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Performance of Variable Capacity Heat Pumps in a Mixed Humid Climate

April 2012

Prepared by

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Energy and Transportation Science Division

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NOMENCLATURE

α	alpha level
E	Energy
ECM	Electronically Commutated Motor
EER	Energy Efficiency Ratio
EEV	Electronic Expansion Valve
ϵ	uncertainty error
HP	Heat Pump
HSPF	Heating Seasonal Performance
k	predictor variables
n	number of samples
OAT	Outdoor Air Temperature
s	standard deviation
SEER	Seasonal Energy Efficiency Ratio
w.c.	water column
X	Independent Variable Matrix

Subscripts

p	predicted
TMY	typical meteorological year

EXECUTIVE SUMMARY

Variable capacity heat pumps represent the next wave of technology for heat pumps. In this report, the performance of two variable capacity heat pumps (HPs) is compared to that of a single or two stage baseline system. The units were installed in two existing research houses located in Knoxville, TN. These houses were instrumented to collect energy use and temperature data while both the baseline systems and variable capacity systems were installed. The homes had computer controlled simulated occupancy, which provided consistent schedules for hot water use and lighting.

The temperature control and energy use of the systems were compared during both the heating and cooling seasons. Multiple linear regression models were used along with TMY3 data for Knoxville, TN in order to normalize the effect that the outdoor air temperature has on energy use. This enables a prediction of each system's energy use over a year with the same weather.

The first system was a multi-split system consisting of 8 indoor units and a single outdoor unit. This system replaced a 16 SEER single stage HP with a zoning system, which served as the baseline. Data was collected on the baseline system from August 2009 to December 2010 and on the multi-split system from January 2011 to January 2012. Soon after the installation of the multi-split system, some of the smaller rooms began over-conditioning. This was determined to be caused by a small amount of continuous refrigerant flow to all of the indoor units when the outdoor unit was running regardless of whether they were calling for heat. This, coupled with the fact that the indoor fans run continuously, was providing enough heat in some rooms to exceed the set point. In order to address this, the indoor fans were disabled when not actively heating per the manufacturer's recommendation. Based on the measured data, the multi-split system was predicted to use 40% more energy in the heating season and 16% more energy in the cooling season than the baseline system, for the typical meteorological year weather data. The AHRI ratings indicated that the baseline system would perform slightly better than the multi-split system, but not by as large of a margin as seen in this study. The multi-split system was able to maintain more consistent temperature throughout the house than the baseline system, but it did allow relative humidity levels to increase above 60% in the summer.

The second system was a split system with an inverter driven compressor and a single ducted air handler. This unit replaced a 16 SEER two stage HP with a zoning system. Data was collected on the baseline system from July 2009 to November 2010 and on the ducted inverter system from December 2010 to January 2012. The ducted inverter system did not offer a zone controller, so it functioned as a single zone system. Due to this fact, the registers had to be manually adjusted in order to better maintain consistent temperatures between the two levels of the house. The predicted heating season energy use for the ducted inverter system, based on the measured energy use, was 30% less than that of the baseline system for the typical meteorological year. However, the baseline system was unable to operate in its high stage due to a wiring issue with the zone controller. This resulted in additional resistance heat use during the winter and therefore higher energy use than would be expected in a properly performing unit. The AHRI ratings would indicate that the baseline system would use less energy than the ducted inverter system, which is opposite to the results of this study. During the cooling season, the ducted inverter system was predicted to use 23% more energy than the baseline system during the typical meteorological year. This is also opposite of the results expected by comparing the AHRI ratings. After a detailed comparison of the ducted inverter system's power use compared to that of a recently installed identical system at a retro-fit study house, there is concern that the unit is not operating as intended. The power use and cycles indicate that the unit is performing more like a single stage unit than a variable capacity unit. Analysis of the data indicates that a change in operating behavior occurred during a service call shortly after the installation of the unit. The logbook only indicates that refrigerant charge was added, but does not indicate any other change. This is being investigated further.

While the energy comparison results of these two variable capacity heat pumps is generally underwhelming, it is difficult to draw any hard conclusions about the maximum attainable efficiency of these units when optimally installed. Both units appear to have undesirable conditions associated with the installation or operation, which could have had an adverse effect on their energy use. This highlights the need for careful system design, installation, and servicing, in order to achieve the maximum energy efficiency possible from any given system. In order to gain a better understanding of the absolute performance possible with variable capacity systems, detailed air-side and refrigerant-side measurements will be made on additional variable capacity heat pumps. The results of this analysis will be contained in a future report.

1. INTRODUCTION

Variable capacity heat pumps with inverter driven compressors are on the cutting edge of technology for residential heating and cooling in the United States. Their major benefits include the ability to vary the compressor speed and output capacity in order to match the load on the house, thereby reducing cycling losses, and the ability to provide higher heating capacities at lower temperatures by running the compressor at higher speeds. This results in less reliance on comparably inefficient electric resistance heat.

Two variable capacity heat pump systems are compared to conventional single and two stage heat pumps in this study. Details of the systems are listed in Table 1. One system is a multi-split heat pump system that consisted of eight indoor units, rated at a total of 4.75 tons, paired with a 4 ton outdoor unit. This unit was installed in the retrofit research house, identified as "CC2", and replaced a single stage16 SEER heat pump with a zoning system splitting the house into two zones, upstairs and downstairs.

The second system analyzed was a ducted inverter system that consisted of a 2 ton outdoor unit with an inverter driven compressor paired with a single ducted indoor air handler that is more traditional in the United States. This system was installed in a home designed to maximize energy efficiency, which is identified as "CC3", and replaced a two stage 16 SEER heat pump with a zoning system splitting the house into two zones, upstairs and downstairs.

Both of these homes are unoccupied and have computer controlled, simulated occupancy to control the lights, shower, dishwasher, clothes washer, dryer, and refrigerator doors. The performance and construction of these homes are discussed in detail in Christian et al. (2010).

Table 1: Equipment Models and AHRI Ratings

	CC2 Baseline System	CC2 Multi-Split System	CC3 Baseline System	CC3 Ducted Inverter System
Indoor Unit Size(s) (Btu/hr)	42000	3 @ 9000 5 @ 6000	36000	24000
Outdoor Unit Size (Btu/hr)	36000	48000	24000	24000
AHRI Cooling Capacity (Btu/hr)	34600	48000	24000	24000
AHRI EER Rating (Btu/W)	13.00	8.30	12.50	12.50
AHRI SEER Rating (Btu/W)	16.00	15.00	16.00	18.00
AHRI Heating Capacity @ 47°F (Btu/hr)	34400	54000	23000	27000
AHRI Heating Capacity @ 17°F (Btu/hr)	21000	33000	15000	14500
AHRI Region IV HSPF Rating (Btu/W)	9.75	8.70	9.50	8.89

2. BACKGROUND

The performance of variable speed mini-split heat pumps with inverter driven compressors has been evaluated in a laboratory environment in Winkler (2011). The major conclusions were that the mini-split systems operating at intermediate capacity provided cooling performance similar to that of a high-end two stage conventional unit operating in low stage. However, the performance of high end two stage conventional units was 10%-15% higher than that of the mini-split systems at maximum capacity. It was also found that the mini-split systems tested had higher than usual cycling degradation coefficients when cycling on and off during low load conditions, which would be experienced in actual installations. Determining the real life impact of cycling losses and variable capacity operation requires monitoring the units when installed in an actual house.

3. ANALYSIS APPROACH

The performance of the replacement systems was compared to that of the baseline systems by analyzing their energy consumption in both the heating and cooling modes and annually. In order to account for variances in outdoor air temperature (OAT), a linear regression model was developed for each system based on the independent variable of heating degree days (HDD) or cooling degree days (CDD) as the season would dictate. Since the standby energy use of each system is a known value with very little variability, its value was subtracted from all of the energy use data points and the linear regression model was calculated with a y-intercept of 0. The base temperature for the HDD calculations was taken as the highest average OAT at which the system supplied heat. Likewise, the base temperature for the CDD calculation was taken as the lowest average OAT at which the system delivered cooling. This allows the linear regression model to only fit data in which there was active heating or cooling demand. After the linear regression model was determined with a 0 y-intercept, the measured standby energy was substituted back in as the appropriate y-intercept. This ensures that the standby energy use of the system is accurately captured for days with zero degree days, while not constraining the linear regression model to fitting these values. Higher order HDD or CDD variables were added if their p-value was less than 0.05. This indicates a less than 5% likelihood that results this extreme would be encountered if the independent variable truly had no impact on the energy use of the HP. The residuals of the linear regression model were visually checked for homoscedasticity, Figures located in Appendix A, in order to ensure that the error associated with the model was being accurately represented. In cases where the residuals were obviously heteroscedastic, a transformation was applied that produced homoscedastic results. The 95% confidence prediction intervals are also calculated based on equation 1 (Reliasoft 2011), which takes into account both the error from the fitted model and the error associated with future observations, and plotted along with the linear regression model.

