTER®

CONTRACT REPORT

Measured Impacts of Proper Air Conditioning Sizing in Four Florida Case Study Homes

STAC Solicitation #03-STAC-1
Closing the Gap: Getting Full Performance from Residential Central Air Conditioners
Task 3.2: Benefits of Proper Sizing

Final Report
FSEC-CR-1641-06

October 25, 2006

Submitted to:

National Association of State Energy Officials
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Alexandria, VA 22314

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A Research Institute of the University of Central Florida

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TABLE OF CONTENTS

EXECUTIVE SUMMARY

Previous studies have shown that residential air-conditioning system oversizing is a common practice that has both energy and comfort penalties. A Florida Power and Light / Florida Solar Energy Center (FSEC) study conducted in the mid-1990s involving over 350 homes showed that 50% of residential AC systems are oversized by 120% or more (James et al. 1997). The same study found oversizing AC systems by 1.0 to 1.2 times Manual J resulted in 3.7% higher cooling energy use and oversizing systems 1.2 to 1.5 times Manual J resulted in 9.3% higher cooling energy use.

The goal of Task 3.2, Benefits of Proper Sizing of the State Technologies Advancement Collaborative (STAC) project, was to demonstrate the benefits of proper air conditioner sizing to contractors, customers and utilities. Field tests were conducted in four Florida case study homes. Testing the benefits of properly-sized AC systems was accomplished via a pre/post monitoring study. Indoor air conditions and AC energy use in the four project homes were monitored during the summer of 2004 with the original, oversized AC systems. Then the AC systems were changed out with properly-sized systems (according to a strict interpretation of the Air Conditioning Contractors of America Manual J procedure) and conditions and energy-use monitoring continued with the new systems in place.

While three 2004 Florida hurricanes significantly limited the post-change out monitoring, useful comparisons in three of the four project homes were still possible. Analysis of the pre- and post-change out monitored data showed mixed energy savings and relative humidity results. In a Jacksonville project house, AC energy use was approximately the same and indoor relative humidity (RH) levels were slightly higher with the properly-sized AC system compared to the original oversized system. In a second house, located on the east central Florida coast in Merritt Island (Figure E-1), the properly-sized AC system provided similar indoor RH levels as the oversized system but increased energy use. In a third house, located in Lakeland, the properly-sized system modestly lowered energy use but also increased indoor RH levels compared with the original system. In a fourth project house, located near the southwest Florida coast in North Port, the very limited data available suggests higher RH levels with the properly-sized system with inconclusive energy use results. Project results suggest potentially significant utility coincident peak demand savings from the properly-sized systems.

Figure E-1. Outdoor unit of new, properly-sized AC system at Merritt Island house showing footprint of original oversized system on the concrete pad

While a full investigation of the reasons for the mixed results is beyond the scope of this project, there are several identified factors. The longer run times of the smaller air-conditioning systems compared to the oversized systems mean increased duct air leakage penalties and since cold air is flowing through the ducts for longer periods, heat conduction through the ductwork is also increased. Also, since the ductwork size was not reduced when the properly-sized systems were installed, the same duct surface area that was present for the oversized system now has cold air flowing through it for longer periods. The relatively large duct work may also explain why the properly-sized AC systems all had higher airflow rates per ton of cooling than the original systems, which may in turn partially explain higher post-change out RH levels. Variable speed air handlers can provide better control of evaporator coil airflow and may produce better results, but were not included in this study.

Measured Impacts of Proper Air Conditioning Sizing in Four Florida Case Study Homes
STAC Solicitation #03-STAC-1, Task 3.2 Benefits of Proper Sizing

Final Report

Jeff Sonne, Danny Parker and Don Shirey
Florida Solar Energy Center

INTRODUCTION

Previous studies have shown that residential air-conditioning system oversizing is a common practice that has both energy and comfort penalties. A Florida Power and Light / Florida Solar Energy Center (FSEC) study involving over 350 homes found that 50% of the study's AC systems were oversized by 120% or more (James et al. 1997). The same study found oversizing AC systems by 1.0 to 1.2 times Manual J resulted in 3.7% higher cooling energy use and oversizing systems 1.2 to 1.5 times Manual J resulted in 9.3% higher cooling energy use.

The goal of Task 3.2, Benefits of Proper Sizing, was to demonstrate the benefits of proper air conditioner sizing to contractors, customers and utilities. Field tests were conducted in case study homes; four homes were tested in Florida by FSEC and additional homes were tested in Wisconsin by the Energy Center of Wisconsin (ECW). The Wisconsin tests will be reported separately.

Project Homes

A search for Florida sizing study project homes began in April 2004 via an internal Florida Solar Energy Center email and by contacting personnel at several Florida electric utility companies, asking them to forward information on the study to other employees. The criteria specified homes:

- between 1,600 and 2,400 square feet with typical Florida construction
- built between 1999 and 2003 (occupied at least 1 year by present owners)
- owned by the current occupants (not rented)
- having one AC system for entire house (heat pump, electric resistance or gas heat)
- having "typical" occupancy (e.g. 2-6 occupants) and use.

To pre-qualify homes for the study, after getting an indication of interest in the study from the homeowners, energy audits were conducted at a total of five homes. The audits included measurements/verifications of building components (wall and floor construction types, window orientations, types, areas and overhangs, ceiling insulation levels, etc.) along with building and duct airtightness measurements. A detailed ACCA Manual J 8th edition load calculation was then completed for each home (using a "strict" interpretation of Manual J) to determine if/how much the present AC system was oversized. Summaries of these sizing calculations are provided in Appendix A. Four of the five homes that were audited had sufficiently oversized AC systems (> 25%) to qualify for the study, so no additional pre-qualifications were necessary.

All four project homes were identified by early-July 2004. The homes are located in Jacksonville, Merritt Island, Lakeland and North Port, providing locations throughout peninsular Florida from the northeast to the southwest. The four project home locations are shown in Figure 1.

Figure 1. Project home locations

Air conditioner capacity reductions for the four project homes ranged from 13,200 to 17,000 Btu/hr or an average of 31%. In each case, the replacement AC system selected was from the same manufacturer as the original system, and matched as closely as possible to the original system (e.g., model line and SEER). Table 1 provides a summary of the original and properly-sized (downsized) air conditioner capacity and efficiency characteristics for each house. ARI performance information for the original and properly-sized systems at each project house is provided in Appendix B.

Table 1. Original and New (Properly Sized) AC System Characteristics

Site	House Conditioned Floor Area (sq. ft.)	Manual J Load (Btu/hr)	Original AC System		New AC System	
			Capacity (Btu/hr)	SEER/HSPF	Capacity (Btu/hr)	SEER/HSPF
Jacksonville	2,255	28,418	47,000	12.05 / 7.3	33,000	12.05 / 7.5
Merritt Isl.	2,250	30,206	44,500	12.75 / NA	29,400	12.50 / NA
Lakeland	2,518	38,607	58,500	11.60 / NA	42,000	12.00 / NA
Northport	2,012	23,147	41,000	13.15 / 8.65	27,800	13.75 / 8.25

MONITORING

To compare performance of the original and downsized AC systems, air conditions and power use at each home were monitored on a 15-minute basis (2-minute basis for a subset of the measurements). Monitoring included:

- Air conditioner power use (total, condenser and air handler as shown in Figure 2),
- Air temperature and relative humidity at the thermostat (see Figure 3),
- Supply register air temperature (2-minute data; see Figure 4),
- Air temperatures entering and leaving coil (2-minute data), and
- Outdoor air temperature, relative humidity and horizontal surface solar radiation (see Figure 5).

Figure 2. Power measurements

In addition to the above, partway through the study period, monitoring of condensate removal and AC compressor on/off time was also added.

Monitoring was designed to include three phases: pre-tune up, post-tune up and finally post AC change out. Pre-tune up monitoring provided data on the existing AC systems as they were operating before the project technician inspected the systems and tuned them up. These were all relatively new systems, so the only maintenance required was adding refrigerant in three cases. Pre-tune up monitoring began in June 2004 at one house and in July 2004 at the other three houses.

Figure 3. Recorded thermostat temperature and humidity

Post-tune up monitoring began in July 2004 for three houses and in early-August 2004 for the fourth house. Both the post-tune up and post-change out monitoring phases were significantly affected by hurricanes Charley, Frances and Jeanne that summer. Data collection time was lost due to the hurricanes for a number of reasons including having to remove the weather station at each house at least once, boarding up windows at three of the four houses, inclement weather and electric power outages.[1]

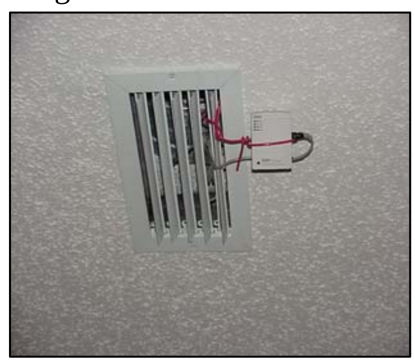

Figure 4. Recorded supply air temperature

AC changeouts were also delayed due to the hurricanes because of lost work days and significantly increased AC contractor work loads repairing storm-damaged systems. Three change outs were performed in September 2004 (16^{th}, 17^{th} and 24^{th}), with the final change out completed in early-October 2004. In three cases a performance check was completed at the time of the AC change out while in the fourth home the performance check was performed five days later.

Figure 5. Project weather station at the Merritt Island site

Due to the hurricane postponed and shortened post-monitoring period, a letter was sent to all four participants requesting that they allow approximately one month of additional monitoring during the summer of 2005. Due to budget constraints, summer 2005 monitoring was limited to the Merritt Island and Lakeland houses. The decision to use these two houses was made because of the proximity of the houses to our office location, cooperation of the homeowners, and the sale of the Jacksonville house in March 2005.

[1] For such future projects, because of the adverse impact of the late season installs, we would target all the AC change outs for the first two weeks in July with pre-change out data collection beginning in May.

DATA ANALYSIS

Lakeland

For the Lakeland project house (Figure 6), the original air conditioner was changed out on September 16, 2004. The original unit (Figure 7) had a nominal ARI 95°F outdoor/80°F indoor/ 67°F wet bulb condition cooling capacity of 58,500 Btu/hr. The *Manual J 8th Edition* estimated size for this 2,518 square foot house was 38,600 Btu/hr and thus a system with a nominal capacity of 42,000 Btu/hr was installed. Both the air handler and outdoor unit were changed out. The original system had a nameplate SEER of 11.6 Btu/W-hr; the new downsized system had a similar value of 12.0 Btu/W-hr. Tested total duct leakage (at 25 pascals, pre-retrofit) at this house was 273 cubic feet per minute (Qn_{tot} = 0.11) and leakage to outside was 86 cubic feet per minute (Qn_{out} = 0.03).

Figure 6. Project house in Lakeland,

Figure 7. Checking performance of existing AC system in Lakeland, FL

The original unit was oversized by approximately 52% – a typical condition based on previous survey data (Vieira et al. 1996). One key factor in the system change out, however, was the fact that single speed air handlers were used. These air handlers had multiple speed taps for the permanent split capacitor (PSC) motors. In each case, however, we found it impossible to match the nominal CFM/ton of cooling capacity in the pre-retrofit system to that in the post system. An *Energy Conservatory* flow plate was used to precisely measure pre- and post- air flow rates. At this project home, the oversized original system had an evaporator air flow of 1,660 cfm or 341 cfm/ton. With the post-retrofit downsized system, even choosing the lowest speed tap, the flow was 1,490 cfm or 426 cfm/ton. The difference in air flow per ton of cooling capacity was largest at this site compared to the other three test sites. As shown in other evaluations, such a disparity in evaporator coil flow rate can be expected to significantly affect coil temperatures and humidity removal – particularly at the higher flow rate.[2]

Detailed data were taken on the systems pre and post as documented in this report. Critical to the system evaluation, this included outdoor conditions (temperature, relative humidity and solar radiation) and air conditioner electric power. Comfort conditions included indoor temperatures and relative humidity. Cooling system supply air temperatures and condensate removal measurements were also made. In general, data were taken every fifteen minutes although some data were collected every two minutes.

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[2] Future projects of a like nature would be well advised to use variable speed air handlers in such a sizing study so that this important variable could be controlled to provide similar conditions pre and post AC change out.

Unfortunately for the project, the hurricanes of 2004 played havoc with the data analysis. This included Hurricanes Charley and Frances which affected the data prior to change out on August 13-18th inclusive and September 1st-15th. Thus, these data were lost for the pre-change out period and were removed from the available data stream. Similarly, just days after the installation of the new AC system, Hurricane Jeanne struck leading to loss of data for the dates from September 24th through October 1st. These data were removed prior to the analysis, but the piecemeal nature of the resulting "cleaned" data stream made it necessary to carefully match up weather data in the pre and post periods so that reasonable conclusions could be reached relative to performance. Fortunately, measured data were obtained in the post period for the new air-conditioning system in August 2005 which substantially improved the available data set.

To estimate the impact of the new AC systems, three previously-utilized techniques were used:

1. Comparison of long term pre and post periods with similar weather match.
2. Comparison of selected pre and post days with a close statistical match of weather conditions.
3. Linear regression of daily energy use against daily inside to outside temperature difference.

Lakeland Data Analysis

Figure 8 shows the average AC power and interior air relative humidity over the summer of 2004 when the AC was changed out. Note that maximum AC power drops after the retrofit, but interior relative humidity increases.

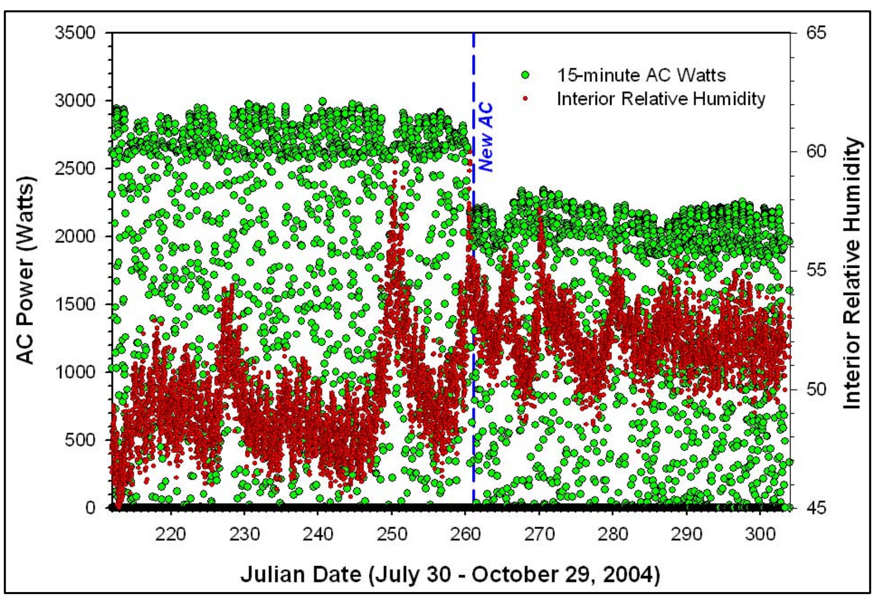

Figure 8. Time series data for AC power and interior relative humidity at the Lakeland site

Figure 9 shows a summary of the fundamental data from the project when evaluated over the longest periods of time with good weather match. The pre data includes the entire cleaned data set with all 15-minute data from July 29 - September 16, 2004 and comprises 2521 fifteen minute observations – 26 days of data. The post data consists of the cleaned data from September 22, 2004 through August 25th of 2005 with 3,980 observations – 41 days of data. The averages in the two periods are summarized below.

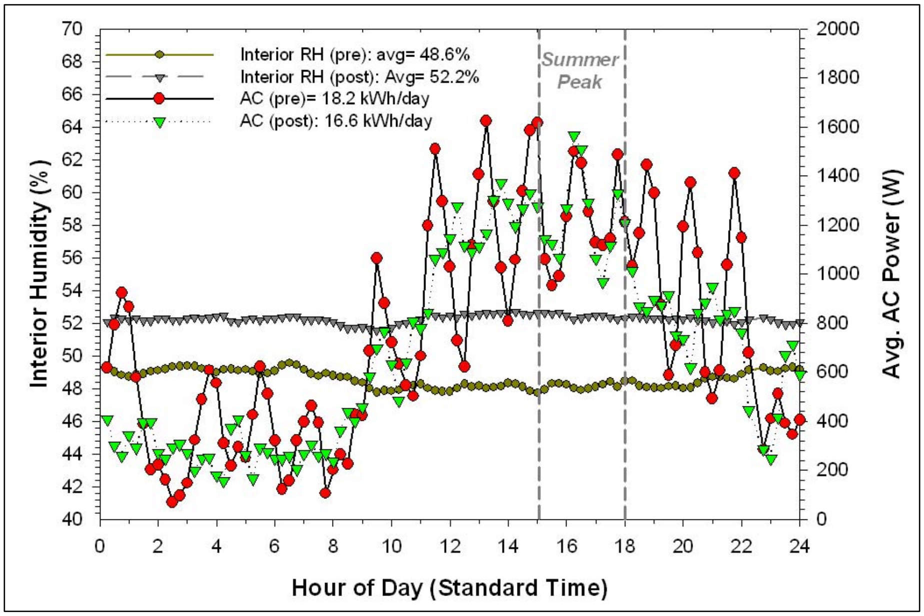

Figure 9. Lakeland site cooling load profile and interior humidity performance, pre and post AC retrofit matched weather. Ambient Air Temperature Pre 80.0 °F; Post 80.1 °F

Table 2. Summary of Lakeland Data Prior to Retrofit

Variable	Obs	Mean	Std. Dev.	Min.	Max.
AC (kWh/day)	2,521	18.1	26.5	0	72.0
Ambient Dry-bulb Temp. (F)	2,521	80.1	6.5	70.6	98.4
Ambient Dewpoint Temp. (F)	2,521	75.9	2.3	70.2	83.2
Insolation (W/m²)	2,395	190.9	281.8	0.6	1114.4
Interior Temp. (F)	2,521	77.1	0.9	73.1	78.6
Interior RH (%)	2,521	48.6	1.4	44.1	55.2
Condensate (oz)	2,521	5.49	6.8	0.0	41.9

Table 3. Summary of Lakeland Post-Retrofit Data

Variable	Obs	Mean	Std. Dev.	Min.	Max.
AC (kWh/day)	3,980	16.6	21.6	0	56.3
Ambient Dry-bulb Temp. (F)	3,980	80.0	7.9	60.5	102.3
Ambient Dewpoint Temp. (F)	3,188	72.3	3.6	60.2	80.3
Insolation (W/m²)	3,980	200.8	297.1	0.6	1079. 4
Interior Temp. (F)	3,978	76.9	0.9	73.5	85.6
Interior RH (%)	3,978	52.3	1.3	47.4	62.6
Condensate (oz)	3,980	7.3	9.3	0	54.2

The data show that the weather match in the two periods is quite good. The average outdoor air dry-bulb temperature is within 0.1°F for the two aggregate periods. Solar irradiance was similar with a variation of 10 W/m^2 on average (±5%). The ambient dewpoints were somewhat lower in the post period, indicating less outdoor moisture – not surprising given the hurricanes which saturated Central Florida in the summer of 2004.

Lakeland Energy Savings

The data summarized in the Tables 2 and 3 and Figure 9 reveal that the average air conditioning electric consumption (air handler, compressor and condenser fan) was 18.2 kWh/day in the pre period and 16.6 kWh/day in the post – a modest energy savings of about 9%, somewhat higher than the 3% that would be suggested by the difference in SEER. The aggregate plot (Figure 9) includes the full data set pre and post, less the removed data compromised by the hurricanes during summer. Note that the original AC system shows greater cycling than the new downsized system.

Figure 10 shows an analysis of all the days pre and post with the daily measured air conditioning electric consumption regressed against the recorded interior to exterior temperature difference. Although scatter is readily apparent, the slope term of the regression terms are identical, but with a change in the intercept term. When evaluated at a 3°F temperature difference (to approximate a typical summer day where the average outdoor temperature is 80^{o}F and the interior is maintained at an average of 77^{o}F), the regression estimates that space cooling electric consumption is lower with the new AC system by about 8%.

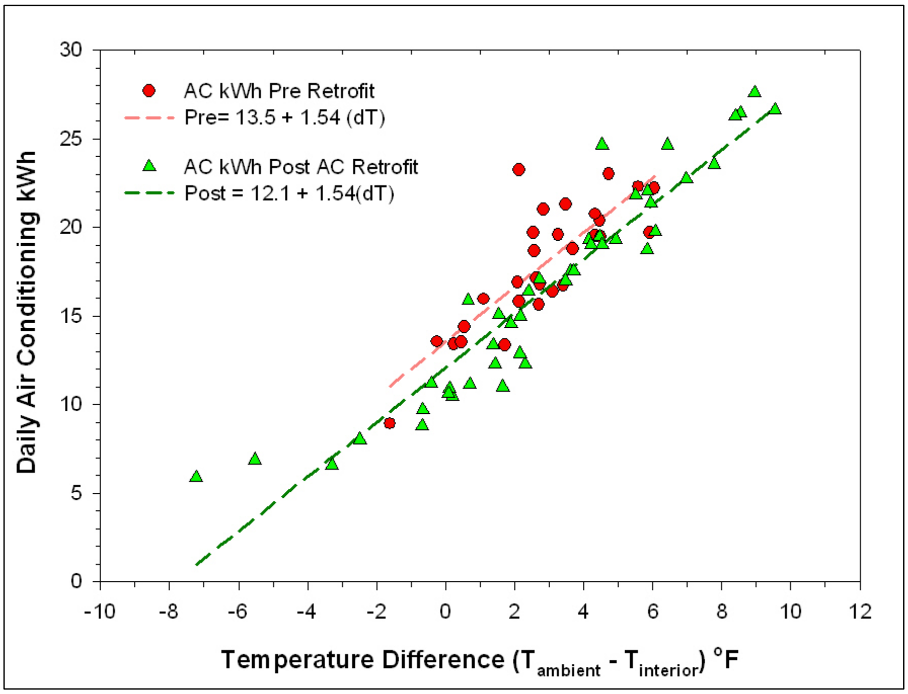

Figure 10. Regression of daily AC use pre and post against site temperature difference

Figure 11 shows an evaluation of two near-design days selected to yield very close weather in the pre and post periods. Here we selected data from August 22, 2004 – a very hot day, and compared that against data for October 3 of the same year. Note that the maximum temperatures for the two days are 96°F and 94°F, respectively as compared with the *Manual J 8th Edition* design day of 91°F for Lakeland. Interior temperatures on the two days were quite comparable. Thus, the selected day is hotter than the typical design day. The relative match of selected weather parameters is shown below in Table 4.

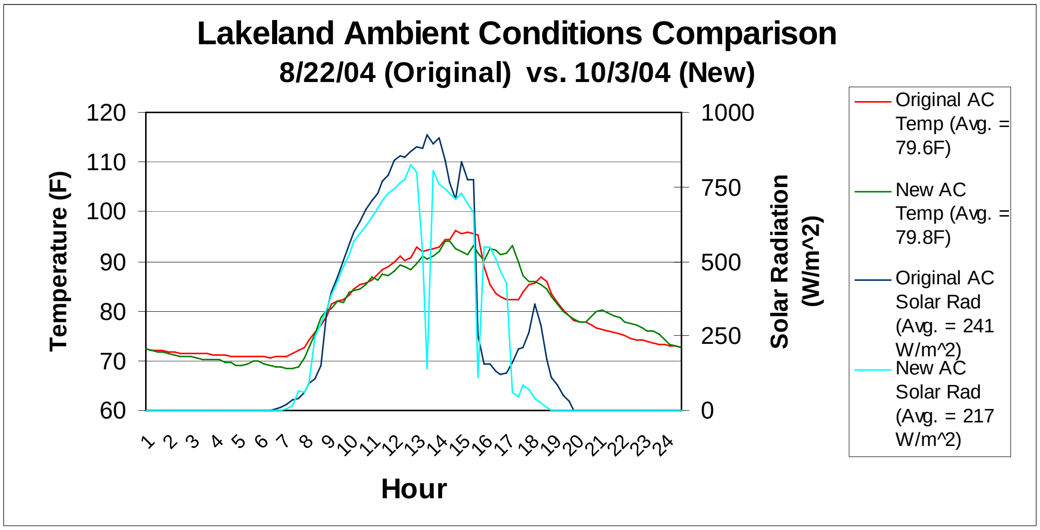

Figure 11. Weather on matched days pre and post for the Lakeland site

Table 4. Comparative Weather Conditions on Pre and Post Day for Matched Day Analysis

Pre, August 22, 2004

Variable	**Obs**	**Mean**	**Std. Dev.**	**Min.**	**Max.**
Ambient Dry-bulb Temp. (F)	96	79.6	8.2	70.6	96.1
Ambient Dewpoint Temp. (F)	96	74.1	2.3	70.5	79.6
Insolation (W/m²)	96	241.2	327.6	0.6	926.9
Interior Temp. (F)	96	77.1	0.9	74.6	78.5
Interior RH (%)	96	49.4	0.8	47.2	51.7
Condensate (oz)	96	39.1	48.1	0	215.0
AC Power (Watts)	96	820.6	1162.5	0	2956

Post, October 3, 2004

Variable	**Obs**	**Mean**	**Std. Dev.**	**Min.**	**Max.**
Ambient Dry-bulb Temp. (F)	96	79.8	8.4	68.4	94.1
Ambient Dewpoint Temp. (F)	96	71.5	1.7	68.5	76.6
Insolation (W/m²)	96	271.1	298.2	0.6	823.1
Interior Temp. (F)	96	77.1	1.0	74.8	78.4
Interior RH (%)	96	53.3	0.8	51.5	54.8
Condensate (oz)	96	55.2	70.4	1	339
AC Power (Watts)	96	711.5	940.4	0	2312

Three plots (Figures 12a, b, and c) show how the temperature, relative humidity, AC power and supply air temperature varied during the comparative days. Note that similar to the other analytical methods, the data shows a savings in AC power of about 13% (19.7 kWh/day vs. 17.1 kWh/day), albeit with worse relative humidity control. Note, however, that the supply air temperature for the new system with the higher coil air flow per unit capacity was greater by about 1°F. The runtime of the air-conditioning system was about 7.3 hours per day for the new system versus 6.7 hours per day for the original – an increase in runtime of 9% – less than expected given the ratio in the nominal capacity of the original and new equipment (58,500 Btu/hr vs. 42,000 Btu/hr). It should be noted, however, that with the greater coil air flow, the sensible capacity of the new equipment is almost likely greater than its nominal ARI rating.

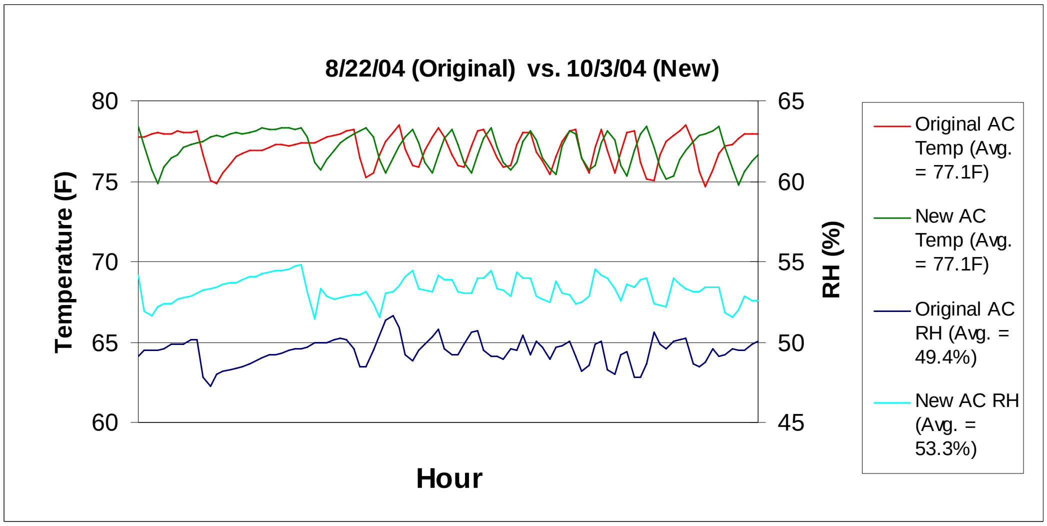

Figure 12a. Indoor air conditions comparison for the Lakeland site

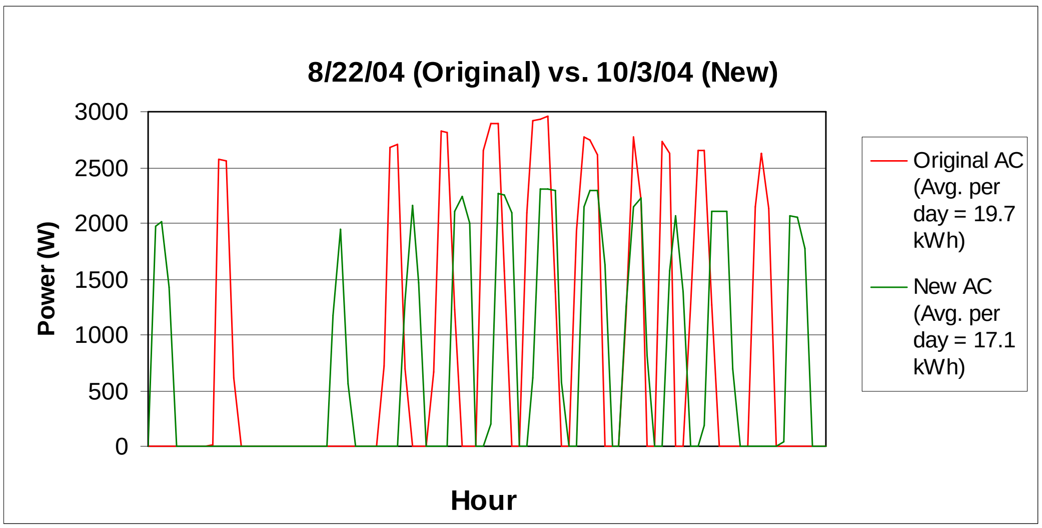

Figure 12b. AC power use comparison for the Lakeland site

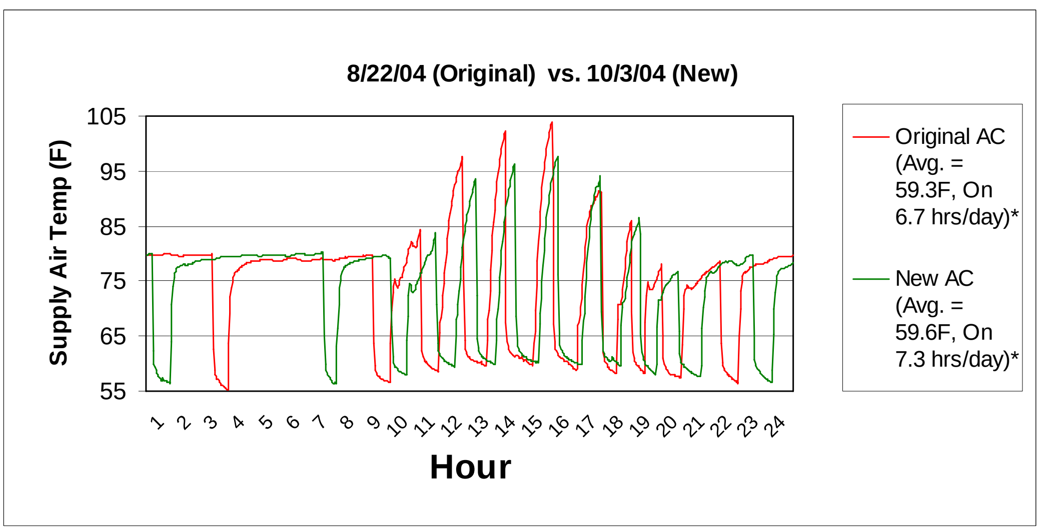

Figure 12c. Supply air temperature comparison for the Lakeland site

* All supply temperatures below 63^{o}F assumed as system on and used in averages and "on time" estimates.

We conclude from the three analysis methods that the new downsized system saved between 8 and 13% of daily space cooling energy use. This is higher than would be expected from the difference in SEER (3%) between the two systems.

Impacts on Relative Humidity and Moisture Removal

From a substantial body of previous work, we know that better equipment sizing should result in positive impacts to interior relative humidity control as short cycling reduces the latent capacity of cooling equipment (Shirey, Henderson and Raustad 2006).

Unfortunately, the data from the Lakeland site revealed that the new air conditioner, with its higher evaporator air flow per unit cooling capacity, did a worse job at controlling interior moisture levels. The measured interior relative humidity was 3.6% higher with the properly-sized air conditioner. While it would be convenient to attribute this slightly higher interior humidity level to the higher coil air flow, the higher measured condensate removal in the post period and pre and post dew points suggest that somehow the moisture load was much greater with the new machine. Note that dew points were lower in the post-monitoring period by over three degrees.

If a higher evaporator temperature and lower outdoor dewpoints were reducing the moisture being removed by the air conditioner, then we would expect to see lower moisture removal rates. In fact, we find just the opposite as shown in Figure 13. Here we see that the new air conditioner actually removed an average of 1.4 additional gallons of water each day after the unit was changed out. Given the lower outdoor dewpoint, this means that somehow an increased moisture load was being placed on the air conditioner. A likely explanation is that with the greater runtime of the new air-conditioning system, return side duct leakage and leakage from the air handler is placing additional load on the AC system. Indeed, a study done during the project to estimate the impact of reducing AC oversizing using the *EnergyGauge USA* simulation software suggested that most – if not all – of the benefit of right-sizing would be lost due to duct losses from conduction and induced air infiltration. This seems all the more likely given the fact that the air handler in the Lakeland house is in the attic space. Although the existing duct system would operate under lower pressure across the existing leakage, this influence was likely outmatched by increased fan runtime, particularly with return leaks in a hostile environment (attic air handler).