Once a linear regression model was obtained for each system, the models were applied to TMY3 data for Knoxville, TN. The predicted energy use represents the average energy use of the system for the average weather data in the area. The uncertainty of the prediction was calculated as shown by equations 2 and 3 (Reliasoft 2011).

$$\epsilon_p = \pm t_{\frac{\propto}{2},n-k} \sqrt{s^2 \left(1 + X_p^{\mathrm{T}} \left(X^{\mathrm{T}} X\right)^{-1} X_p\right)} \quad (1)$$

$$s_p = \sqrt{s^2 \left(1 + X_p^{\mathrm{T}} \left(X^{\mathrm{T}} X\right)^{-1} X_p\right)} \quad (2)$$

$$\epsilon_{TMY} = \pm t_{\frac{\propto}{2},n-k} \sqrt{\sum_i^n s_{p,i}^2} \quad (3)$$

4. CC2 MULTI-SPLIT SYSTEM

4.1 BASELINE SYSTEM

The baseline system in CC2 was a single stage HP with an ECM blower motor that was connected to a zone controller with 2 zones (upstairs and downstairs). Resistance heat of 9.5kW was supplied as auxiliary heat.

The combination of a single stage HP with an ECM blower motor and a zoning system is a unique combination. The ECM blower motor is designed to provide a near constant airflow regardless of the external static pressure on the system. It accomplishes this by increasing or decreasing the fan speed as required. The combination of a single speed HP with an ECM blower motor and a zoned system highlights the need for careful duct design. In this type of system, the ducts for each individual zone must be sized with respect to the total airflow of the unit. This was not the case for the unit at CC2. While the external static pressure with both zones open was ~0.65" w.c. (which is high, but not extreme), the external static pressure of the upstairs and downstairs zones individually was ~0.9" w.c. In order to cope with this restrictive ductwork, the ECM motor ran faster and therefore used more power, up to 600 W.

4.2 INSTALLATION

In November of 2010 the installation of a multi-split system began at CC2. The installation included 8 indoor units paired with a 4 ton outdoor unit. The outdoor unit was connected to two branch boxes that housed the EEVs for the 8 indoor units as well as some of the electrical controls. The refrigerant lines for the individual units were run from the branch boxes either through the garage or along the outside of the house (Figure 1). The installers indicated that this was their typical practice due to its convenience and lower cost compared to running the lines through the interior of the house while maintaining aesthetics acceptable to customers. This practice, unfortunately, exposes large lengths of refrigerant lines to outdoor temperatures. While the lines were insulated, the insulation is only around R-4. This issue would typically only affect the heating mode in a traditional HP system with expansion valves located at both the indoor and outdoor unit; however this is not the case with this mini-split system. Since all of the expansion valves are located in the garage, the refrigerant lines running outside carry low pressure two-phase refrigerant that is typically around 50°F in the cooling mode. One touted benefit of mini-split systems is that distribution losses from a duct system are eliminated. However, the same considerations that apply to a quality duct system still need to be applied to a mini-split's refrigerant distribution system.

Figure 1: CC2 Multi-Split Refrigerant Distribution Lines

4.3 CONTROLS

At the time of installation, no residential focused centralized controller was available for the multi-split system. Due to the desire to perform setback testing and the inability to program schedules with the standard remote controls for the indoor units, a central thermostat designed for commercial applications was installed. This presented quite a few hurdles with getting the system commissioned properly. Additionally, when the system did work properly, the set point temperature could only be set to odd numbered degrees. We also experienced problems with the units not turning back on after brief power interruptions. This required shutting off the power for an extended period of time in order to get the units to reset properly. For the duration of the test, the fans on the indoor units were set to the automatic setting. This had to be performed with the individual unit remotes, since the central controller did not support the automatic fan speed setting.

4.4 HEATING SEASON

During the heating season, the rooms upstairs began to over-condition and heat beyond the set point. This is believed to have been due to the fact that the compressor could not modulate down to a low enough speed to serve only a small number of indoor units. In order to cope with this, the expansion valves on units that are not calling for heat are never fully closed. This results in a portion of hot refrigerant being directed to units in rooms that are not calling for heat whenever the compressor is running. This fact, coupled with the continuous operation of the fans, provided enough capacity to over-condition some of the rooms. In order to address this issue, the continuous fan operation of the indoor units was disabled at the recommendation of the manufacturer. This helped the over-conditioning problem, but also resulted in 2 phase refrigerant leaving the indoor units and most likely a reduction in heating output (albeit to rooms that were not calling for heat anyways).

4.4.1 Temperature Control

Figure 2 shows the temperatures near each thermostat of the baseline system as well as the total system power during a cold day with a 23.0°F average OAT. The temperature on both levels varied by ~1.5°F during each cycle, and the temperature on the 1st floor stayed about 1°F higher than the 2nd floor. The high power use indicates that auxiliary heat was being used frequently by the system. Figure 3 shows the same temperatures and the system power use during a similar day with respect to average OAT and insolation for the multi-split system. The multi-split system maintains a much tighter temperature band and runs continuously, which is expected of a HP with an inverter driven compressor. This fact should lead to better comfort for the home occupants.

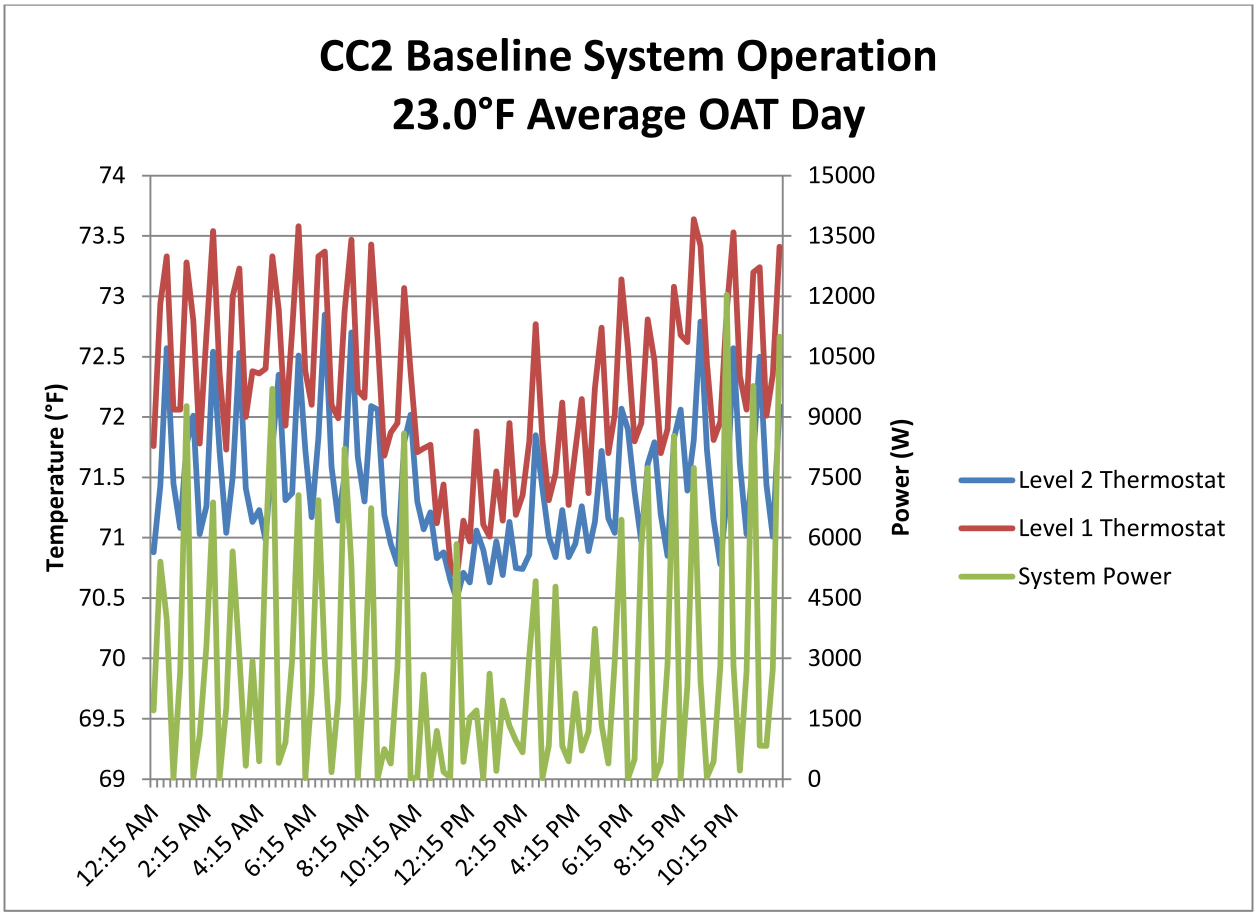

Figure 2: CC2 Baseline System Cycling - 23.0°F Average OAT Day

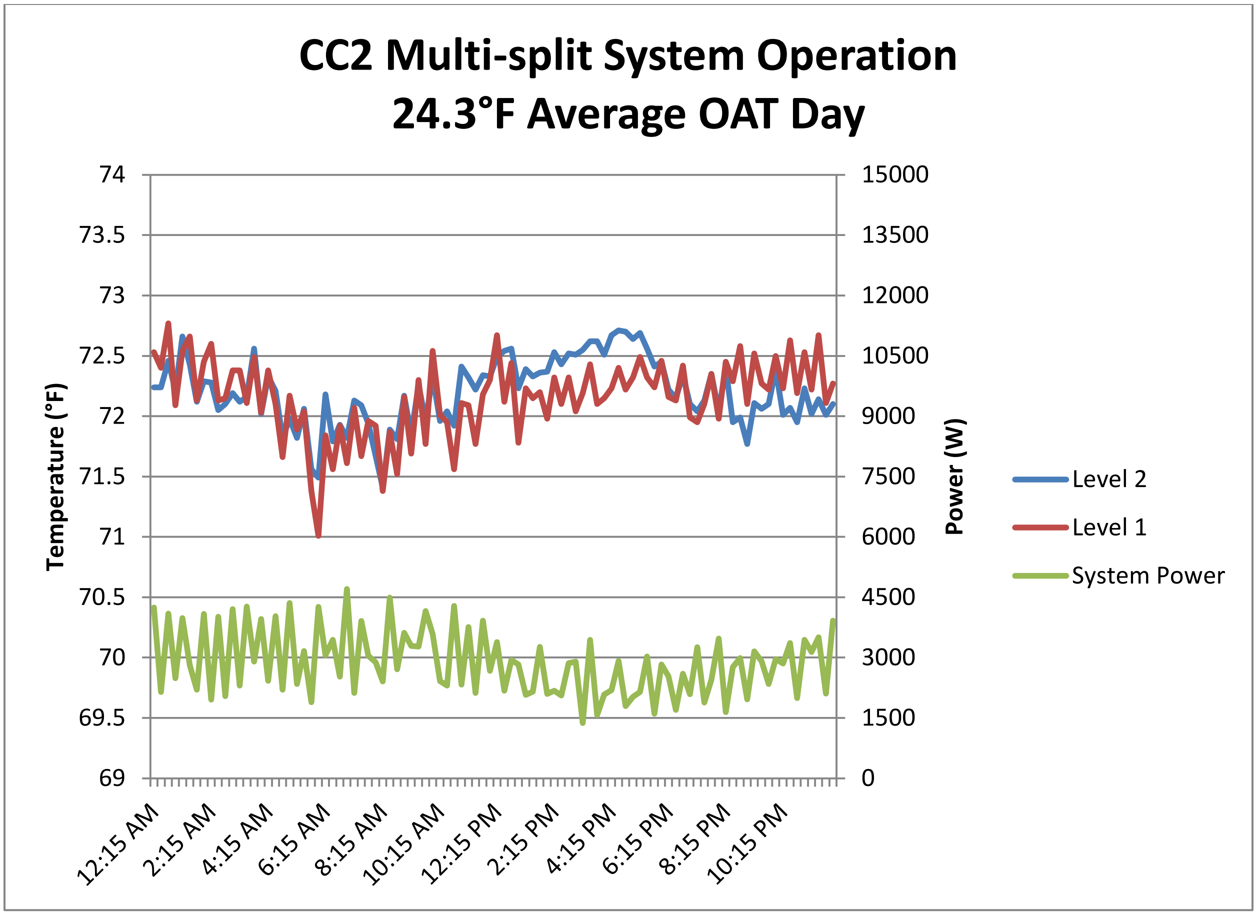

Figure 3: CC2 Multi-split System Operation - 24.3°F Average OAT Day

Figure 4 shows the mean radiant temperatures for each room while the baseline system was operating during the same day that was used in Figure 2. There is about a 2°F difference in the mean radiant temperature between many of the rooms at any given time, in addition to temperature swings during cycles of up to 2.5°F. Despite the system being zoned, the individual room temperatures still require the ductwork to be properly balanced in order to maintain similar average temperatures. The mean radiant temperatures of the individual rooms during the same day as shown in Figure 3 are plotted for the multi-split system in Figure 5. In general, the mean radiant temperatures show more variation than the temperatures in the upstairs' hallway and dining room (where the baseline system's thermostats were located), shown in Figure 3. However, with the exception of the master bedroom, the variation is smaller than that of the baseline system. The use of 8 units in the house ensures that each room's temperature is monitored and maintained. Comparing the temperatures during heating between the two systems indicates that the multi-split system consequently maintained a higher average temperature throughout the house compared to the baseline system. This will penalize the energy use of the multi-split system slightly.