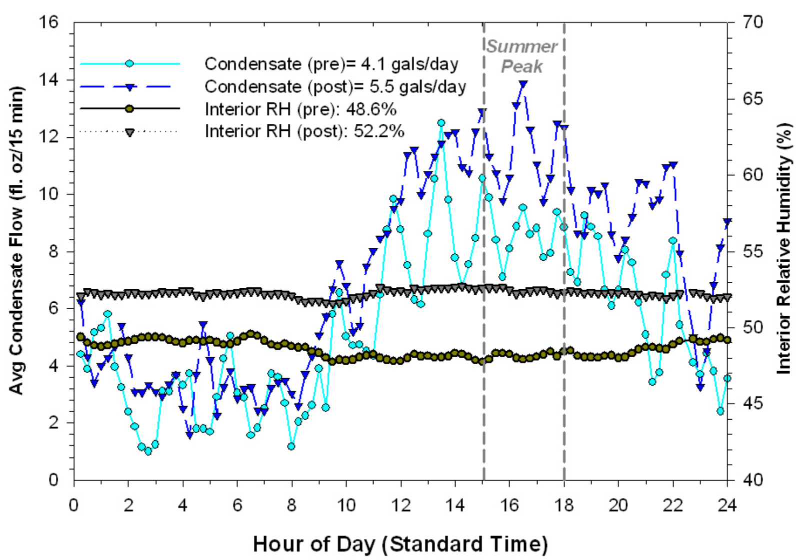

Figure 13. Lakeland site condensate profile and interior humidity performance pre and post AC retrofit matched weather.
Outdoor Dewpoint Temperature, Pre 75.9°F; Post 72.3°F

Since previous FSEC research shows that the average air handler in Florida homes leaks 70-80 cfm during operation (Cummings et al. 2003), the air handler is drawing in additional outdoor air during the extended runtimes with the right-sized system. This provides a satisfactory explanation for the fact that interior moisture removal rates were greater with the new system in spite of lower outdoor moisture conditions and higher evaporator coil air flow rates. With the properly-sized system, the longer runtimes relate to greater volumes of attic air being drawn into the air handler which results in the greater observed condensate removal. What is not as obvious is that the longer runtimes also are necessarily associated with greater heat gains from the duct system.

Merritt Island

For the Merritt Island house (Figure 14), the original air conditioner was changed out on September 24, 2004. The original unit had a nominal ARI 95°F outdoor/80°F indoor/67°F wet bulb condition cooling capacity of 44,500 Btu/hr. The *Manual J 8th Edition* estimated size for the cooling system for this 2,250 ft^2 home was 30,200 Btu/hr and thus a system with a nominal capacity of 29,400 Btu/hr was installed. Both the air handler and outdoor unit were changed out. The air handler was located in the garage (Figure 15). The original system had a nameplate SEER of 12.75Btu/W-hr; the new downsized system had a similar value of 12.5 Btu/W-hr. Tested total duct leakage (at 25 pascals, pre-retrofit) at this house was 178 cubic feet per minute (Qn_{tot}=0.08) and leakage to outside was 87 cubic feet per minute (Qn_{out}=0.04).

Figure 14. Project house in Merritt Island, FL

Figure 15. New air handler is installed at the Merritt Island site

According to *Manual J*, the original unit was oversized by approximately 47%. As in the other retrofits, a single speed air handlers was used. This air handler had multiple speed taps for the permanent split capacitor (PSC) motor. However, as with the Lakeland change out, we found it impossible to match the nominal CFM/ton of cooling capacity in the pre-retrofit system to that in the post system. The oversized system had an evaporator air flow of 1,330 cfm or 359 cfm/ton. With the post retrofit system, even choosing the lowest speed tap, the flow was 910 cfm or 372 cfm/ton. Thus, the relative coil air flow was about 4% higher post retrofit, but well within the typical recommendation for air flow for single speed systems (350 to 400 cfm/ton).

Detailed 15-minute data were taken on the systems pre and post. Unfortunately, hardware problems resulted in the post retrofit condensate data being lost. As at the other project sites, the hurricanes of 2004 substantially reduced the available data prior to the AC change outs. This included Hurricanes Charley and Frances which affected the data prior to change out on August 13-18th inclusive and September 1st-15th. Thus, these data were lost for the pre-change out period and were removed from the available data stream. Similarly, just days after the installation of the new system, Hurricane Jeanne struck leading to loss of data for the dates from 24th-28th September. These data were removed prior to the analysis, but the piecemeal nature of the data stream made it necessary to carefully match up weather data in the pre and post periods so that reasonable conclusions could be reached relative to performance.

Figure 16. Tune-up and commissioning of new AC system in Merritt Island

Fortunately, additional post-change out data was obtained for the new downsized air-conditioning system in August 2005 which substantially improved the available data set. Even here, we had to improvise to work past instrumentation hardware problems (Figure 16). In August 2005, the outdoor unit power measurement equipment failed, although we found that air handler fan power and outdoor air temperature could be used to quite accurately estimate the missing compressor power data by regressing these two parameters against measured compressor power from 2004 which was available for the new system. For this particular system we estimated the outdoor unit compressor power for the missing data as:

Compressor-Watts = 5.1313 (Fan Power) + 10.136 (Outdoor Temperature) -816.74
$R^2 = 0.9951$

To estimate the power use and conditions impacts of the new, properly sized AC system, we used the same three previously-utilized techniques:

1. Comparison of long term pre and post periods with similar weather match.
2. Comparison of selected pre and post days with closely matched weather conditions.
3. Linear regression of daily energy use against daily inside to outside temperature difference.

Merritt Island Data Analysis

Figure 17 shows the average AC power and interior relative humidity over the summer of 2004 when the AC was changed out. Note that maximum AC power drops after the retrofit, but interior relative humidity increases slightly.

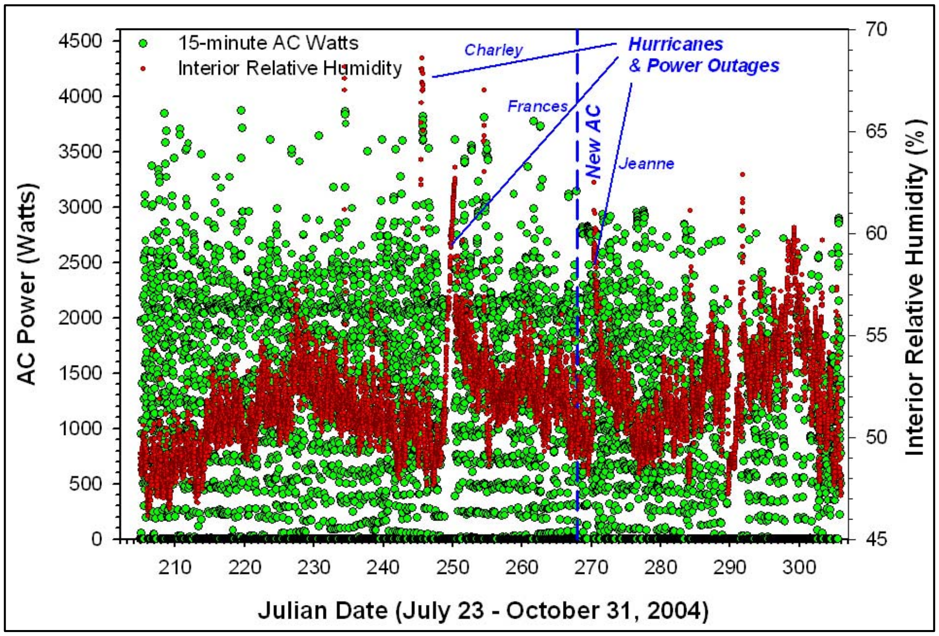

Figure 17. Time Series for AC power and interior relative humidity at the Merritt Island site

Figure 18 shows a summary of the fundamental data from the project when evaluated over the longest periods of time with good weather match. The pre data includes the entire cleaned data set with all 15-minute data from July 23 - August 31, 2004 and comprises 3456 fifteen-minute observations – 36 days of data. The post data consists of the cleaned data from September 28, 2004 through October 15, 2004 and July 16-31st of 2005 with 3,200 observations – 33 days of data. The averages from the two long-term periods are summarized below.

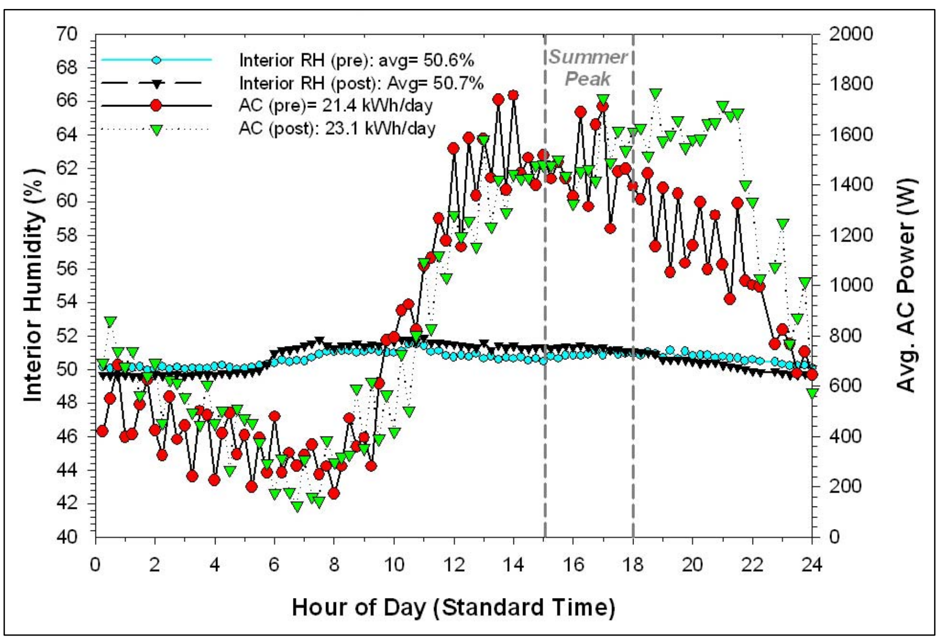

Figure 18. Merritt Island site cooling load profile and interior humidity performance pre and post AC retrofit matched weather.
Ambient Air Temp.; Pre 81.4°F; Post 81.1°F
Indoor Temp.; Pre 78.7°F, Post 78.9°F

Table 5. Summary of Merritt Island Data Prior to Retrofit

Variable	Obs	Mean	Std. Dev.	Min.	Max.
AC (kWh/day)	3455	21.4	23.1	0	92.8
Ambient Dry-bulb Temp. (F)	3456	81.4	5.0	72.8	94.5
Ambient Dewpoint Temp. (F)	3456	76.6	1.9	68.6	83.1
Insolation (W/m^2)	3456	200.5	289.4	0.6	1160.6
Interior Temp. (F)	3456	78.7	0.5	77.6	84.0
Interior RH (%)	3456	50.6	1.7	43.1	68.2

Table 6. Summary of Merritt Island Post-Retrofit Data

Variable	Obs	Mean	Std. Dev.	Min.	Max.
AC (kWh/day)	3200	23.1	23.3	0	71.1
Ambient Dry-bulb Temp. (F)	3200	81.1	5.8	66.9	97.1
Ambient Dewpoint Temp. (F)	3200	74.6	4.2	49.7	81.6
Insolation (W/m^2)	3200	210.6	299.0	0.6	1125.6
Interior Temp. (F)	3200	78.9	1.0	76.9	81.7
Interior RH (%)	3200	50.7	1.8	46.5	61.1

The data show that the weather match in the two periods is quite good. The average outdoor dry-bulb temperature is within 0.3°F for the two aggregate periods. Solar irradiance was similar with a variation of 10 W/m^2 on average (±5%). The ambient dewpoint temperatures were somewhat lower in the post periods, indicating less outdoor moisture – not surprising given the hurricanes which saturated Central Florida in the summer of 2004. Note that the post AC power peaked later in the day than the pre AC power. This difference is due to the homeowner using a daytime thermostat setup strategy during the 2005 portion of post-retrofit period, with the thermostat setpoint being increased in the morning and lowered in the evening (approximately 6 PM eastern standard time). However, also note that average pre/post interior temperatures for the comparison were still similar as shown in Tables 5 and 6.

Merritt Island Energy Savings

Energy savings were negative in this air conditioning change out. The data summarized in Tables 5 and 6 and Figure 18 reveal that the average air conditioning power (air handler, compressor and condenser fan) was 21.4 kWh/day in the pre period and 23.1 kWh/day in the post – a negative energy savings of about 8%, which is greater than the 2% decrease in performance that would be suggested by the difference in SEER. The aggregate plot (Figure 18) includes the full data set pre and post, less the removed data compromised by the hurricanes during summer. Note that the new downsized AC system shows increased energy use during the late afternoon and early evening hours, but very similar interior humidity levels pre and post. Energy use was generally the same or lower for the new system between midnight and 7 AM indicating that duct gains were likely responsible for the poorer afternoon performance.

Figure 19 shows an analysis of all days pre and post retrofit with the daily measured air conditioning electric consumption regressed against the recorded interior to exterior temperature difference. Although scatter is apparent -- and pre-retrofit data is limited -- both the slope and intercept term of the regression suggest worse performance for the new AC system. When evaluated at a 3°F temperature difference (to approximate a typical summer day where the average outdoor temperature is 80°F and the interior is maintained at an average of 77°F), the regression estimates that space cooling electric consumption is higher with the new AC system by about 15%.

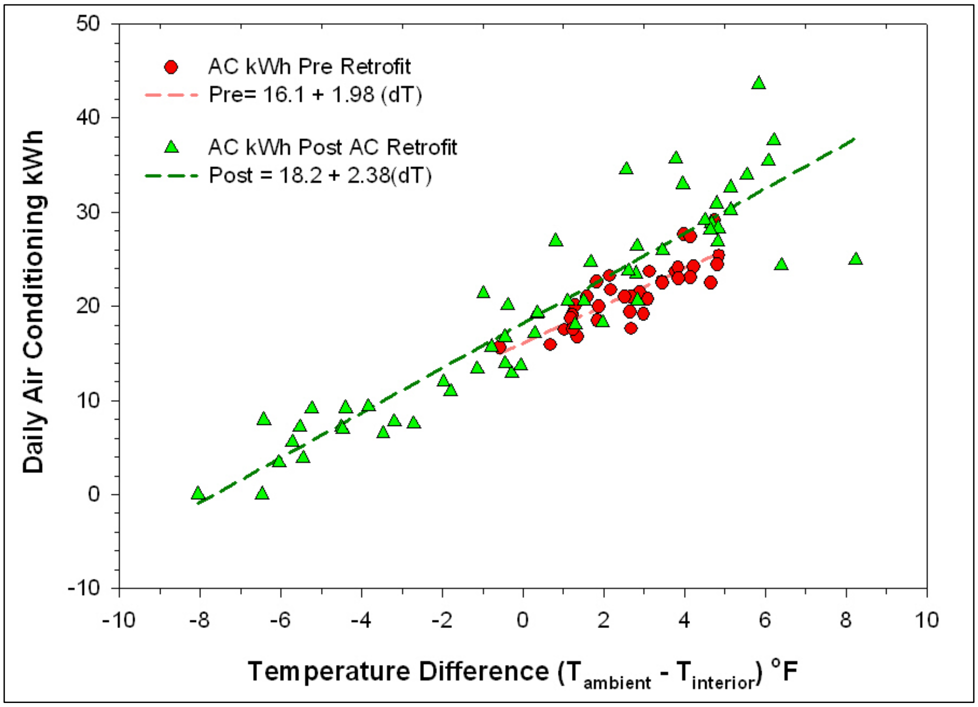

Figure 19. Impact of AC right-sizing on daily AC consumption.
Merritt Island, FL: 2004 & 2005

Figure 20 shows an evaluation of two days selected to yield close weather in the pre and post periods. Here we selected data from July 28, 2004 – a typical summer day, and compared that against data for October 3rd of the same year. Note that the maximum temperatures for the two days are 87.3°F and 89.4°F, respectively as compared with the *Manual J 8th Edition* design day of 90°F for Cape Kennedy. Interior temperatures on the two days were quite comparable. The relative match of selected weather parameters is shown below in Table 7.

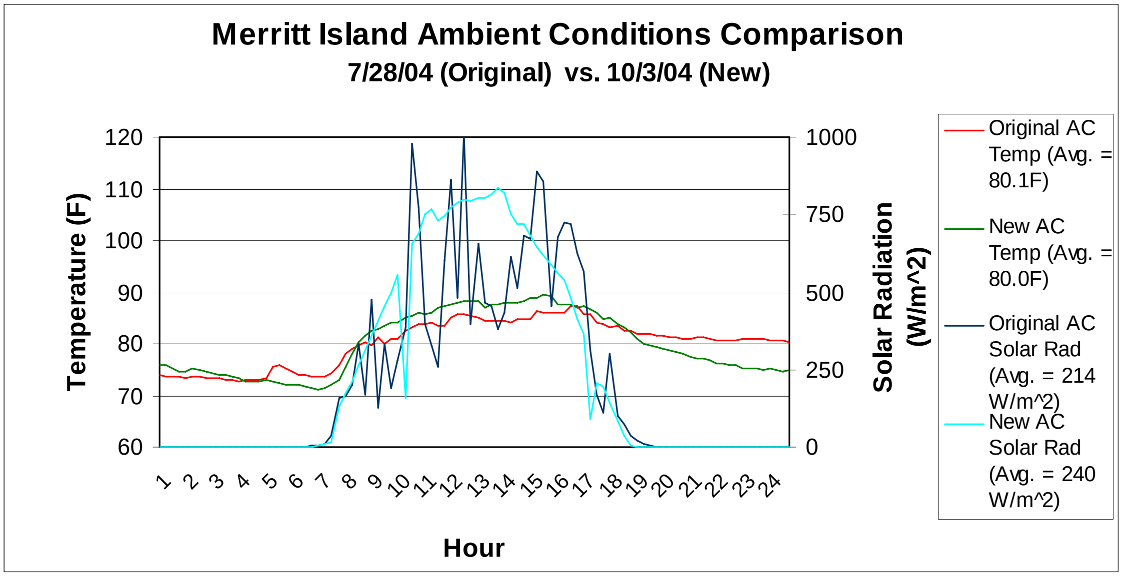

Figure 20. Weather on matched days pre and post for the Merritt Island site

Table 7. Comparative Weather Conditions on Pre and Post Day for Matched Day Analysis

Pre, July 28, 2004

Variable	Obs	Mean	Std. Dev.	Min.	Max.
Ambient Dry-bulb Temp. (F)	96	80.1	4.6	72.8	87.3
Ambient Dewpoint Temp. (F)	96	75.7	1.7	72.8	79.0
Insolation (W/m²)	96	213.7	286.7	0.6	101.9
Interior Temp. (F)	96	78.5	0.5	77.7	79.4
Interior RH (%)	96	49.1	0.7	47.8	50.9
AC Power (Watts)	96	875.5	1078.7	0	3700

Post, October 3, 2004

Variable	Obs	Mean	Std. Dev.	Min.	Max.
Ambient Dry-bulb Temp. (F)	96	80.00	6.0	71.2	89.1
Ambient Dewpoint Temp. (F)	96	73.64	1.3	69.0	76.1
Insolation (W/m²)	96	239.7	312.3	0.6	838.1
Interior Temp. (F)	96	78.3	0.2	77.8	78.8
Interior RH (%)	96	49.6	0.5	48.6	51.1
AC Power (Watts)	96	1027.5	942.5	0	2964

Three plots (Figure 21a, b, and c) show how the temperature, relative humidity, AC power and supply air temperature varied during the comparative days. Note that similar to the other analytical methods, the data shows an increase in AC electric consumption of about 17.6% (21.0 kWh/day vs. 24.7 kWh/day), and similar relative humidity control (49.1% vs. 49.6%). From a statistical standpoint, there was no difference in the interior humidity levels. Note, however, that the supply air temperatures for the new system with the higher coil air flow per unit capacity was greater by about 1.3°F -- indicative of somewhat warmer evaporator coil temperatures. The runtime of the AC system was about 6.8 hours per day for the new system versus 5.0 hour per day for the original – an increase in runtime of 36% – somewhat less than the ratio of the nominal capacity of the original and new equipment (44,500 Btu/hr vs. 29,400 Btu/hr or 51%).

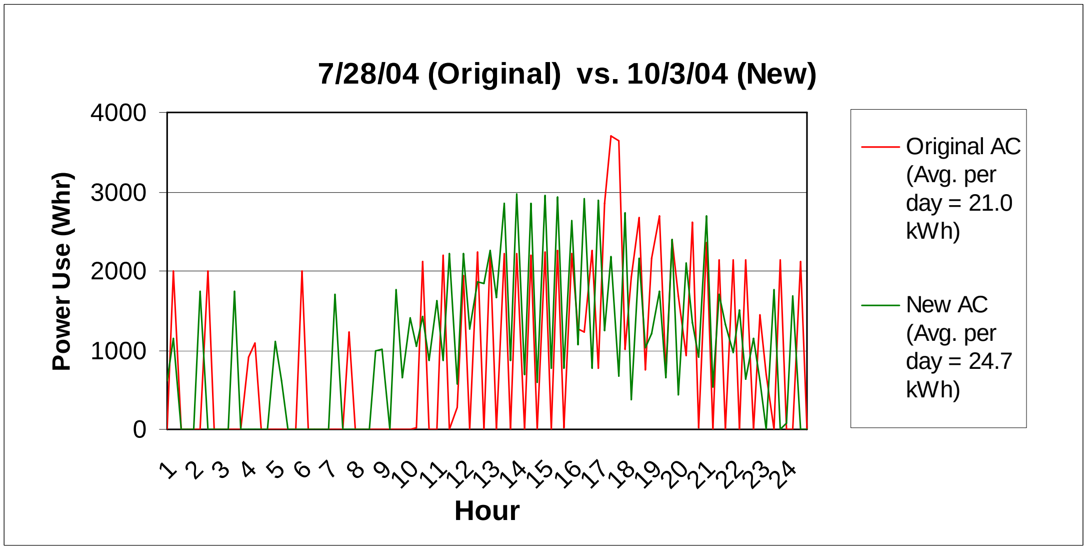

Figure 21a. Indoor air conditions comparison for the Merritt Island site

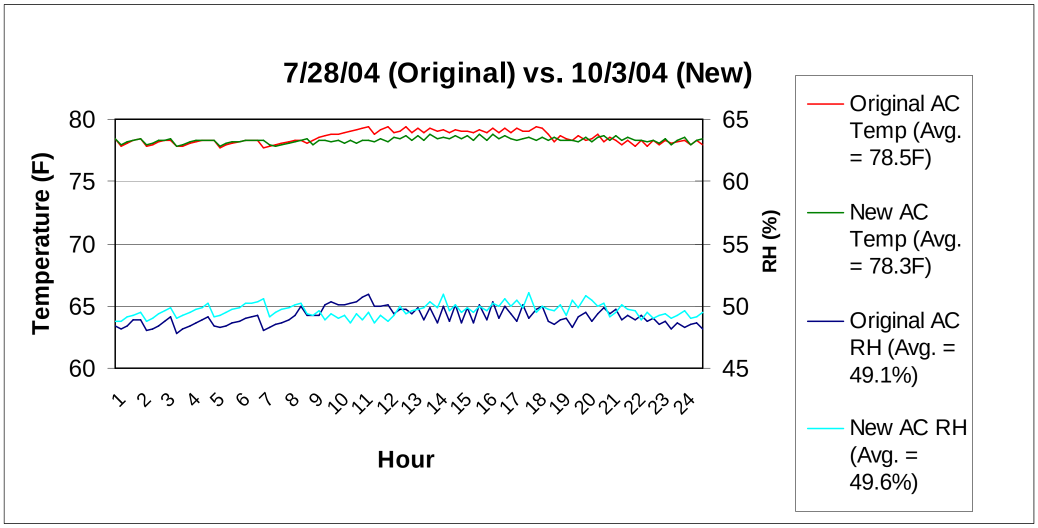

Figure 21b. AC power use comparison for the Merritt Island site

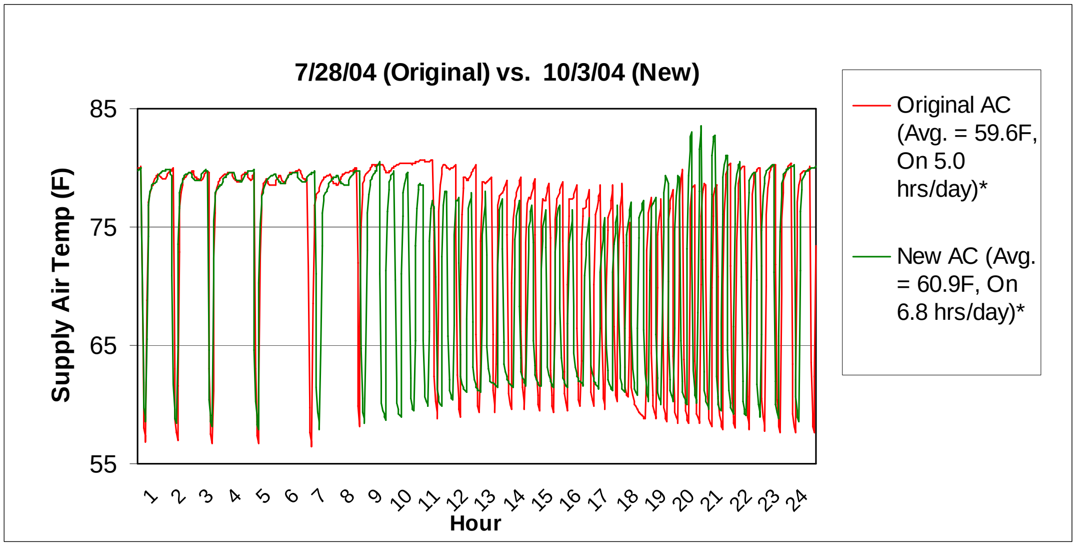

Figure 21c. AC supply air temperature comparison for the Merritt Island site

* All supply temperatures below 63°F assumed as system on and used in averages and "on time" estimates.

We conclude from the three analysis methods that the new downsized system increased daily space cooling energy use by 8 to 18%. The most likely explanation for the poorer performance of the new air conditioner system is that with the greater runtime of the new air-conditioning system, that duct leakage and conduction to the attic duct system is placing a considerable additional load on the AC system and, in the case of any supply leaks, loss of conditioned air. The home has a light gray shingle roof, which FSEC research shows can have mid attic air temperatures often reaching 120°F or more on hot summer days (Parker et al. 2000). It is perhaps significant that the period with the greatest increase in energy use with the new system is in the early evening hours when the attic space remains hot and increased runtime can be expected to increase duct conduction losses during this period.

Jacksonville

Figure 22. Project house in Jacksonville,

For the Jacksonville Florida house, the original air conditioner was changed out on September 17, 2004. The original unit was a heat pump with a nominal ARI 95°F outdoor/80°F indoor/ 67°F wet bulb condition cooling capacity of 47,000 Btu/hr. The *Manual J 8th Edition* estimated size for the system for this 2,255 square foot house was 28,420 Btu/hr and thus a system with a nominal capacity of 33,000 Btu/hr was installed. Both the air handler and outdoor unit were changed out. The original system had a nameplate SEER of 12.05 Btu/W; the new downsized system had an identical nameplate performance. Tested total duct leakage (at 25 pascals, pre-retrofit) at this house was 153 cubic feet per minute (Qn_{tot} = 0.07) and leakage to outside was 55 cubic feet per minute (Qn_{out} = 0.02).

Figure 23. New air handler at Jacksonville site

The original unit was oversized by approximately 65%. As in the other retrofits, a single speed air handler was used. This air handler had multiple speed taps for the permanent split capacitor (PSC) motor (Figure 23). As with the other project homes, we found it impossible to match the nominal CFM/ton of cooling capacity in the pre-retrofit system to that in the post system. The original oversized system had an evaporator air flow of 1,710 cfm or 436 cfm/ton. With the post retrofit system, even choosing the lowest speed tap, the flow was 1,273 cfm or 463 cfm/ton. Thus, the relative coil air flow was about 6% higher post retrofit, and somewhat higher in both cases than the typical recommendation for air flow for single speed systems (400 cfm/ton).

Detailed data were taken on the systems pre and post. Critical to the system evaluation, this included outdoor conditions (temperature, relative humidity and solar radiation) and air conditioner electric power. Comfort conditions included indoor temperatures and relative humidity. Cooling system supply temperatures were also taken, but hardware problems resulted in the post retrofit condensate data being lost (Figure 24). Data were taken every fifteen minutes, although some data were collected at 2-minute intervals.

Figure 24. New outdoor condenser at Jacksonville site

As at the other project sites, the hurricanes of 2004 reduced the available data before and after the AC change outs and made it difficult to obtain good exterior dew point matches as well as interior relative humidity conditions. These weather events included Hurricanes Charley and Frances which affected the data prior to change out on August 13th-18th inclusive and September 1st-15th. Thus, these data were lost for the pre-change out period and were removed from the available data stream. Similarly, just days after the installation of the new system, Hurricane Jeanne struck leading to loss of data for the dates from 24 - 28 September. These data were removed prior to the analysis, but the piecemeal nature of the data stream made it necessary to carefully match up weather data in the pre and post periods so that reasonable conclusions could be reached relative to performance.

To estimate the impact of the new AC system, we used the same three previously-utilized techniques:

1. Comparison of long term pre and post periods with similar weather match.
2. Comparison of selected pre and post days with a close statistical match of weather conditions.
3. Linear regression of daily energy use against daily inside to outside temperature difference.

Analysis of Matched Long Term Periods

Figure 25 shows the average AC power and interior relative humidity over the summer of 2004 when the AC was changed out. Note that maximum AC power drops after the retrofit, but interior relative humidity does not appear substantially changed.

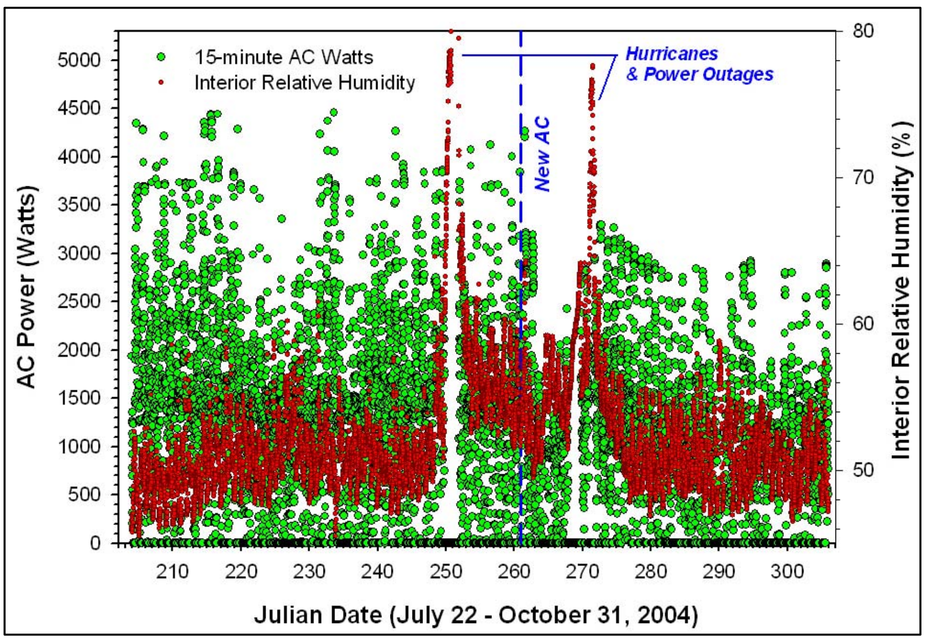

Figure 25. Time series data for AC power and interior relative humidity at the Jacksonville site

Figure 26 shows a summary of the fundamental data from the project when evaluated over the longest periods of time with good weather match. Unfortunately, due to the late date which the new system went in (September 17, 2004) the weather was much cooler post retrofit, requiring that the data be segmented in both the pre and post periods to obtain reasonable match to outdoor temperature conditions. Another problem was that the new air conditioner generally maintained a cooler indoor temperature in the post period – averaging about 1°F lower. A final problem was that Hurricane Jeanne struck the week after the system was changed out. This resulted in one day without power at the site and increases in interior moisture that were not removed for days after the hurricane. This problem is graphically illustrated in Figure 27 which shows air conditioner power and interior air conditions superimposed over the period. This week-long period after the storm with the new air conditioner had to be removed from the data set to prevent bias in the loads and interior relative humidity with the new machine.

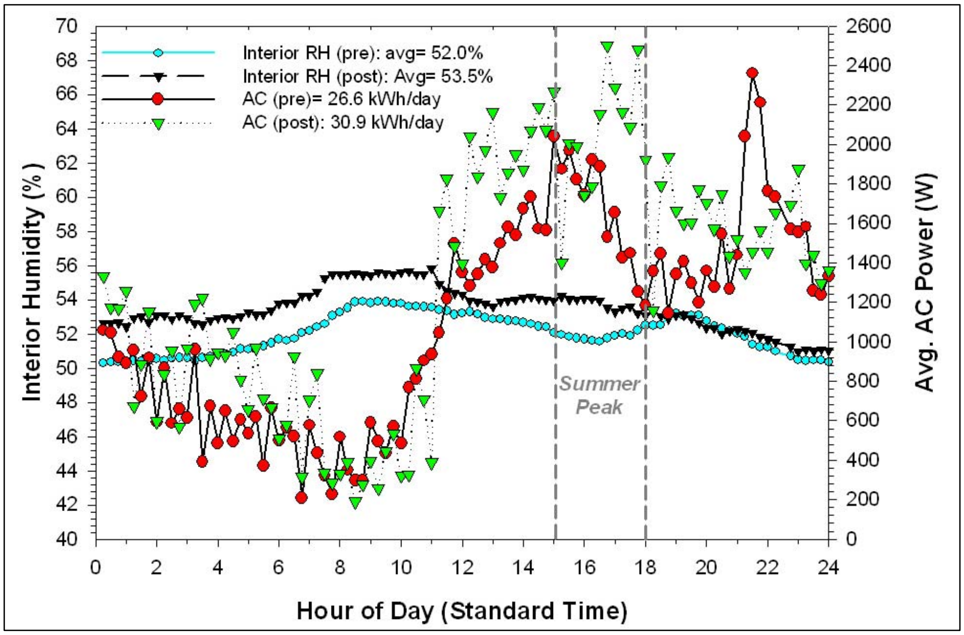

Figure 26. Jacksonville site cooling load provide and interior humidity performance pre and post AC retrofit matched weather.
Ambient Air Temp.; Pre 79.2°F, Post 79.1°F
Indoor Temp.; Pre 74.8°F, Post 74.0°F

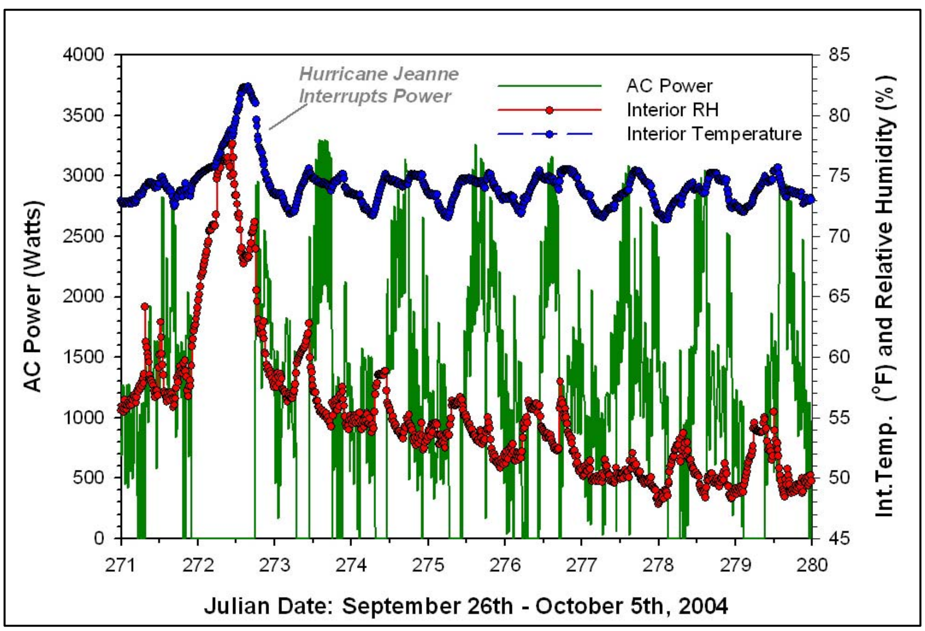

Figure 27. Impact of hurricane-related moisture on long-term interior humidity control

The pre data includes the entire data cleaned data set with all 15-minute data from August 4-September 16, 2004 and comprises 3,840 fifteen-minute observations – 40 days of data. The post data consists of only the cleaned data in the post period from September 18, 2004 through October 3, 2004 with 768 observations – 8 days of data. The averages in the two periods are summarized below in Tables 8 and 9.