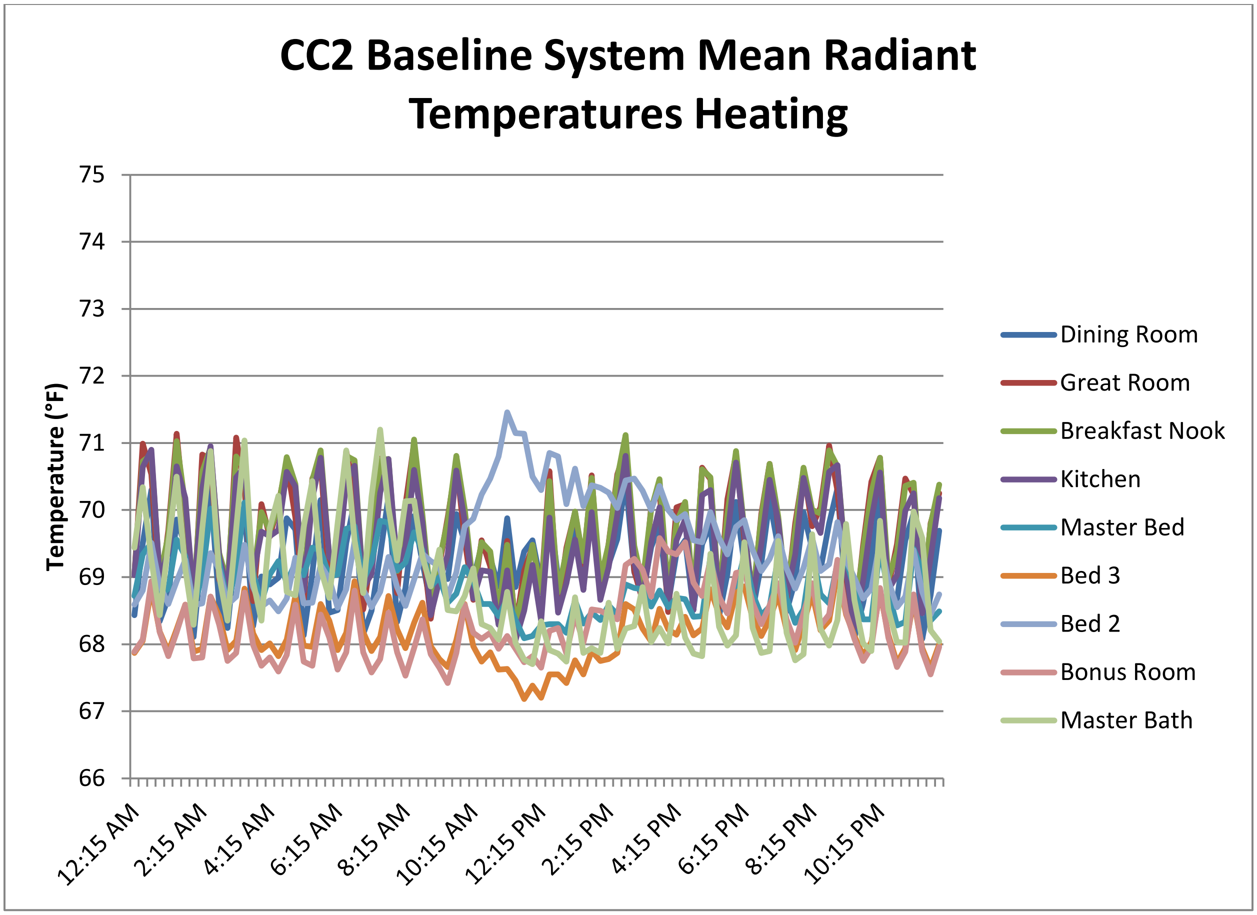

Figure 4: CC2 Baseline System Mean Radiant Temperatures - Heating

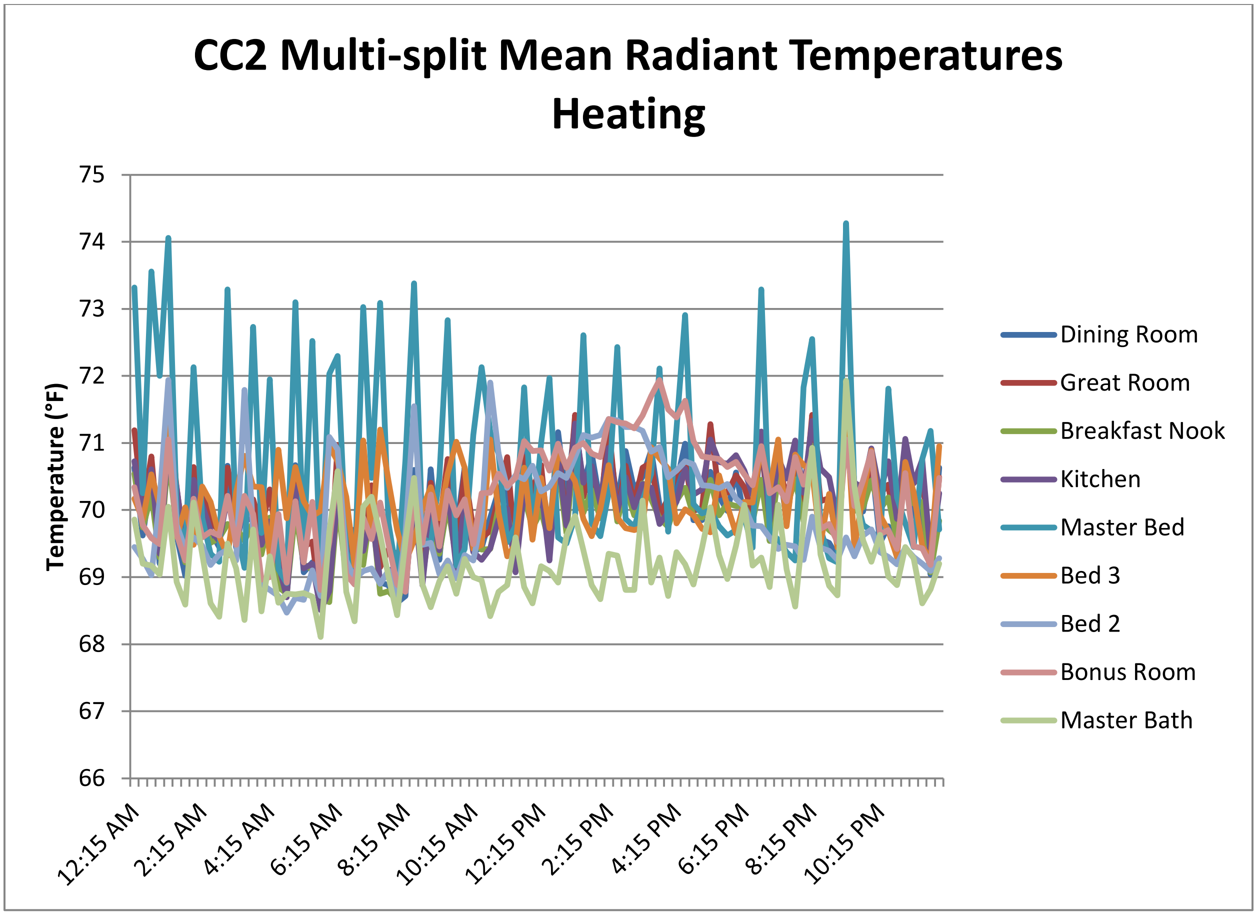

Figure 5: CC2 Multi-split Mean Radiant Temperatures - Heating

4.4.2 Linear Regression Models

Figure 6 shows a plot of the heating season data for the baseline unit as well as the linear regression model.

A plot of residuals from the baseline linear regression versus HDD showed that they are heteroscedastic and increase with HDDs, see Appendix A. In order to correct this, the energy use and independent variables were transformed by dividing by the HDD plus 4. The resulting residuals were much more unifromly distributed and a plot of these can also be found in the Appendix. Once the transformed linear regression model was transformed back into standard units, it closely matched that of the original linear regression, as shown in Figure 6, and provides a standard deviation of 0.157 kW·h/d·(HDD+4).

Figure 7 shows a plot of the energy used by the multi-split system as well as the energy use predicted by the linear regression model. The energy use is much more linear than the baseline with respect to the outdoor air temperature, which is expected due to its lack of auxiliary heat.

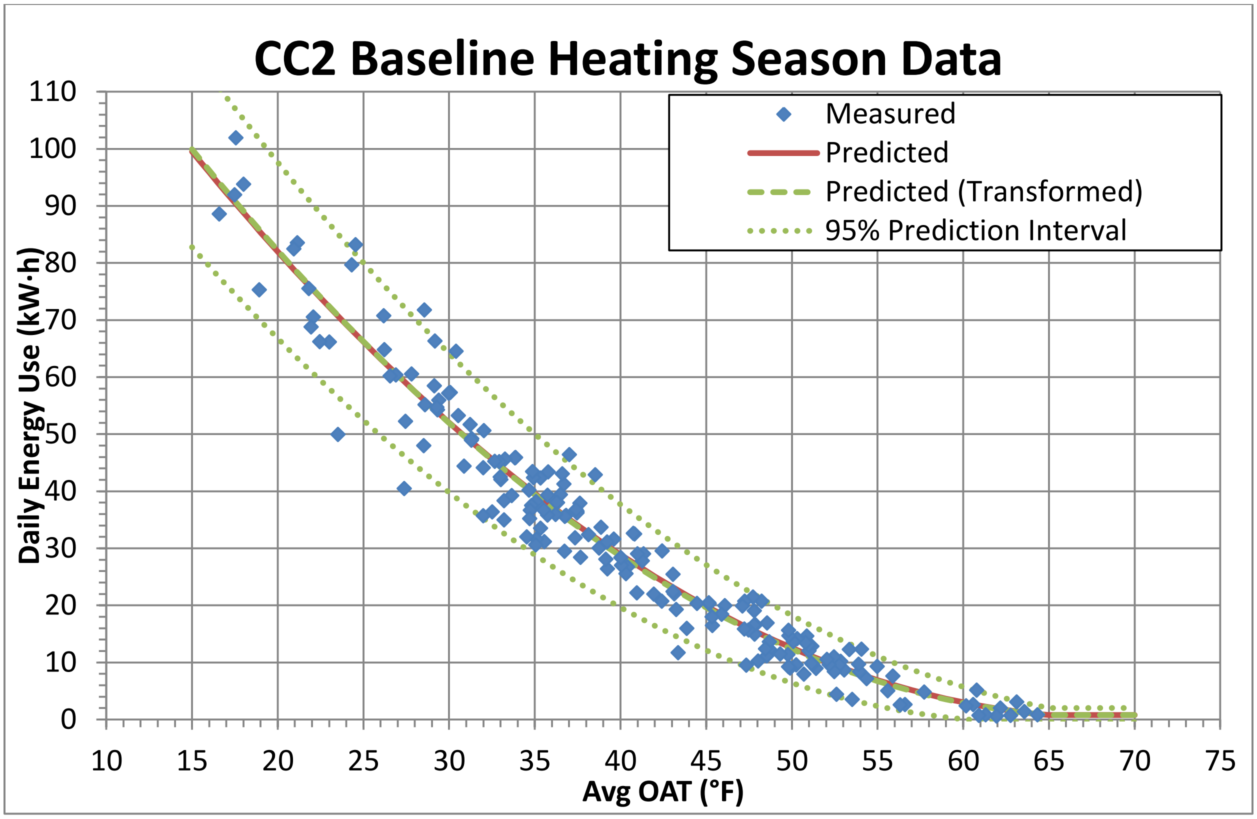

Figure 6: CC2 Baseline Heating Season Data and Linear Regression

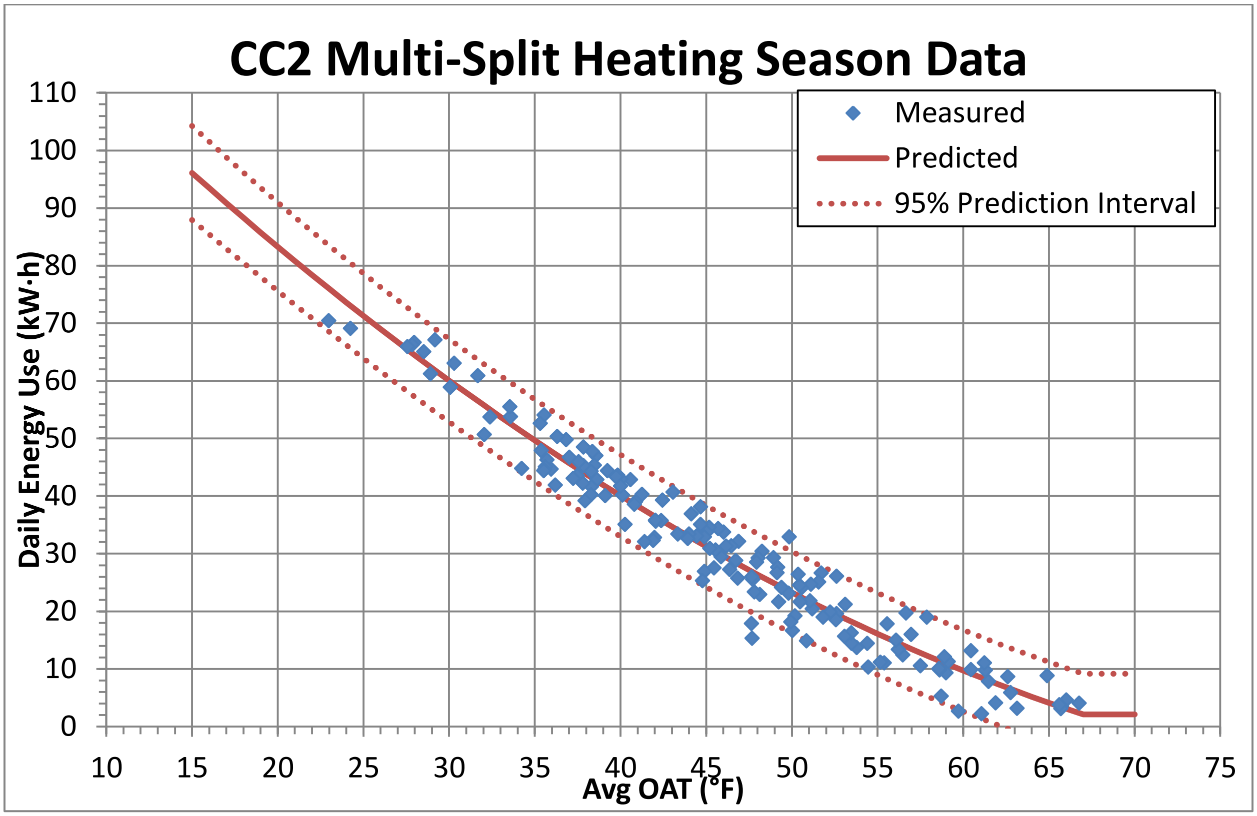

Figure7: CC2 Multi-Split Heating Season Data and Linear Regression

4.4.3 Performance Comparison

Plotting the two multiple linear regression models on the same graph gives some insight into how the units performed as seen in Figure 8. One observation is that the multi-split system uses over twice the amount of energy when in standby. This is not unexpected as the multi-split system has significantly more electronics to power. It is also seen that on average the multi-split system used more energy while heating the house than the baseline system during days with an average OAT greater than 20°F. It is only when the temperature drops below ~19°F that the multi-split system begins to use less energy than the baseline system due to its increasing use of auxiliary heat in order to supplement the reduced heating capacity.