Table 8. Summary of Jacksonville Data Prior to Retrofit

Variable	Obs	Mean	Std. Dev.	Min.	Max.
AC (kWh/day)	3833	26.6	22.4	0	107.0
Ambient Dry-bulb Temp. (F)	3840	79.2	6.5	67.0	99.4
Ambient Dewpoint Temp. (F)	3840	74.1	2.9	63.1	81.5
Insolation (W/m²)	3840	141.3	236.4	0.6	1079.4
Interior Temp. (F)	3840	74.8	1.3	69.9	78.6
Interior RH (%)	3840	52.0	2.8	45.5	64.6

Table 9. Summary of Jacksonville Post-Retrofit Data

Variable	Obs	Mean	Std. Dev.	Min.	Max.
AC (kWh/day)	768	30.9	22.11	0	102.3
Ambient Dry-bulb Temp. (F)	768	79.1	7.3	67.0	94.0
Ambient Dewpoint Temp. (F)	768	72.7	3.3	57.0	79.3
Insolation (W/m²)	768	188.3	270.1	0.6	909.4
Interior Temp. (F)	768	74.0	1.4	71.4	80.1
Interior RH (%)	768	53.5	3.0	47.8	64.3

The data show that the weather match in the two periods is fair. The average outdoor temperature was quite good – within 0.1°F of the two aggregate periods. However solar irradiance differed being within 40 W/m^2 on average (±33%). Unfortunately, this could not be remedied without poor matches on temperature which was considered the more important weather parameter. The dewpoints were somewhat lower in the post periods, indicating less outdoor moisture – not surprising given the hurricanes which saturated Central Florida in the earlier summer of 2004.

Jacksonville Energy Savings

Data analysis revealed no energy savings in this air conditioning change out. The data summarized in Tables 8 and 9 and Figure 26 reveal that the average air conditioning power (air handler, compressor and condenser fan) was 26.6 kWh/day in the pre period and 30.9 kWh/day in the post – a negative energy savings of about 16% greater energy use in the post period. However, note that the post data had a lower interior temperature of about 0.8°F which could not be adjusted. The aggregate plot (Figure 26) includes the full data set pre and post, less the removed data compromised by the hurricanes during summer. Note that the downsized system shows increased energy use during daytime hours between noon and 8 PM, but similar energy use in other hours. The observation fits the hypothesis that duct conductive heat gains and duct leakage from extended runtimes were impacting energy use during daytime hours when attic temperatures are high. Interior humidity levels were slightly higher in the post period. This fits the supply air temperature data which showed slightly higher coil air temperatures post retrofit.

Jacksonville Regression Analysis

Figure 28 shows an analysis of all the days pre and post retrofit with the daily measured air conditioning electric consumption regressed against the recorded interior to exterior temperature difference. Although scatter is apparent, both the slope and intercept term of the regression suggests similar performance for the new system. When evaluated at a 3°F temperature difference (to approximate a typical summer day where the average outdoor temperature is 80°F and the interior is maintained at an average of 77°F), the regression estimates that space cooling electric power is virtually identical for the system pre and post. Thus, this method alone corrects for the lower thermostat temperature with the new system and predicts similar performance between the original and new AC systems.

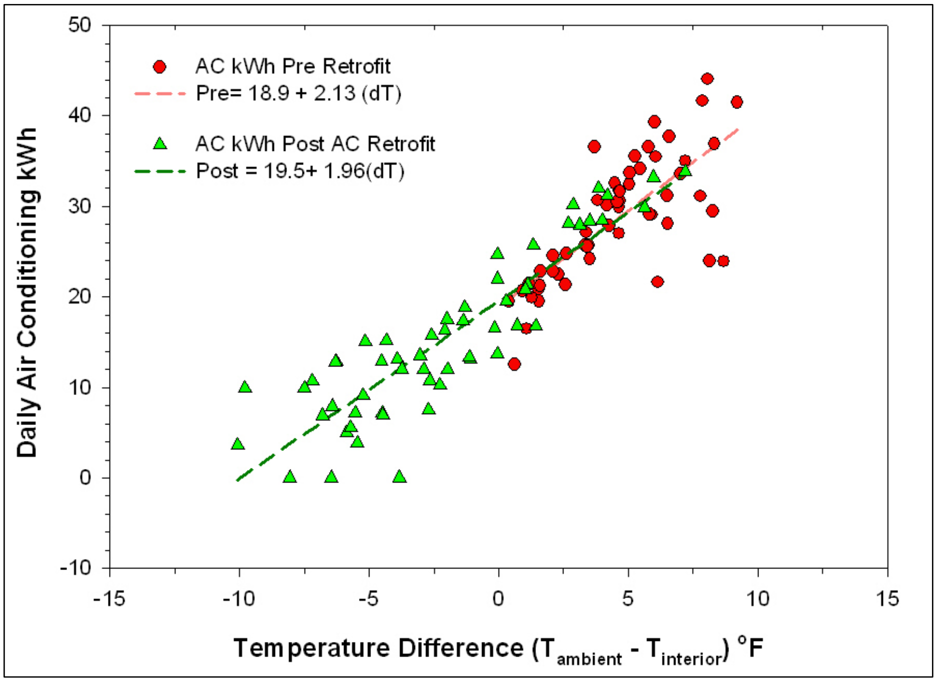

Figure 28. Impact of AC right-sizing on daily AC consumption. Jacksonville, FL: 2004

Figure 29 shows an evaluation of two days selected to yield close weather in the pre and post periods. Here we selected data from August 30, 2004 – a typical summer day, and compared that against data for October 2nd of the same year. Note that the maximum temperatures for the two days are 93°F and 94°F, respectively as compared with the *Manual J 8th Edition* design day of 93°F for Jacksonville. Interior temperatures on the two days were quite comparable. The relative match of selected weather parameters is shown below in Table 10.

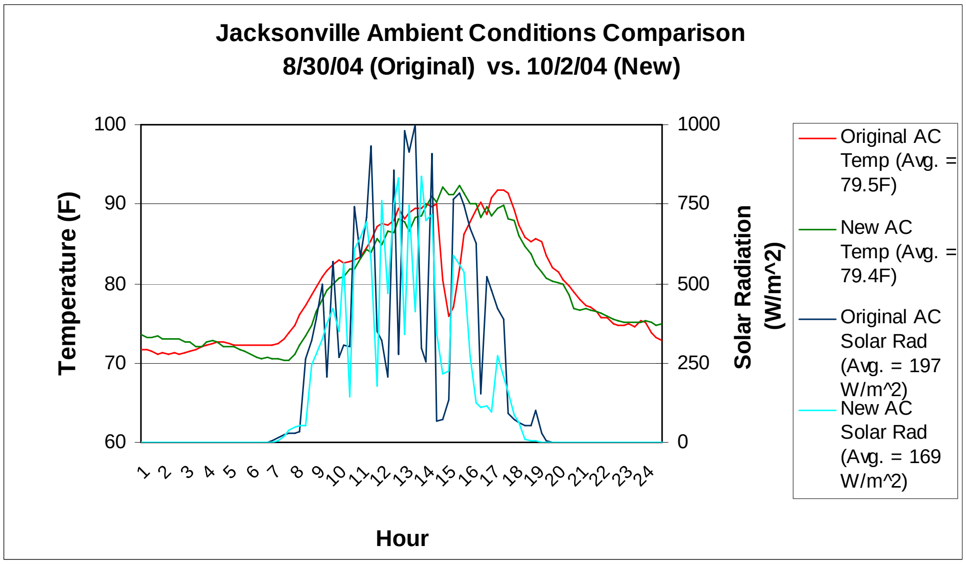

Figure 29. Weather on matched days pre and post for the Jacksonville site

Table 10. Comparative Weather Conditions on Pre and Post Day for Matched Day Analysis

Pre, August 30, 2004

Variable	Obs	Mean	Std. Dev.	Min.	Max.
Ambient Dry-bulb Temp. (F)	96	80.1	7.3	71.1	93.2
Ambient Dewpoint Temp. (F)	96	75.4	2.6	71.2	80.4
Insolation (W/m²)	96	196.9	292.3	0.6	1001.9
Interior Temp. (F)	96	74.8	1.6	73.0	77.4
Interior RH (%)	96	50.0	1.5	47.1	54.2
AC Power (Watts)	96	1320.1	1064.9	0	3924

Post, October 2, 2004

Variable	Obs	Mean	Std. Dev.	Min.	Max.
Ambient Dry-bulb Temp. (F)	96	79.9	7.4	70.4	94.0
Ambient Dewpoint Temp. (F)	96	73.4	1.1	71.5	75.4
Insolation (W/m²)	96	168.8	250.8	0.6	839.4
Interior Temp. (F)	96	74.6	1.7	70.8	77.3
Interior RH (%)	96	50.2	0.7	48.3	52.1
AC Power (Watts)	96	1383.4	674.5	28	3088

Three plots (Figure 30a, b, and c) show how the temperature, relative humidity, AC power and supply air temperature varied during the comparative days. Note that consistent with the other analytical methods, the data shows a slight increase in AC electric consumption of about 5% (31.7 kWh/day vs. 33.2 kWh/day), with very similar relative humidity control. Note, however, that the supply air temperature for the new system with the higher coil air flow per unit capacity was very similar. The runtime of the air-conditioning system was about 12.8 hours per day for the new system versus 8.5 hours per day for the original – an increase in runtime of 51% – longer than the ratio in the nominal capacity of the original and new equipment (47,000 Btu/hr vs. 33,000 Btu/hr = +42%). This may indicate that the runtime itself was adding load to the operation of the machine during daytime hours when the attic is hot.

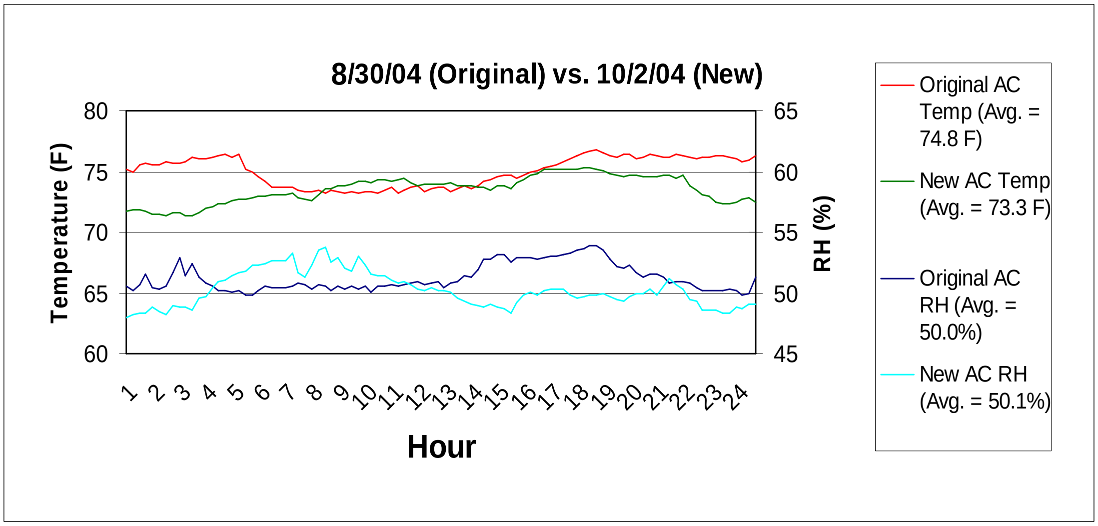

Figure 30a. Indoor air conditions comparison for the Jacksonville site

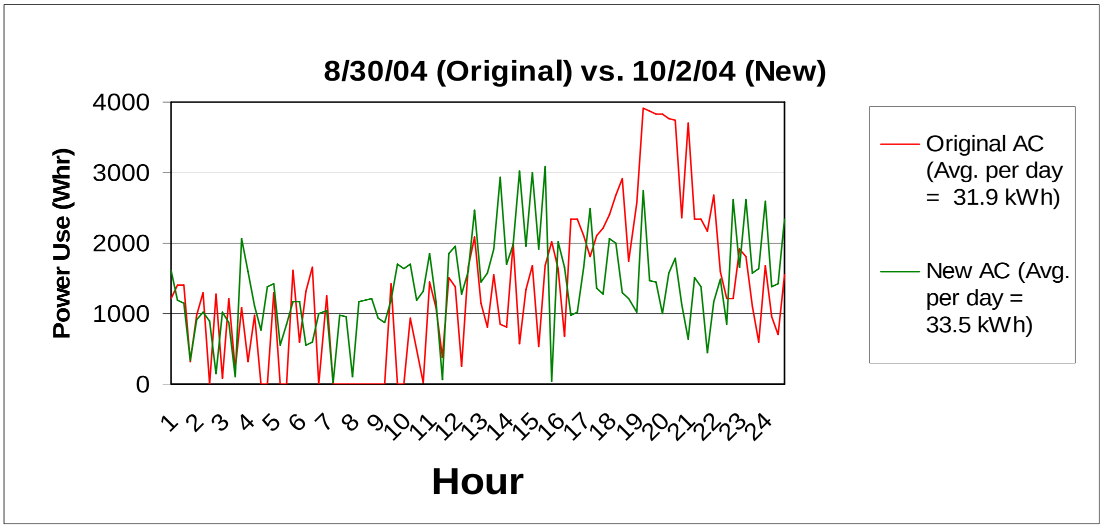

Figure 30b. AC power use comparison for the Jacksonville site

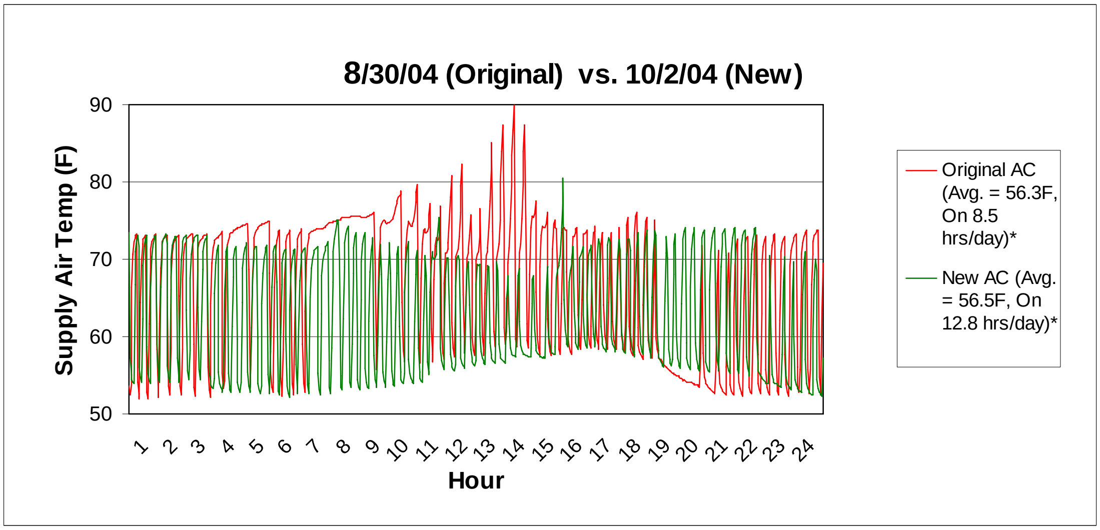

Figure 30c. AC supply air temperature comparison for the Jacksonville site

* All supply temperatures below 63^{o}F assumed as system on and used in averages and "on time" estimates.

Jacksonville Summary and Discussion

We conclude from the three analysis methods that the new system increased daily space cooling energy use by 0 to 16%. The most likely numbers are those emerging from the regression which controls for the lower indoor temperature post AC change out. Thus, from a statistical standpoint, energy use was unchanged with the new machine.

The most likely explanation for the lack of savings for the new air conditioner system is that with the greater runtime of the new air conditioning system, that duct leakage and conduction to the duct system is placing additional load on the AC system and, in the case of any supply leaks, losses of conditioned air. It is perhaps significant that the period with the greatest increase in energy use with the new system is during the daytime hours between noon and 8 PM when the attic space remains hot and increased runtime can be expected to increase duct conduction losses during this period. Indeed, a simulation study of the impact of reducing AC oversizing with the *EnergyGauge USA* simulation software had already suggested that most – if not all – of the benefit of right-sizing would be lost due to duct losses from conduction and induced air infiltration due to duct leakage.

North Port

For the North Port Florida house (Figure 31), mainly due to the 2004 hurricanes, the original air conditioner was changed out very late, on October 7, 2004. The original unit was a heat pump with a nominal ARI 95°F outdoor/80°F indoor/ 67°F wet bulb condition cooling capacity of 41,000 Btu/hr. The *Manual J 8th Edition* estimated size for the system for this 2,012 square foot house was 23,150 Btu/hr and thus a system with a nominal capacity of 27,800 Btu/hr was installed. Both the air handler and outdoor unit were changed out. The original system had a nameplate SEER of 13.15 Btu/W; the new downsized system had a nameplate SEER of 13.75. Tested total duct leakage (at 25 pascals, pre-retrofit) at this house was 178 cubic feet per minute (Qn_{tot} = 0.09) and leakage to outside was 34 cubic feet per minute (Qn_{out} = 0.02).

Figure 31. Project house in North Port, Florida

There is no 2004 post-change out North Port monitored data that can be directly compared with the pre-change out data (the two highest maximum daily temperatures during the post-change out period were only 86°F and 88°F). However, a comparison using just this data from two post period days indicates that both power use and RH levels are higher with the smaller post-change out AC. Indoor conditions data from the North Port home over the summer of 2005 indicate that relative humidities for the new smaller system averaged around 3% higher than those seen with the original larger system (Figure 32).

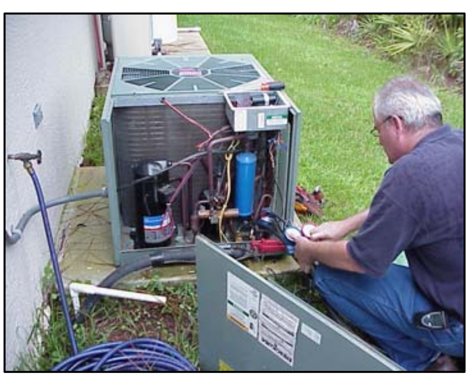

Figure 32. Technician verifies performance of new AC system at North Port site

All Sites

Homeowner Satisfaction

Initial feedback from two of the homeowners indicated overall satisfaction with the new, smaller AC systems. One homeowner noted that the relative humidity seemed to be higher in his house after the change out. Another homeowner indicated that they preferred the higher airflow rates of the original system although they were otherwise satisfied with the new unit (note that the supply air diffusers were sized for the original system so reduced flow with the new, downsized system reduced air velocities at the supply diffusers which impacted air circulation in the conditioned spaces). The Jacksonville homeowner was satisfied with the new system and the house was sold with this system. The other three homeowners were asked to make a final decision on if they would be keeping the new systems after the 2005 summer season. The Merritt Island and North Port homeowners decided to keep their new properly-sized AC systems. Mainly due to the higher RH levels experienced with the smaller system, the Lakeland homeowner had the original AC system re-installed.

System Airflows

Table 11 shows the measured air flows for each of the original and new AC systems in each project house. In each house, the original AC system's airflow per ton was lower than the new system's airflow per ton, with the properly-sized systems running from 13 to 85 cfm/ton higher than original systems' airflows. This difference is due to the fact that the duct work in these homes were sized for the larger systems and now the smaller systems are operating at lower pressures, allowing more airflow.

Table 11. Summary of Original and New (Downsized) AC System Capacities and Air Flow Rates

Site	Original AC System		New AC System	
	Size (Btu/hr)	**Air Flow (cfm/ton)**	**Size (Btu/hr)**	**Air Flow (cfm/ton)**
Jacksonville	47,000	436	33,000	463
Merritt Isl.	44,500	359	29,400	372
Lakeland	58,500	341	42,000	426
Northport	41,000	372	27,800	424

Figure 33 shows the relationship between the differences in airflow per ton rates and differences in RH levels between the original and properly-sized systems. It shows that all new system average RH levels are higher than those of the original systems, and that as the airflow per ton differences increased, the RH levels in the houses also increased.

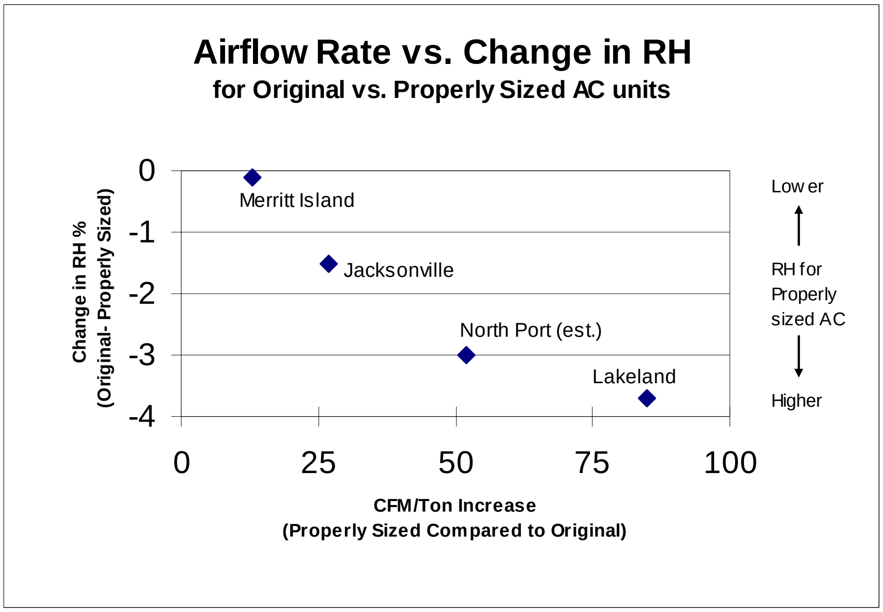

Figure 33. Air handler air flow rate and RH differences pre and post-change out in the four project houses (North Port is an estimate)

Impacts of Machine Sizing on Utility Coincident Peak Demand

While utility coincident summer peak demand (kW) savings due to machine downsizing was found to be an important impact, reporting these values is not uniformly possible. The reason is that the "matched days" chosen for the matched-days analysis were not necessarily peak weather days, but rather the warmest days for which we could find a good weather match (without changes to the interior thermostat). One could argue that the Lakeland matched days can be characterized as “peak days” as they were both quite hot (94°F and 96°F, respectively) and the same is true for Jacksonville where the peak outdoor temperatures were fairly similar (93.2°F and 94°F). This was not true, however, for the Merritt Island matched days where the maximum outdoor temperatures were 87-89°F. There was no analysis possible with Northport; the data were taken at the end of the season and cannot be considered to characterize peak under any circumstances.

Also, one must be very careful with the definition of the peak kW savings. The relevant peak kW savings are those that occur during the utility coincident system peak. Nor is this the instantaneous peak. Certainly, one cannot just examine the peak demand of the relative AC systems and conclude these differences are the peak savings-- unless comparing two systems which are activated right at the beginning of the utility peak window and spend the entire time "pulling down" the interior conditions to the set point.

In Florida, the electric utility summer peak period extends from 4 - 7 PM EDT (our data is reported in standard time in the graphs). Over this long of a period, diversity of AC operation in a large group of air conditioners becomes important -- with longer runtimes of smaller machines, some amount of what is gained from a lower short-term kW demand will be lost with a longer period of AC demand within the cycle. Thus, smaller sizes will increase the number of AC systems in a large statistical group which are operating at any given time.

Since the monitoring was compromised by the weather in the summer of 2004 and the matched days coming from different times of year with differing solar conditions, any analysis of peak reductions must necessarily be crude. With those cautions, estimates of peak electric demand reduction for the Lakeland and Jacksonville homes are given below:

Table 12. Estimated Coincident Peak Demand Reduction from Downsized AC Systems

Site	Pre-Retrofit	Post Retrofit	Savings
Lakeland	1288 W (Aug. 22, 2004)	911 W (Oct. 3, 2004)	377 W (29%)
Jacksonville	3124 W (Aug. 30, 2004)	1574 W (Oct. 2, 2004)	1550 W (50%)

Note: Average peak demand from 4-7 PM (EDT)

These impacts are potentially important. The ratios of the changed nominal system size (Btu/hr) were roughly 30% for either of the above cases (see Table 11). In Table 12, we see that the change in peak demand were fully as large (or larger) than the change to nominal capacity. While the AC sizing issue does not look to have large ramifications on energy consumption (kWh) for consumers with ducts in attics, it does appear to have potentially larger impacts for utilities during their peak generation periods.

In another study done for *Progress Energy Florida* with a very large statistical sample of 171 sub-metered homes with air conditioners, the AC size had a statistically-significant impact on peak electric demand (Parker 2002). The present study tends to reinforce that previous finding.

DISCUSSION

Cycling Losses in Modern Air Conditioners under Part-Load Conditions

Several factors conspire to make modern vintage air-conditioning systems less susceptible to impacts by cycling losses. Manufacturers make significant efforts to reduce the cooling coefficient of degradation (C_D) which enters into the calculation of seasonal energy efficiency ratio (SEER). Given the mathematical formulation of C_D, the energy losses associated with a C_D value are approximately one half the fractional value. This is possibly a major factor in why this study's results differ from earlier studies that indicate energy savings from smaller systems (e.g. James et al. 1997).

Larger unitary air conditioners (>65,000Btu/hr) are rated using EER, a rating standardized by ARI, which reports steady-state efficiency at 95°F outdoor and 80°F indoor temperature. However, smaller air conditioners (<65,000 Btu/hr) are rated using SEER, a rating developed by the U.S. DOE and based on EER measurements, intended to better indicate average seasonal performance, i.e., a season average EER. Government programs, many utility programs and consumers have relied on seasonal energy efficiency ratio (SEER) as the indicator of central system cooling equipment performance. As implemented, SEER is estimated to incorporate both weather related influence to the compressor coefficient of performance as well as cycling losses due to equipment operation under part-load conditions. However, for single-speed equipment, SEER is simply estimated as the EER at test condition "B" which consists of an 82°F outdoor and 80°F indoor temperature condition.

$$SEER = EERb\,(1 - 0.5C_D)$$

To obtain SEER, the "B" test condition result is then lowered by a cooling degradation coefficient (C_D) to account for cycling losses, which varies depending on a host of factors: coil thermal capacitance and configuration, refrigerant expansion device, fan time delay and refrigerant control strategies. Air conditioning equipment is usually tested for C_D, with a median cooling value of about 0.09 for typical units. A default value for C_D of 0.25 may be used by manufacturers in lieu of testing, but that option is rarely exercised because of the high default value and the resulting deleterious impact on estimated nameplate SEER (Dougherty 2003).

Although there is considerable scatter in Dougherty's data from 2003 on 322 air-conditioning systems (see Figure 34), the general consensus was the key hardware features influencing C_D are indoor fan (off) delay and whether the high and low sides of the refrigerant circuit equalize quickly during the off cycle. For analysis of the data, the AC industry favored 3 levels (do neither, do one or the other, do both). Dougherty found that breaking the middle section into its two natural parts was marginally justified: fan delay gains you a little more than hardware that delays equalization (e.g., non-bleed TXV, liquid line solenoid, electronic expansion device). The median values for C_D were 0.09 for do neither, 0.07 for do refrigerant control or fan delay and 0.04 for both. Interestingly, the data did not show C_D values to be low only for variable speed or two-stage equipment. Single stage equipment also had low C_D values.

More recently, CDH Energy has analyzed a very large data set of over 5,100 residential split system air conditioners in the online 2006 California Energy Commission (CEC) database (http://www.energy.ca.gov/appliances/appliance/excel_based_files/). The EXCEL file includes manufacturer model information as well as the steady-state capacity and power data at 82°F (Q_{82} and W_{82} at Test B conditions) and at 95°F (Q_{95} and W_{95} at Test A conditions). In addition, the file includes the rated EER (defined as Q_{95}/W_{95}), the degradation coefficient (C_D), and the rated or listed SEER. As with Dougherty's work, this database shows that C_D in modern equipment is 0.1 with no intervention and less with fan delay and prevention of refrigerant migration.

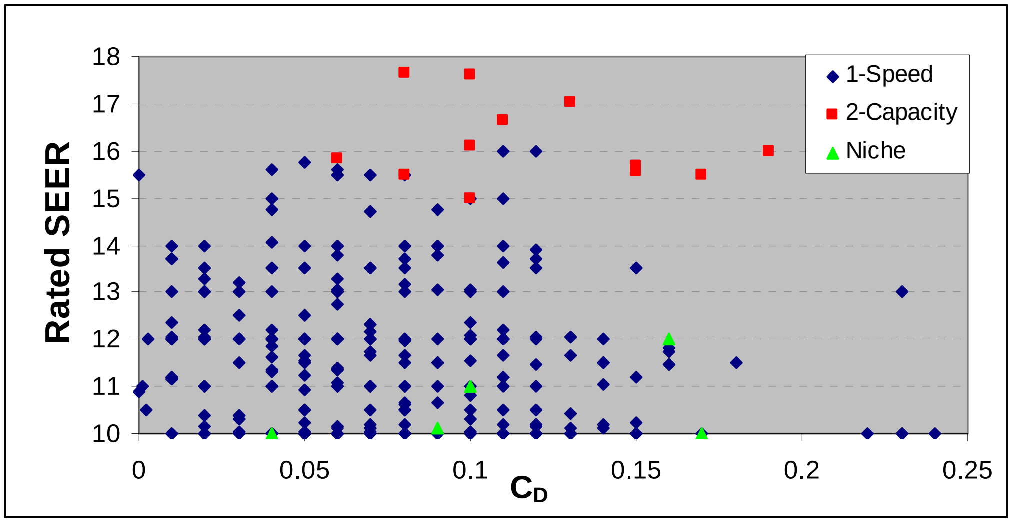

Figure 34. Measured cyclic degradation coefficient for 322 air conditioners by type

Thus, to obtain higher SEER and HSPF ratings, manufacturers have frequently instituted a timed indoor unit fan off delay and control of post-cycle refrigerant migration to achieve C_D values of 0.05 or less. Standard systems have C_D levels of approximately 0.09 even without utilization of these strategies. In general, this means that potential cycling losses on a theoretic basis will be between 2 and 5% under typical conditions ($C_D/2$).

Given the particulars of ARI test condition "B", SEER is also tied to an assumed 80°F indoor condition – at least two degrees higher than the cooling set point commonly observed in air conditioned residences (Parker et al. 2000). The current standards mandate air conditioner efficiency levels using EER and SEER and consumers are typically guided to make energy-wise purchases based on these ratings – the higher the SEER, the more efficient the system. Understandably, manufacturers work to improve the SEER ratings of equipment given the current guidelines. Given the current test procedure, there is strong incentive to produce air conditioning equipment that does best under moderate load conditions (Kavanaugh 2002).

There is also a very large incentive to reduce cycling losses associated with AC performance. This has resulted in manufacturing processes in recent years that have tended to reduce cycling losses through the use of timed indoor unit fan off delay and control of post-cycle refrigerant migration. Thus, modern units may have lower losses in energy efficiency due to system oversizing than seen historically.

Discussion of Interactions

Traditionally, proper sizing of heating and cooling equipment in residences has been viewed as being important to providing residential interior comfort conditions in terms of temperature, humidity and ventilation. Similarly, proper sizing for residential cooling systems has been viewed as particularly vital in order to provide optimal system energy efficiency while maintaining comfort. The conventional view has been that when equipment is oversized, system efficiency is reduced, energy costs increase and interior comfort may be compromised.

Indeed, as noted in the introduction, an earlier study involving early 1990's equipment of over 368 single-system sub-metered homes in Florida found that air conditioners oversized by 120% Manual J incurred a 3.7% increase in annual energy use. There has also been the expectation of poorer humidity control in humid climates where equipment short-cycling can lead to less effective humidity control during the first minutes of cycles where the evaporator is cooling down. However, because of the importance of C_D to SEER, manufacturers have made strenuous efforts to reduce C_D in modern central air conditioners. Older equipment often had a C_D value of around 0.2, implying that cycling losses made efficiency 10% worse due to performance under part-load conditions. Since right-sized equipment can be expected to recoup about half of this impact, the expectation was that better sized equipment would be 5% more efficient than oversized systems.