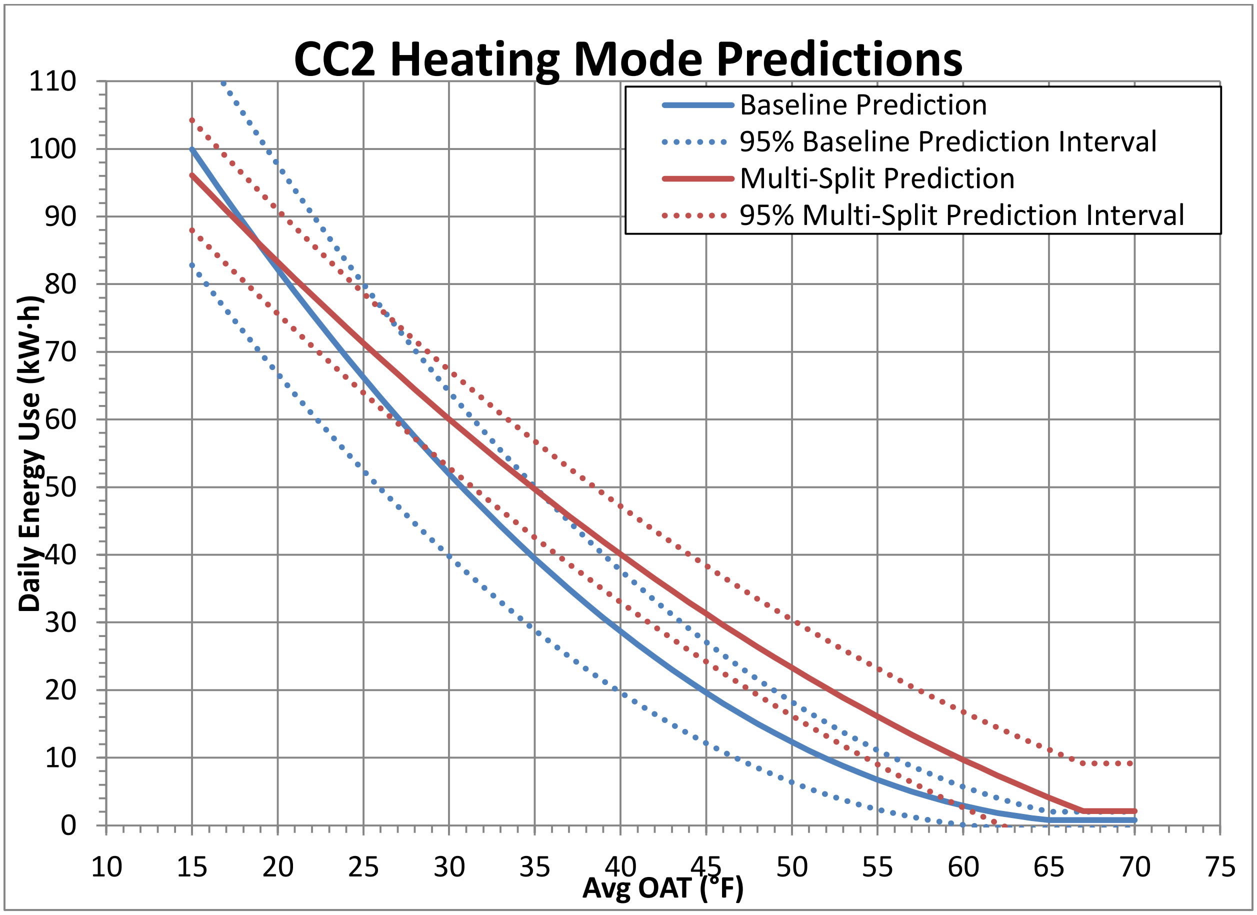

Figure 8: CC2 Heating Mode Predictions

When the linear regression models are applied to the average OATs obtained from the Knoxville, TN TMY3 (typical meteorological year) data file, a representation of the performance of both systems during a typical heating season can be obtained. Figure 9 shows a running total of the predicted mean energy consumptions for both systems along with the daily average OAT as obtained from the TMY3 data file. The total energy use for the baseline system was 4754 ±116 kW·h compared to 6633 ±102 kW·h for the multi-split system with 95% confidence. This predicts that on average the multi-split system would use 1879 ±155 kW·h (Equation 4) or about 40% more energy during the heating season than the baeline system.

$$E_1 - E_2 \pm \sqrt{s_1^2 + s_2^2} \quad (4)$$

A comparison of the AHRI HSPF ratings would indicate that the multi-split system would use 12% more energy than the baseline system.

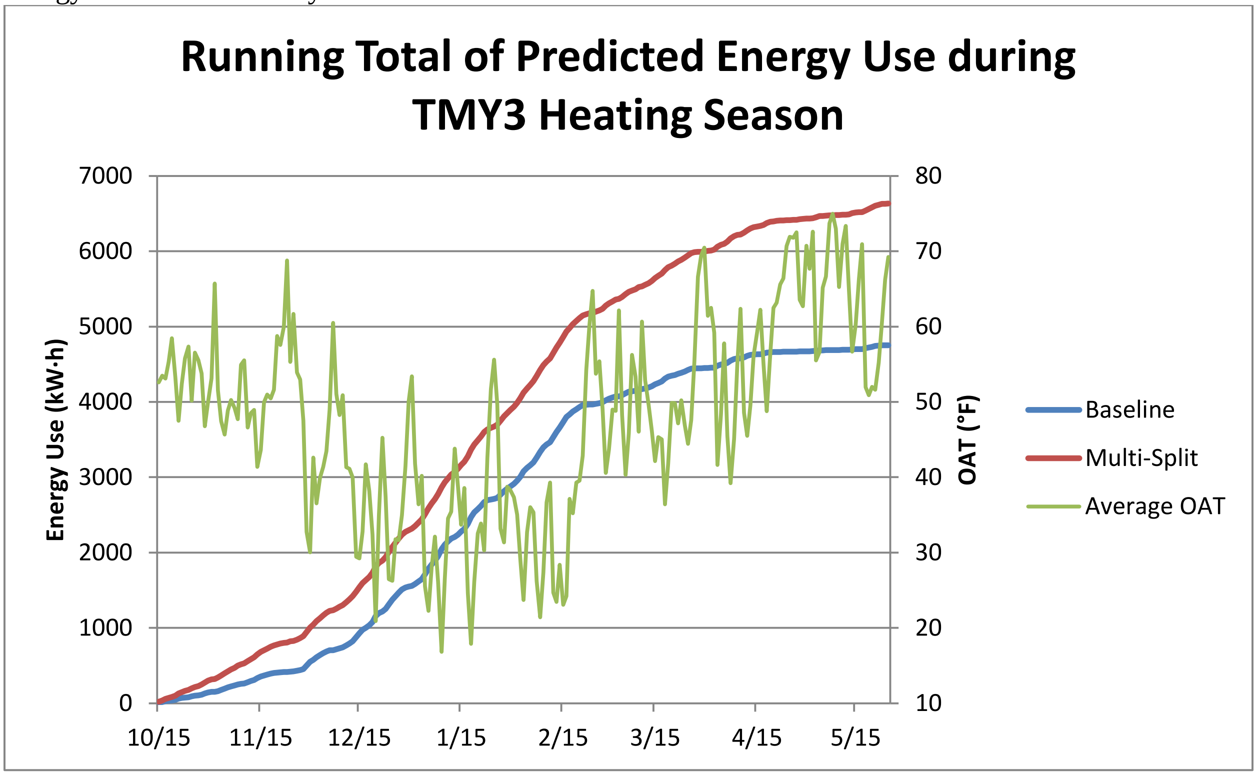

Figure 9: Running Total of Predicted Energy Use during TMY3 Heating Season

While the thermostats for all systems were set to maintain a constant 71°F, there are differences in controls and distributions in these systems. The most notable of these is that the baseline system was setup as a 2 zone system, while the multi-split system was essentially an 8 zone system. Figure 10 shows a histogram of the average daily indoor temperature of the entire house. As seen in the Temperature Control section, the multi-split system maintained a higher average temperature in the house due to its system receiving feedback from 8 locations as opposed to two. Therefore the multi-split system is being slightly penalized when comparing energy use due to this factor.