Because of the emphasis on SEER, data shown in this report as collected by NIST (Dougherty 2003) shows that for a large sample of 2002 vintage air conditioners that C_D is typically 0.09 for standard equipment, 0.07 for standard equipment with a post run fan delay and typically 0.04 for that with a fan delay and solenoid control of refrigerant to prevent post cycle refrigerant migration. Since almost all modern equipment is now shipped with a post run fan delay (an unfortunate fact in hot-humid climates due to the impact on humidity control), this means that cycling losses of modern equipment are typically less than a third of what they were previously – about 3.5% on average. Since only about half of this impact can be recouped through better sizing, the expected theoretical impact is only about 2% on average.

Another previously unaccounted for fact is that whereas machine operational efficiencies are slightly negatively impacted by cycling losses, the greater run-time associated with downsized equipment will mean that duct losses are substantially increased. For instance, a machine downsized by 50% will have approximately 50% greater run-time to serve the same load. These losses include both duct leakage and duct heat gains if the ducts are located in unconditioned zones. For instance, duct leakage and house infiltration rates will be impacted by air handler operation (Cummings et al. 1991). Similarly, duct conduction and heat gains will be highest during periods when the ducts are operating with air flow through the ducts and maximum delta-T across the poorly insulated duct walls. These problems are particularly acute if the duct system is located in attics which can become very hot. Unfortunately, for slab on grade homes, as predominate in the hot climates, ducts are located in the attic space in more than 90% of installations.

Simulation analysis for *ASHRAE Standard 152* (ASHRAE 2001, Gu et al.1998; 2003), and other work shows that losses due to duct air leakage are often about 5-10% of overall cooling load (depending on leakage area and location) and duct heat transfer for conduction gains to ducts in attics is often a similar amount depending on insulation level with typical duct configurations. *Although Manual J, 8th edition accounts explicitly for duct losses, it does not consider how duct losses dynamically interact with system sizing to impact the losses themselves (e.g., <u>duct losses are not static with system sizing and vary proportionately</u>).* These losses are largely proportional to machine runtime; air leakage only occurs when the machine is running. Although duct systems can be re-sized with smaller ducts, the reduction of duct area does not scale linearly with duct air flow. Also as ducts are made smaller, pressures on duct leaks will be increased and fan power will go up.

Impacts of Duct Resizing with Air Conditioner Resizing

To understand the theoretical issue of duct resizing impacts on duct losses, we performed a short theoretical exercise to evaluate the interaction. For instance if a 48,000 Btu/hr air conditioner was oversized by 50%, the right-sized equivalent unit would be a 32,000 Btu/hr unit. Assuming equivalent air flow per unit cooling (400 cfm/ton), the four ton unit would have an air flow of 1,600 cfm while the 2.67-ton unit would have coil air flow of 1,067 cfm. The ratio of these flows is 0.667.

Assuming that the average duct for the original 4-ton unit was properly sized and that the average duct diameter was 10.0 in. for the overall system, the cross-sectional area of the duct would be 78.5 square inches carrying a flow of 1,600 cfm (see Rudd 2003). The circumference of the duct is 34.16 inches so that 200 lineal feet of the duct would amount to a total duct surface area of 569 square feet for the original system.

Assuming that the re-designed system operates at the same air flow velocities and duct pressures (so that leakage is under the same pressure as well), the cross-sectional area of the new system would be 78.5 * 0.667 or 52.4 square inches. This is most closely approximated by an 8 in. duct which has a cross-sectional area of 50.3 square inches. As the lineal distance for the trunk lines and run-outs must be the same (assuming we have the same house), the resulting surface area for the re-designed duct system would be 419 square feet. Thus the duct area for heat transfer would be 74% of the original. Since duct conductance losses are proportional to duct area, the duct conductance losses would be reduced by about 26%. Although perhaps logical, it may be presumptuous to assume that duct air leakage scales linearly with duct area. However, for resized AC systems without resized duct systems (e.g., the four test homes included in this field study), the ducts will operate under lower pressure leading to a lower air leakage rate, but for a greater runtime period.[3]

Under the typical assumption that total duct losses are about 20% of air conditioner peak load during runtime with half of this from conduction, this means that resized ducts would reduce the duct losses associated with re-sized equipment from 20% to 17%. Assuming that leakage was proportional to duct area, the impact would change to 15%. In either case, this exercise shows that although duct re-sizing would help reduce the fact that duct losses increase in proportion to runtime, it would not compensate for the fact that duct losses during runtime are typically much greater (2-3 times larger) than those from machine cycling losses.

Thus, under most circumstances, the losses due to increased runtime from ducts in non-conditioned spaces will be greater than the gains in efficiency from reduced cycling with modern vintage equipment. Of course, these same impacts were present in older equipment, but there the cycling losses were about three times greater than they are with modern equipment. Thus, one fundamental conclusion is that with modern equipment, better cooling performance with properly-sized equipment is likely only to be realized with ducts inside the conditioned space or with well sealed ducts in non-hostile environments (crawlspaces, basements or in attics with cool roof construction).

CONCLUSIONS

Four case studies were performed where over-sized, new air conditioners were replaced with properly-sized systems. These systems were all located in Florida – two in Central Florida (Lakeland and Merritt Island), one in North Florida (Jacksonville) and one in Southwest Florida (North Port[4]). The systems were oversized in each case by 47-65% according to *Manual J 8th Edition*. Each of the systems was installed and then commissioned to make certain they were operating properly. All of the systems had the duct systems located in the attic. Each house had a shingle roof. The Lakeland system had the air handler in the attic; two others had the air handler located in the garage, and one system had the air handler located in an interior utility closet.

Fifteen minute data on air conditioner power, outdoor weather conditions and interior humidity levels were evaluated with each of the AC systems before and after change out. The houses were

[3] In systems without re-sized ducts, airflow rates might average about 25% lower for the smaller systems. For the same ductwork, this translates into static pressures that would be about 45% lower, assuming that pressure drops are proportional to airflow squared in the ductwork. Lower duct static pressure should result in a lower duct leakage rate. However, the aggregate impact depends on the duct operating pressure (at air leakage sites) but also the extent of the increased supply fan run time due to the lower cooling capacity of the smaller systems.

[4] Due to the hurricanes of 2004, late installation of this system made it difficult to extract useful data for use in this report.

occupied, but the homeowners generally did well at maintaining a constant interior thermostat setting.

Outdoor condenser units and indoor air handlers were changed to new smaller systems, although ductwork remained the same. All of the original and replaced systems had single-speed air handlers. These air handlers had multiple speed taps for the permanent split capacitor (PSC) motors. Generally, we found it impossible to match the nominal CFM/ton of cooling capacity in the pre-retrofit system to that in the post system.

For the Lakeland house, the measured indoor humidity averaged 3.6% higher post retrofit. One key factor in the system change out, however, was the fact that single speed air handlers were used. The change out did show energy savings of 8 - 13% – greater than the 3% expected in the difference from SEER between the pre and post retrofit machines. Taken at face value, the change in machine size was associated with an energy savings of 5-10%.

The disparity of the air handler flow was greatest in the Lakeland AC system. Here, the pre-retrofit oversized system had an evaporator air flow of 1,660 cfm or 341 cfm/ton. With the post retrofit system, even choosing the lowest speed tap, the flow was 1,490 cfm or 426 cfm/ton. As shown in other evaluations (Palani et al. 1992; Parker et al. 1997), such a disparity in evaporator coil flow rate can be expected to adversely impact coil temperatures and humidity removal – particularly at the higher flow rate.

Based on earlier work, it appears that most of the lower energy use in the Lakeland system was achieve by trading off the sensible heat ratio (SHR) of the old equipment (lower) against the new equipment (higher). While it would be convenient to attribute this poor interior moisture control to the higher coil air flow, the higher measured condensate in the post period and lower post period outdoor dew points suggest that somehow the moisture load was much greater with the new machine.

If a higher evaporator temperature and lower outdoor dewpoints were reducing the moisture being removed by the air conditioner, then we would expect to see that the moisture removal rates were lower. In fact, we found just the opposite; the new air conditioner actually removed an average of 1.4 additional gallons of water each day from the house after the unit was changed out. Given the lower outdoor dew point in the post period, this means that somehow an increased moisture load was being placed on the air conditioner. We hypothesize that the greater runtime of the new unit (11% increase) resulted in more moisture from the attic being drawn into the air handler located there. This observation also fits the general theory that duct losses and leakage are proportional to machine run time.

The new air-conditioning system in Merritt Island (44.5 kBtu/hr to 29.4 kBtu/hr against a *Manual J* estimate of 30.2 kBtu/hr) showed increased energy use in all three methods of evaluation (long term matched weather periods, matched individual days, and regression). The estimates indicate increases in energy use of 8-18%. The nameplate SEER of the new unit was 2% worse for the new system. Although the condensate measurement was not available to look for evidence of increased duct leakage as seen at the Lakeland site, the long-term profile data clearly showed an increase in late afternoon and early evening AC power use in the post change-out data that would fit the hypothesis of increased attic duct loads being the culprit in the increased energy use with the downsized air-conditioning system. Relative humidity was very similar in the pre and post periods (no statistical difference).

The downsized air-conditioning system in Jacksonville, Florida (47 kBtu/hr to 33 kBtu/hr against a *Manual J* estimate of 28.4 kBtu/hr) also showed slightly elevated energy use in the post period. The three methods showed impacts of 0% saving to a 16% increased energy use. Because the new AC system seemed to maintain an interior temperature about 1°F lower in the post period, the regression analysis method seemed to show the most reliable estimates of the impact which was 0% – no impact. Again, however, the profile plot of the pre and post period showed strongly elevated consumption of the new air-conditioning system between noon and 8 PM as would be expected if increased duct conduction and duct leakage from hot attic conditions were the driving force for impacts. Similarly, the downsized system generally showed lower energy use when the attic was cool between midnight and 7 AM. The interior relative humidity was slightly worse (~1%) in the post period.

The fundamental conclusions from the study in brief:

- Monitored data from three case studies where oversized AC systems were replaced in Florida saw only one case where energy use was lower – and this was in a system where high evaporator flow rates largely traded off moisture removal for sensible heat performance – not appropriate in a hot-humid climate. In the other two systems, energy use was clearly increased in one system and about the same in the other. However, in both of these two systems, comparison of the AC demand profiles showed that the downsized systems indicated increased loads during afternoon and early evening hours in agreement with the hypothesis that duct losses are overwhelming part-load gains to machine performance.

- Relative humidity performance did not appear to be improved by downsizing. However, this appeared largely due to the increase in per ton evaporator coil cfm in the post period. This would argue in the future for retrofits of better sized air conditioners to be tied to variable speed air handlers where the proper coil air flow can be readily selected. Generally, post downsizing moisture removal performance appeared tied to the nominal evaporator air flow per ton of rated capacity. However, greater return side duct leakage due to increased run times from downsized systems can easily overwhelm other factors.

- Energy savings from rightsizing of modern higher efficiency central air-conditioning equipment may be lower than earlier vintage machines due to diminished cycling losses due to manufacturer focus on improvements to SEER which are inexpensively accomplished through the use of post cycle fan delay and suspension of refrigerant migration.

- Potential energy savings from current generation right-sized machines may average 2-3% for systems with sealed duct systems with the ducts located within the conditioned space. In the past this number was 5-7%.[5]

[5] This number is based on the implicit assumptions of typical cooling load factor (load/steady-state capacity) in the SEER procedure of 50%. We further assume that cycling losses can only be reduced by 50% by improved sizing.

- Downsized machines with the duct systems located in attics may see that increases in duct losses substantially exceed the savings in increased air-conditioning system part load performance.

- While AC downsizing does not look to have large ramifications on energy consumption for customers with ducts in attics, it does appear to have potentially larger impacts for utilities during their peak generation periods.

- Downsizing air conditioners in a retrofit situation may be difficult without redesigning the duct systems so that proper air flows can be maintained. The most straightforward solution to this issue is to use variable speed air handlers so that proper flow can be achieved. However reduced air velocities at the supply air grilles may dictate the need to adjust grille blades/dampers or install new grilles to maintain adequate “throw” of the conditioned air and entrainment of room air.

- Practically for future such studies of rightsizing, it is recommended that change-outs be done in mid summer with adequate pre and post data. Given the findings from our case studies, we suggest that a future study should examine right-sized machines in a home with attic ducts and another with interior ducts. Variable speed air handlers should be used so that flow can be made equivalent in the pre and post periods. Data should also be taken on condensate, attic temperature, coil, return and supply temperatures to aid the later data analysis.

While we can emphasize that this represents only three case studies where new air conditioners were properly sized after being downsized, the observations fit with what is indicated by simulation, reinforcing the idea that optimal machine sizing will be strongly impacted by duct system leakage and duct location. Best results will be achieved with sealed duct systems and with ducts inside the conditioned envelope or in crawlspace, basements or beneath cool attic assemblies. Variable speed air handlers allow appropriate choice for coil air flow.

ACKNOWLEDGEMENTS

Thanks to Brian Dougherty with NIST for sharing data on the cooling system cyclic degradation coefficients for tested AC systems.

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APPENDIX A

Air Conditioner Sizing Calculations for Each Case Study Home

Lakeland – Loads

Rhvac - Residential & Light Commercial HVAC Loads
Florida Solar Energy Center
Cocoa, FL 32922-5703

Elite Software Development, Inc.

Page 1

System 1 Summary Loads LAKELAND

Component Description	Area Quan	Sen Loss	Lat Gain	Sen Gain	Total Gain
1A-cm-o: Glazing-Single pane, operable window, clear, metal frame no break, outdoor insect screen with 50% coverage, light color blinds at 45° with 100% coverage	220.4	8,676	0	8,503	8,503
10A-m: Glazing-French door, single pane clear glass, metal frame no break, outdoor insect screen with 100% coverage, light color blinds at 45° with 100% coverage	96	4,968	0	2,568	2,568
1A-cm-o: Glazing-Single pane, operable window, clear, metal frame no break, outdoor insect screen with 100% coverage, light color blinds at 45° with 100% coverage	54	2,126	0	1,124	1,124
1A-cm-o: Glazing-Single pane, operable window, clear, metal frame no break, outdoor insect screen with 100% coverage	9.2	364	0	530	530
10A-m: Glazing-French door, single pane clear glass, metal frame no break, outdoor insect screen with 100% coverage	24	1,242	0	1,210	1,210
11N: Door-Polystyrene Core	42	375	0	345	345
13A-5ocs: Wall-Block, board insulation only, R-5 board insulation, open core, siding finish	1946.8	7,542	0	3,968	3,968
12B-0sw: Part-Frame, R-11 insulation in 2 x 4 stud cavity, no board insulation, siding finish, wood studs	435.6	845	0	845	845
16C-30: Roof/Ceiling-Under attic or knee wall, Vented Attic, No Radiant Barrier, White or Light Color Shingles, Any Wood Shake, Light Metal, Tar and Gravel or Membrane, R-30 insulation	2516.9	2,495	0	3,302	3,302
22A-ph-t: Floor-Slab on grade, No edge insulation, no insulation below floor, tile covering, passive, heavy moist soil	276	11,620	0	0	0
Subtotals for structure:		40,253	0	22,395	22,395
People:	5		1,000	1,150	2,150
Equipment:			0	1,200	1,200
Lighting:	0			0	0
Ductwork:		7,724	1,200	8,188	9,388
Infiltration: Winter CFM: 88, Summer CFM: 44		2,972	1,396	768	2,164
Ventilation: Winter CFM: 0, Summer CFM: 0		0	0	0	0
AED Excursion:		0	0	1,310	1,310
System 1 Load Totals:		50,949	3,596	35,011	38,607

Check Figures

Supply CFM:	1,604	CFM Per Square ft.:	0.637
Square ft. of Room Area:	2,517	Square ft. Per Ton:	647
Volume (ft³) of Cond. Space:	26,349	Air Turnover Rate (per hour):	3.7

System Loads

Total Heating Required With Outside Air:	50,949 Btuh	50.949 MBH
Total Sensible Gain:	35,011 Btuh	91 %
Total Latent Gain:	3,596 Btuh	9 %
Total Cooling Required With Outside Air:	38,607 Btuh	3.22 Tons (Based On Sensible + Latent)
		3.89 Tons (Based On 75% Sensible Capacity)

Notes

Calculations are based on 8th edition of ACCA Manual J.
All computed results are estimates as building use and weather may vary.
Be sure to select a unit that meets both sensible and latent loads.

Thursday, August 26, 2004, 10:09 AM

Merritt Island – Loads

Rhvac - Residential & Light Commercial HVAC Loads
Florida Solar Energy Center
Cocoa, FL 32922-5703

Elite Software Development, Inc.