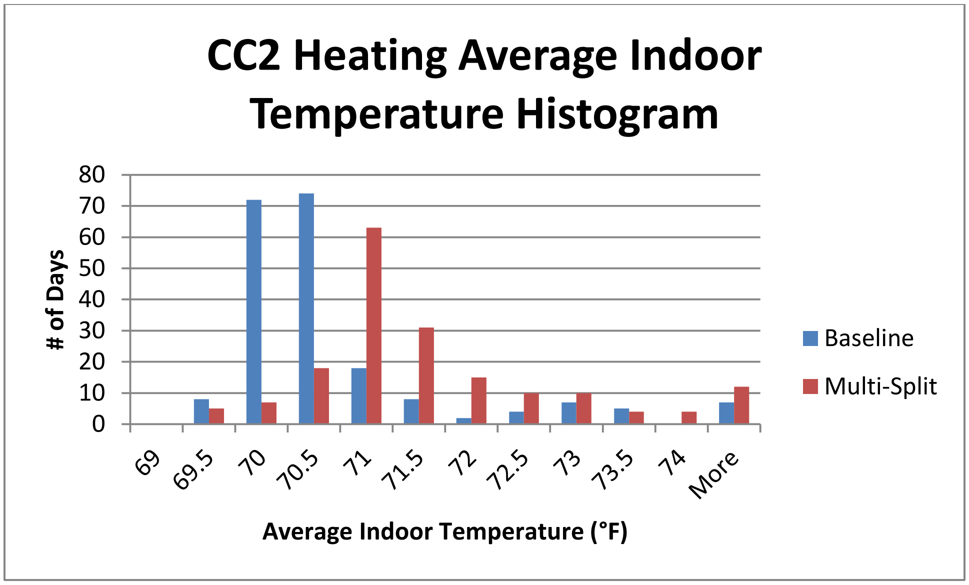

Figure 10: CC2 Heating Average Indoor Temperature Histogram

4.5 COOLING SEASON

During the cooling season, the continuous fan was re-enabled on the 3 units located downstairs. This was a compromise between unit performance and avoiding over-conditioning the upstairs rooms, since it allowed the fans in the downstairs units to provide some additional cooling, while ensuring that the upstairs rooms did not receive too much cooling. The humidity inside the home was noticeably higher than in the other test homes, which averaged between 45% and 51% RH during the cooling seasons. Figure 11 shows a histogram of the daily average indoor relative humidity levels during the cooling season for the multi-split and baseline systems. As seen in the figure, the average indoor relative humidity was roughly 5% higher when the multi-split system was running as opposed to the baseline system. There were also days when the average indoor relative humidity was over 60%.

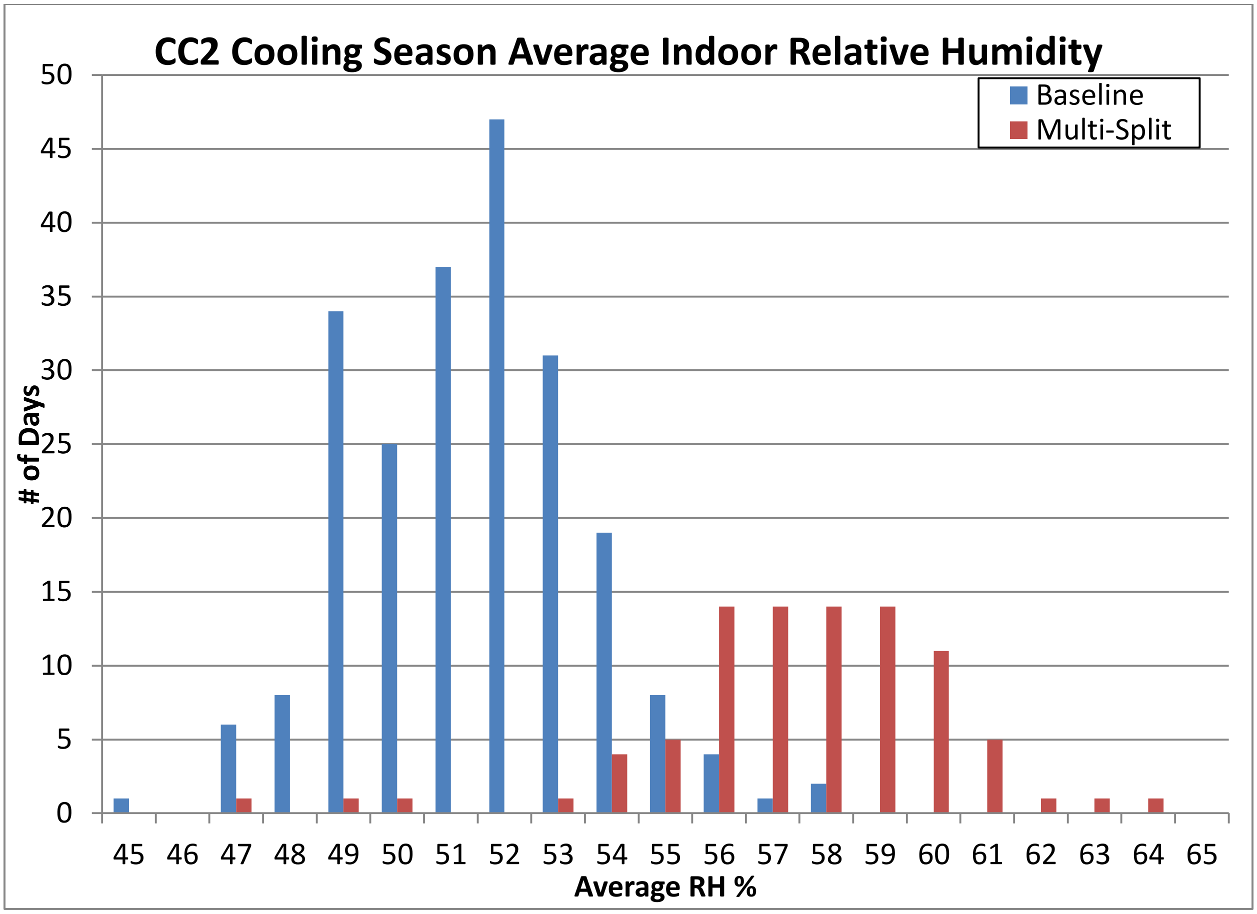

Figure 11: CC2 Cooling Season Average Indoor Relative Humidity

4.5.1 Temperature Control

Figure 12 shows the temperatures at the two thermostats as well as the power consumption for the baseline system during one of the hotter days (average OAT of 82°F). The temperature at the upstairs thermostat maintains a tight temperature band, but the temperature downstairs experiences some over-cooling. The unit cycled on and off frequently which is not unexpected from a single stage unit. However, given the fact that this was one of the hotter days, the cycling may indicate that the unit is slightly oversized for the cooling load.

Figure 13 shows the same information for the multi-split system during a day with a similar average OAT and similar insolation. As with the heating season, the temperatures in the upstairs hallway and the dining room are very consistent. The power use once again shows the unit running for a large portion of time at low power levels.

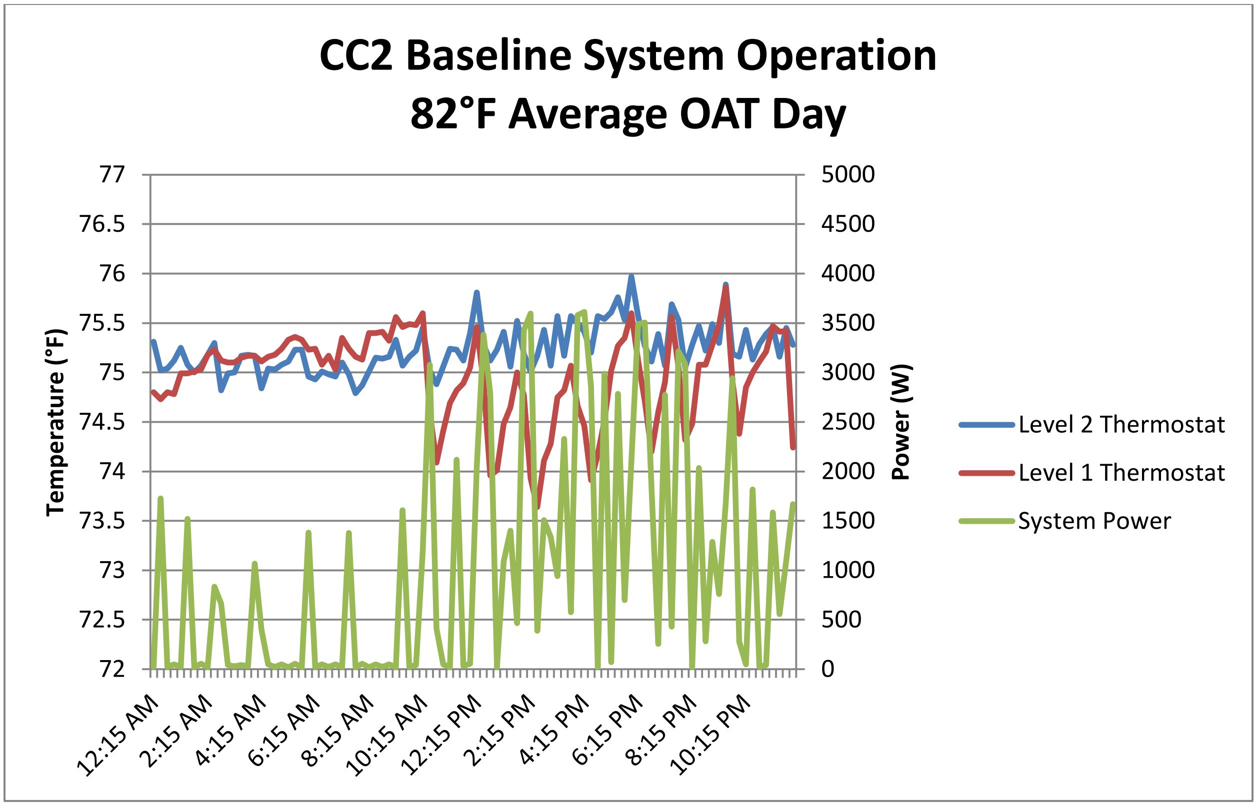

Figure 12: CC2 Baseline System Operation - 82°F Average OAT Day

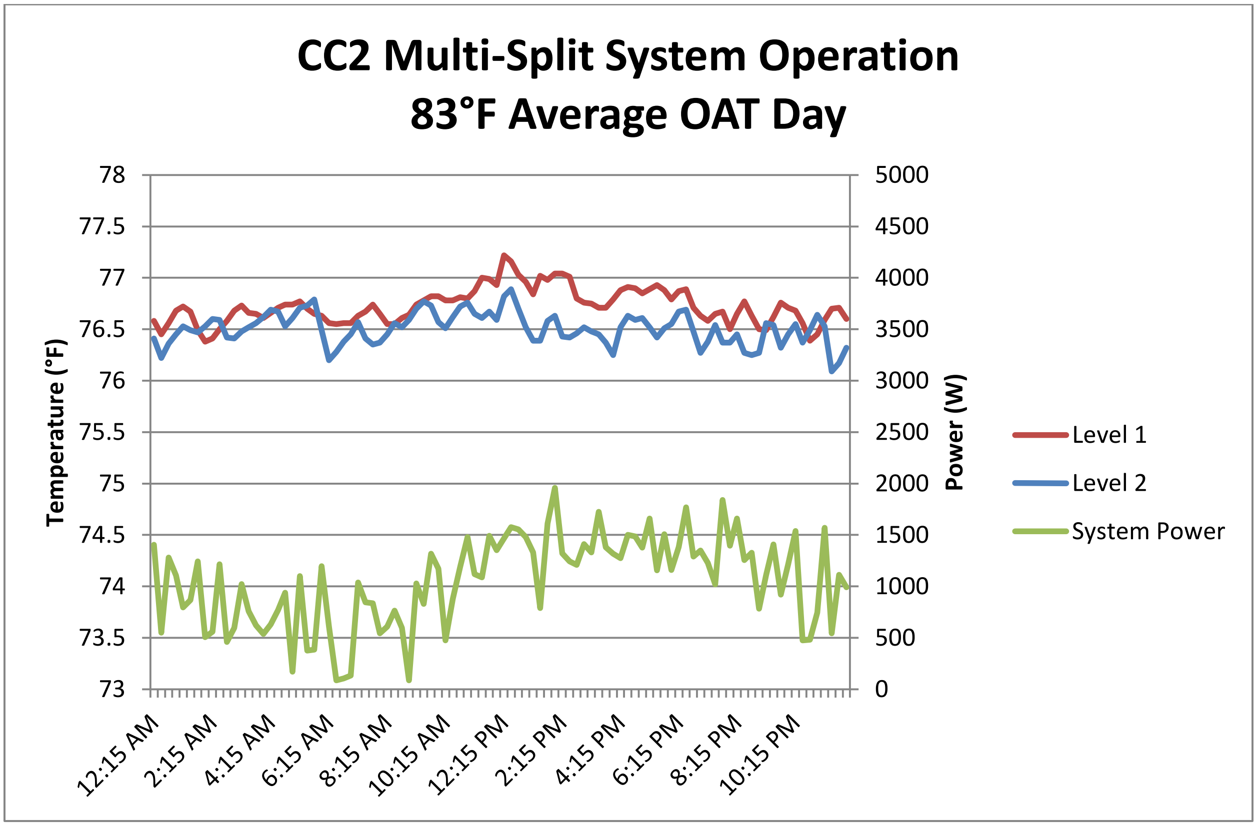

Figure 13: CC2 Multi-Split System Operation – 83°F Average OAT Day

Figures 14 and 15 show the mean radiant temperatures of the individual rooms for each system during the same day as in Figures 12 and 13 respectively. Once again the multi-split system maintains more consistent temperatures than the baseline system. Similar to the heating season, the baseline system maintains a cooler average temperature in the house. While this was an energy benefit in the heating season, it is an energy penalty in the cooling season.