Page 1

System 1 Summary Loads MERRITT ISLAND

Component Description	Area Quan	Sen Loss	Lat Gain	Sen Gain	Total Gain
10A-m: Glazing-French door, single pane clear glass, metal frame no break, ground reflectance = 0.23, outdoor insect screen with 50% coverage, medium color blinds at 45° with 50% coverage	20.1	1,007	0	574	574
1A-cm-o: Glazing-Single pane, operable window, clear, metal frame no break, outdoor insect screen with 50% coverage, light color blinds at 45° with 100% coverage	96.4	3,675	0	3,235	3,235
1A-cm-o: Glazing-Single pane, operable window, clear, metal frame no break, outdoor insect screen with 100% coverage, light color blinds at 45° with 100% coverage	107.2	4,084	0	2,224	2,224
1B-cm: Glazing-Single pane window, fixed sash, clear, metal frame no break, outdoor insect screen with 100% coverage	25.3	858	0	1,194	1,194
1A-cm-o: Glazing-Single pane, operable window, clear, metal frame no break, outdoor insect screen with 100% coverage	7.5	286	0	208	208
10A-m: Glazing-French door, single pane clear glass, metal frame no break, ground reflectance = 0.23, outdoor insect screen with 100% coverage, medium color blinds at 45° with 50% coverage	40.2	2,014	0	1,021	1,021
11N: Door-Polystyrene Core	17.5	123	0	123	123
13A-5ocs: Wall-Block, board insulation only, R-5 board insulation, open core, siding finish	1424.6	5,341	0	2,723	2,723
12B-0sw: Part-Frame, R-11 insulation in 2 x 4 stud cavity, no board insulation, siding finish, wood studs	380.1	737	0	737	737
16C-19: Roof/Ceiling-Under attic or knee wall, Vented Attic, No Radiant Barrier, White or Light Color Shingles, Any Wood Shake, Light Metal, Tar and Gravel or Membrane, R-19 insulation	2257.2	3,318	0	4,424	4,424
22A-ph: Floor-Slab on grade, No edge insulation, no insulation below floor, any floor cover, passive, heavy moist soil	243	9,900	0	0	0
Subtotals for structure:		31,343	0	16,463	16,463
People:	5		1,000	1,150	2,150
Equipment:			0	1,200	1,200
Lighting:	0			0	0
Ductwork:		5,769	1,442	6,003	7,445
Infiltration: Winter CFM: 102, Summer CFM: 51		3,354	2,109	839	2,948
Ventilation: Winter CFM: 0, Summer CFM: 0		0	0	0	0
System 1 Load Totals:		40,466	4,551	25,655	30,206

Check Figures

Supply CFM:	1,167	CFM Per Square ft.:	0.517
Square ft. of Room Area:	2,257	Square ft. Per Ton:	792
Volume (ft³) of Cond. Space:	20,326	Air Turnover Rate (per hour):	3.4

System Loads

Total Heating Required With Outside Air:	40,466	Btuh	40.466	MBH
Total Sensible Gain:	25,655	Btuh	85	%
Total Latent Gain:	4,551	Btuh	15	%
Total Cooling Required With Outside Air:	30,206	Btuh	2.52	Tons (Based On Sensible + Latent)
			2.85	Tons (Based On 75% Sensible Capacity)

Notes

Calculations are based on 8th edition of ACCA Manual J.
All computed results are estimates as building use and weather may vary.

Thursday, August 26, 2004, 10:07 AM

Jacksonville – Loads

Rhvac - Residential & Light Commercial HVAC Loads
Florida Solar Energy Center
Cocoa, FL 32922-5703

Elite Software Development, Inc.
Page 1

System 1 Main Summary Loads JACKSONVILLE

Component Description	Area Quan	Sen Loss	Lat Gain	Sen Gain	Total Gain
1D-cm-o: Glazing-Double pane, operable window, clear, metal frame no break, outdoor insect screen with 50% coverage, light color blinds at 45° with 100% coverage	187.1	6,187	0	4,856	4,856
1E-cm: Glazing-Double pane window, fixed sash, clear, metal frame no break, outdoor insect screen with 100% coverage	27	708	0	594	594
1D-cm-o: Glazing-Double pane, operable window, clear, metal frame no break, outdoor insect screen with 100% coverage, light color blinds at 45° with 100% coverage	58.8	1,944	0	1,028	1,028
10B-m: Glazing-French door, double pane clear glass, metal frame no break, outdoor insect screen with 100% coverage	40.2	2,215	0	1,200	1,200
11G: Door-Panel	21	431	0	340	340
12B-0bw: Wall-Frame, R-11 insulation in 2 x 4 stud cavity, no board insulation, brick finish, wood studs	1296.4	4,779	0	2,100	2,100
12B-0sw: Part-Frame, R-11 insulation in 2 x 4 stud cavity, no board insulation, siding finish, wood studs	264	512	0	512	512
16B-30: Roof/Ceiling-Under attic or knee wall, Vented Attic, No Radiant Barrier, Dark Asphalt Shingles or Dark Metal, Tar and Gravel or Membrane, R-30 insulation	2272	2,763	0	3,926	3,926
22A-ph-t: Floor-Slab on grade, No edge insulation, no insulation below floor, tile covering, passive, heavy moist soil	223	11,510	0	0	0
Subtotals for structure:		31,049	0	14,556	14,556
People:	5		1,000	1,150	2,150
Equipment:			0	1,200	1,200
Lighting:	0			0	0
Ductwork:		6,600	1,198	6,588	7,786
Infiltration: Winter CFM: 102, Summer CFM: 51		4,254	1,663	1,063	2,726
Ventilation: Winter CFM: 0, Summer CFM: 0		0	0	0	0
System 1 Main Load Totals:		41,903	3,861	24,557	28,418

Check Figures

Supply CFM:	1,117	CFM Per Square ft.:	0.492
Square ft. of Room Area:	2,273	Square ft. Per Ton:	833
Volume (ft³) of Cond. Space:	20,369	Air Turnover Rate (per hour):	3.3

System Loads

Total Heating Required With Outside Air:	41,903 Btuh	41.903 MBH
Total Sensible Gain:	24,557 Btuh	86 %
Total Latent Gain:	~~3,861 Btuh~~	14 %
Total Cooling Required With Outside Air:	28,418 Btuh	2.37 Tons (Based On Sensible + Latent)
		2.73 Tons (Based On 75% Sensible Capacity)

Notes

Calculations are based on 8th edition of ACCA Manual J.
All computed results are estimates as building use and weather may vary.
Be sure to select a unit that meets both sensible and latent loads.

Thursday, August 26, 2004, 10:08 AM

North Port – Loads

Rhvac - Residential & Light Commercial HVAC Loads
Florida Solar Energy Center
Cocoa, FL 32922-5703

Elite Software Development, Inc.
Page 1

System 1 Summary Loads

NORTH PORT

Component Description	Area Quan	Sen Loss	Lat Gain	Sen Gain	Total Gain
1A-cm-o: Glazing-Single pane, operable window, clear, metal frame no break	4	152	0	150	150
1A-cm-o: Glazing-Single pane, operable window, clear, metal frame no break, outdoor insect screen with 50% coverage, light color blinds at 45° with 75% coverage	180.1	6,862	0	5,073	5,073
11N: Door-Polystyrene Core	38.5	344	0	329	329
13A-4ocs: Wall-Block, board insulation only, R-4 board insulation, open core, siding finish	1275.5	5,472	0	3,156	3,156
12B-0sw: Part-Frame, R-11 insulation in 2 x 4 stud cavity, no board insulation, siding finish, wood studs	280.1	544	0	544	544
13A-4ocs: Part-Block, board insulation only, R-4 board insulation, open core, siding finish	81.9	234	0	234	234
16C-30: Roof/Ceiling-Under attic or knee wall, Vented Attic, No Radiant Barrier, White or Light Color Shingles, Any Wood Shake, Light Metal, Tar and Gravel or Membrane, R-30 insulation	2032.8	1,952	0	2,732	2,73[illegible]
22A-ph-t: Floor-Slab on grade, No edge insulation, no insulation below floor, tile covering, passive, heavy moist soil	200	8,149	0	0	0
Subtotals for structure:		23,709	0	12,218	12,218
People:	4		800	920	1,720
Equipment:			0	1,200	1,200
Lighting:	0			0	0
Ductwork:		3,953	1,144	4,187	5,331
Infiltration: Winter CFM: 43, Summer CFM: 0		1,417	0	0	0
Ventilation: Winter CFM: 50, Summer CFM: 50		1,648	1,744	934	2,678
System 1 Load Totals:		30,727	3,688	19,459	23,147

Check Figures

Supply CFM:	843	CFM Per Square ft.:	0.415
Square ft. of Room Area:	2,032	Square ft. Per Ton:	940
Volume (ft³) of Cond. Space:	19,777	Air Turnover Rate (per hour):	2.6

System Loads

Total Heating Required With Outside Air:	30,727 Btuh	30.727	MBH
Total Sensible Gain:	19,459 Btuh	84	%
Total Latent Gain:	3,688 Btuh	16	%
Total Cooling Required With Outside Air:	23,147 Btuh	1.93	Tons (Based On Sensible + Latent)
		2.16	Tons (Based On 75% Sensible Capacity)

Notes

Calculations are based on 8th edition of ACCA Manual J.
All computed results are estimates as building use and weather may vary.
Be sure to select a unit that meets both sensible and latent loads.

Thursday, August 26, 2004, 10:09 AM

APPENDIX B

ARI Performance for the Original and Properly-Sized Air Conditioning Systems

Lakeland – Original

Unitary Air-Conditioner

Program & Product Identifier

Today's Date	07/29/2004 12:15:48 PM
Created Date:	
Last Modified Date:	04/09/2003 at 04:04 PM
Status:	Active
Obsolete:	No
Manufacturer:	COMFORTMAKER
ARI Reference Number:	179595
ARI Program:	AC
ARI Type:	RCU-A-CB
Trade/Brand Name:	CAC Series
NAECA Designation:	Yes
Outdoor Model Number:	CAC260A(G)KA**
Indoor Unit(s):	FCX60****
Ventilation Rate:	

Product Performance Ratings

ARI Rating

Cooling Capacity (Btuh):	58500
SEER Rating (Cooling):	11.60

Footnotes

⇦ Back

MANUFACTURER CERTIFIED TO ARI AS COMPLYING WITH CERTIFICATION SECTION OF ARI STANDARD 210

Data management & Internet services are provided by Intertek

ARI

Lakeland – New

Unitary Air-Conditioner

Program & Product Identifier

Today's Date	09/22/2004 10:49:02 AM
Created Date:	
Last Modified Date:	at
Status:	Active
Obsolete:	No
Manufacturer:	COMFORTMAKER
ARI Reference Number:	451613
ARI Program:	AC
ARI Type:	RCU-A-CB
Trade/Brand Name:	CAC Series
NAECA Designation:	Yes
Outdoor Model Number:	CAC242A(G)KC*
Indoor Unit(s):	EBX48****
Ventilation Rate:	

Product Performance Ratings

ARI Rating

Cooling Capacity (Btuh):	42000
SEER Rating (Cooling):	12.00

Footnotes

⇦ Back

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Merritt Island – Original

Certificate of ARI-Certified Performance

The following

Single Phase, Split System: Air-Cooled Condensing Unit, Coil with Blower

Outdoor Unit Model Number: 2TTR2042B1

combined with

Indoor Unit Model Number: TWE048P13

manufactured by: THE TRANE COMPANY

under the Trade/Brand Name: XR12

has been rated in accordance with

ARI Standard 210/240-2005 for UNITARY AIR-CONDITIONING AND AIR-SOURCE HEAT PUMP EQUIPMENT

and is certified by the Air-Conditioning and Refrigeration Institute to meet

the following product performance ratings:

Cooling Capacity (Btuh):	44500
SEER Rating (Cooling):	12.75

* Voluntarily revised, unless accompanied with a WAS in which case the change is involuntary.

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Through Technician Certification by NATE

ARI Reference #:	316982
Today's Date:	08 / 29 /06
Status:	Discontinued

CERTIFIED RATINGS ARE VALID ONLY FOR THE PARTICULAR COMBINATION OF INDOOR AND OUTDOOR UNITS LISTED IN THE AIR-CONDITIONING AND REFRIGERATION INSTITUTE'S DIRECTORY OF CERTIFIED EQUIPMENT. VISIT WWW.ARIDIRECTORY.ORG TO VERIFY THAT THIS COMBINATION IS AN ACTIVE LISTING AND THE DATA LISTED ON THIS CERTIFICATE IS ACCURATE. SEARCH ON THE ARI REFERENCE # TO QUICKLY LOCATE THIS COMBINATION IN THE DIRECTORY.

TERMS AND CONDITIONS

Merritt Island – New

Certificate of ARI-Certified Performance

The following

Single Phase, Split System: Air-Cooled Condensing Unit, Coil with Blower

Outdoor Unit Model Number: 2TTR2030A1

combined with

Indoor Unit Model Number: TWE030P13

manufactured by: THE TRANE COMPANY

under the Trade/Brand Name: XR12

has been rated in accordance with

ARI Standard 210/240-2005 for UNITARY AIR-CONDITIONING AND AIR-SOURCE HEAT PUMP EQUIPMENT

and is certified by the Air-Conditioning and Refrigeration Institute to meet

the following product performance ratings:

Cooling Capacity (Btuh):	29400
SEER Rating (Cooling):	12.50

* Voluntarily revised, unless accompanied with a WAS in which case the change is involuntary.

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Through Technician Certification by NATE

ARI PERFORMANCE CERTIFIED

ARI Reference #:	266874
Today's Date:	08 / 29 /06
Status:	Discontinued

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TERMS AND CONDITIONS

Jacksonville – Original

Certificate of ARI-Certified Performance

The following

Single Phase, Split System: Heat Pump with Remote Outdoor Unit-Air-Source

Outdoor Unit Model Number: 12HPB48-*P

combined with

Indoor Unit Model Number: CB29M-51*P

manufactured by: LENNOX INDUSTRIES, INC.

under the Trade/Brand Name: Value 12

has been rated in accordance with

ARI Standard 210/240-2005, Unitary Air-Conditioning and Air-Source Heat Pump Equipment

and is certified by the Air-Conditioning and Refrigeration Institute to meet

the following product performance ratings:

Cooling Capacity (Btuh):	47000
SEER Rating (Cooling):	12.05
Heating Capacity (Btuh) @ 47 °F:	46000
Region IV HSPF Rating (Heating):	7.3
Heating Capacity (Btuh) @ 17 °F:	29000

* Voluntarily revised, unless accompanied with a WAS in which case the change is involuntary.

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ARI PERFORMANCE CERTIFIED

ARI Reference #:	29808
Today's Date:	08 / 29 /06
Status:	Discontinued

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TERMS AND CONDITIONS

Jacksonville – New

Certificate of ARI-Certified Performance

The following

Single Phase, Split System: Heat Pump with Remote Outdoor Unit-Air-Source

Outdoor Unit Model Number: 12HPB36-*P

combined with

Indoor Unit Model Number: CB29M-41*P

manufactured by: LENNOX INDUSTRIES, INC.

under the Trade/Brand Name: Value 12

has been rated in accordance with

ARI Standard 210/240-2005, Unitary Air-Conditioning and Air-Source Heat Pump Equipment

and is certified by the Air-Conditioning and Refrigeration Institute to meet

the following product performance ratings:

Cooling Capacity (Btuh):	33000
SEER Rating (Cooling):	12.05
Heating Capacity (Btuh) @ 47 °F:	33400
Region IV HSPF Rating (Heating):	7.5
Heating Capacity (Btuh) @ 17 °F:	21200

* Voluntarily revised, unless accompanied with a WAS in which case the change is involuntary.

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ARI Reference #:	**530949**
Today's Date:	**09 / 07 /06**
Status:	**Discontinued**

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TERMS AND CONDITIONS

North Port – Original

Unitary Air-Source Heat Pump

Program & Product Identifier	
Today's Date	07/30/2004 04:08:16 PM
Created Date:	
Last Modified Date:	10/17/2003 at 10:49 AM
Status:	Deleted after next directory
Obsolete:	No
Manufacturer:	RUUD AIR CONDITIONING DIVISION
ARI Reference Number:	229697
ARI Program:	HP
ARI Type:	HRCU-A-CB
Trade/Brand Name:	Ruud UPMC Series
NAECA Designation:	Yes
Outdoor Model Number:	UPMC-042JA
Indoor Unit(s):	UBHK-24+RCHJ-48A1
Ventilation Rate:	
Product Performance Ratings	
ARI Rating	
Cooling Capacity (Btuh):	41000
SEER Rating (Cooling):	13.20
Heat Pump Only	
High Temp 47° F	
Heating Capacity (Btuh):	42000
Region IV HSPF Rating:	8.65
Low Temp 17° F	
Heating Capacity (Btuh):	27000
Sound Rating	
ARI Sound Rating (dB):	
Footnotes	

⇦ Back

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North Port – New

Certificate of ARI-Certified Performance

The following

Single Phase, Split System: Heat Pump with Remote Outdoor Unit-Air-Source

Outdoor Unit Model Number: RPMD-030JAZ

combined with

Indoor Unit Model Number: RBHK-21+RCHA-36A1

manufactured by: RHEEM MANUFACTURING COMPANY

under the Trade/Brand Name: Rheem RPMD Series

has been rated in accordance with

ARI Standard 210/240-2005, Unitary Air-Conditioning and Air-Source Heat Pump Equipment

and is certified by the Air-Conditioning and Refrigeration Institute to meet

the following product performance ratings:

Cooling Capacity (Btuh):	27800
SEER Rating (Cooling):	13.75
Heating Capacity (Btuh) @ 47 °F:	27800
Region IV HSPF Rating (Heating):	8.25
Heating Capacity (Btuh) @ 17 °F:	16600

* Voluntarily revised, unless accompanied with a WAS in which case the change is involuntary.

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ARI PERFORMANCE CERTIFIED

ARI Reference #:	474724
Today's Date:	08 / 28 /06
Status:	Discontinued

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