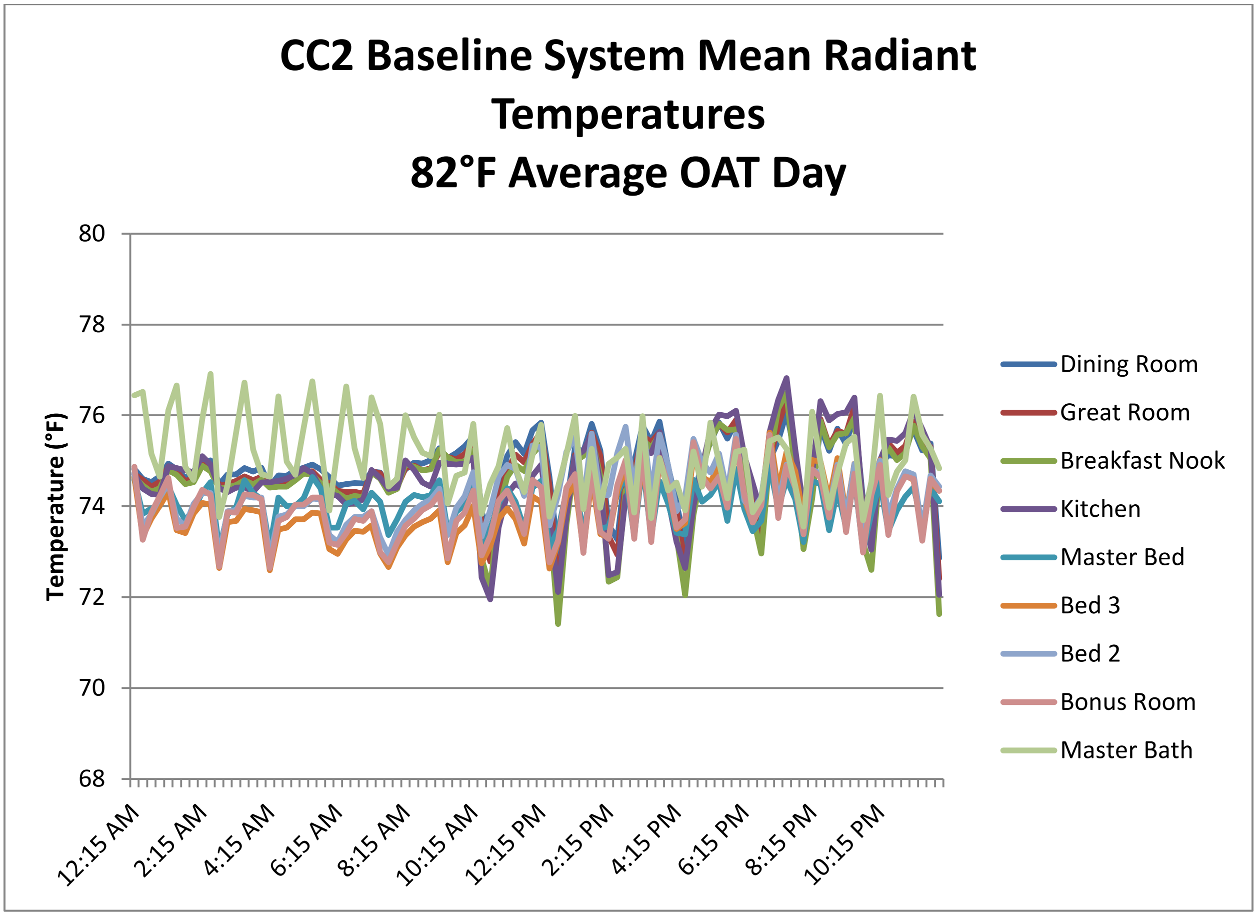

Figure 14: CC2 Baseline System Mean Radiant Temperatures – 82°F Average OAT Day

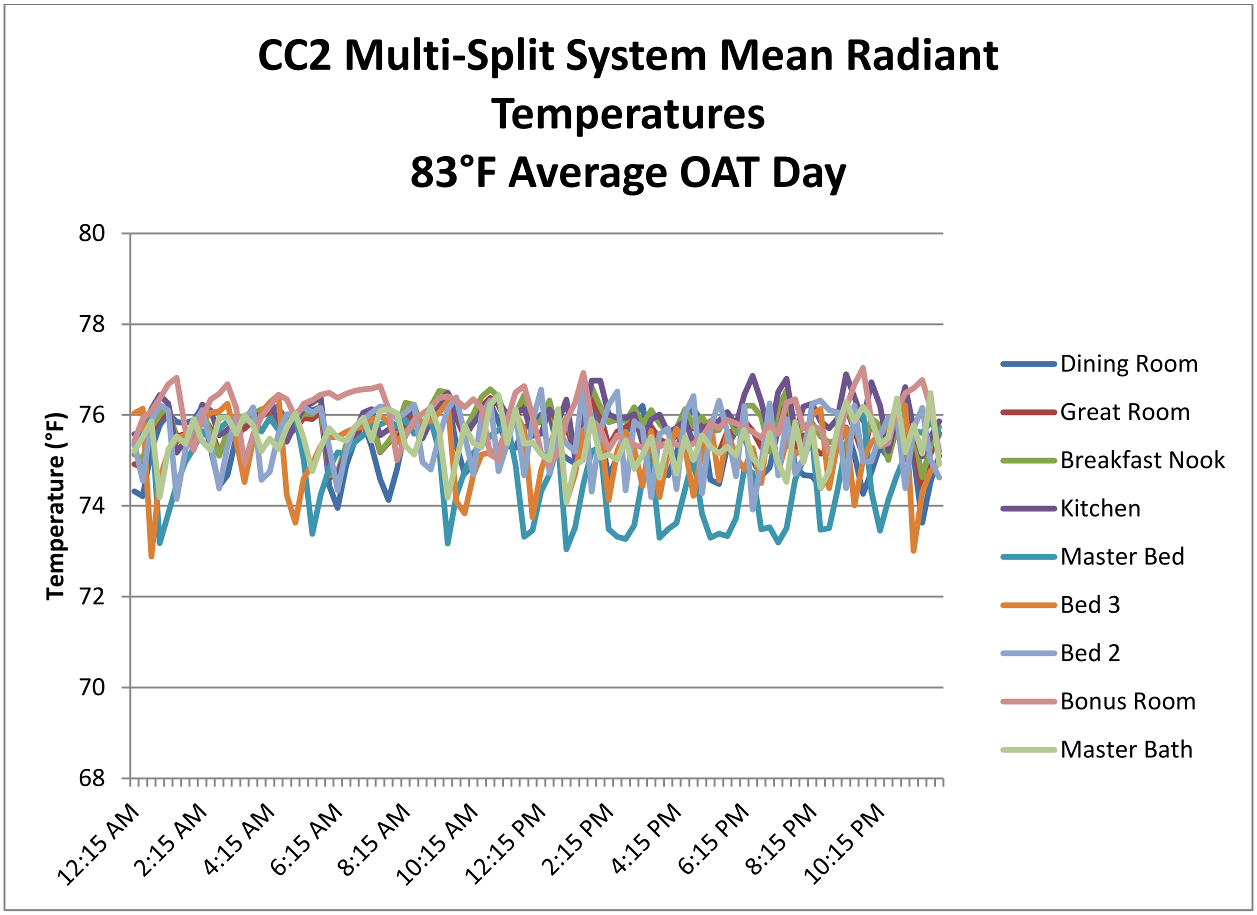

Figure 15: CC2 Multi-Split System Mean Radiant Temperatures – 83°F Average OAT Day

4.5.2 Linear Regression Models

Figure 16 shows the measured data and the resulting linear regression model for the baseline system during the cooling season.

During July and August of 2011, several weeks of thermostat setback testing took place. Since the temperature inside the house was no longer maintained at a constant temperature, these weeks were removed from the data set. Figure 17 shows the measured cooling season data and the resulting linear regression model.

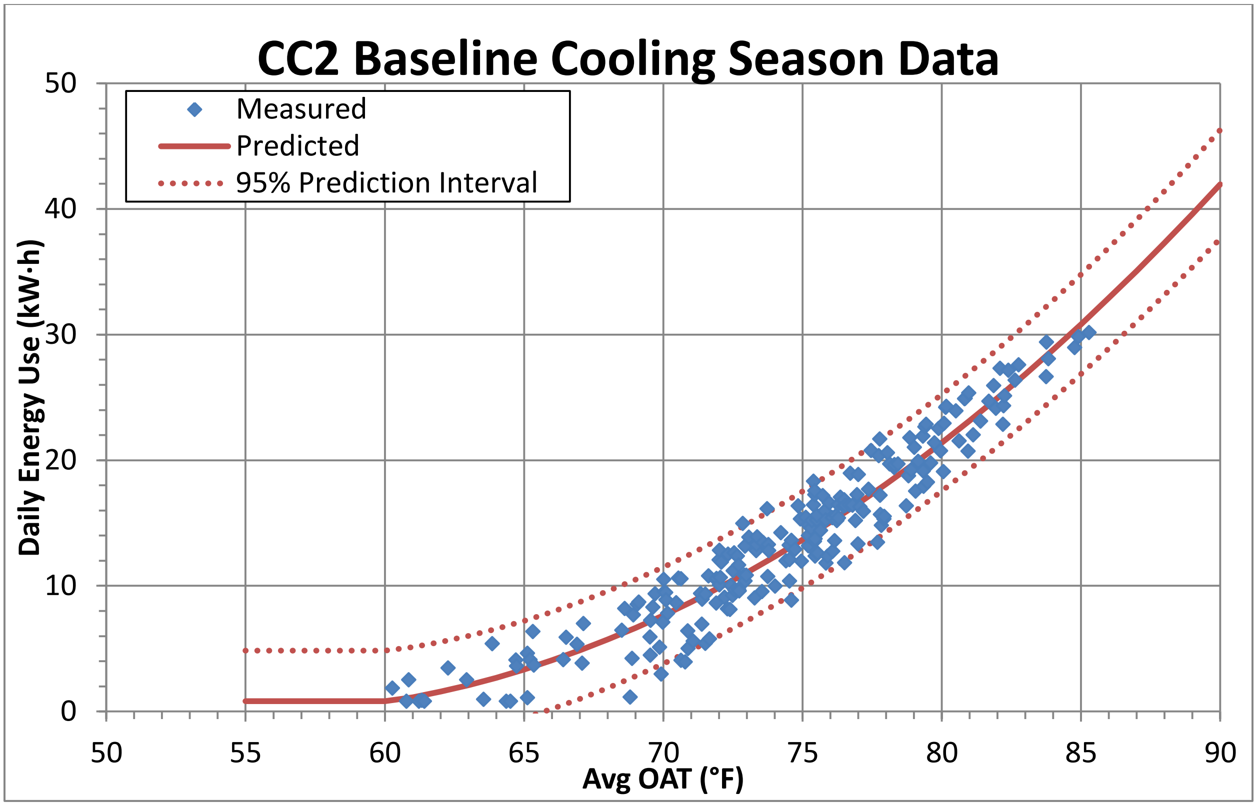

Figure 16: CC2 Baseline Cooling Season Data and Linear Regression Model

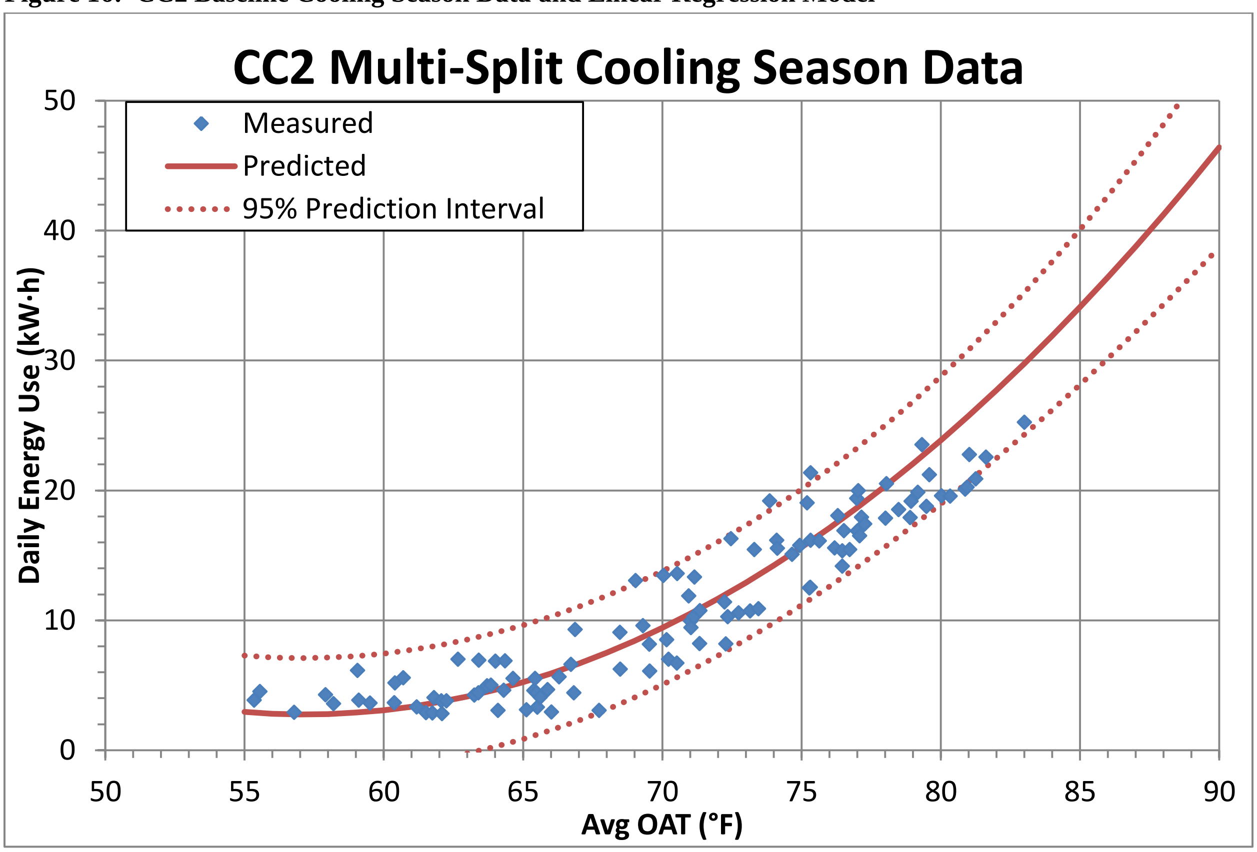

Figure 17: CC2 Multi-Split Cooling Season Data and Linear Regression Model

4.5.3 Performance Comparison

Figure 18 shows the linear regression models of both the baseline system and the multi-split system for the cooling season. As with the heating season data, the baseline system uses considerably less energy during periods of standby while not actively cooling. The baseline system is also predicted to perform slightly better for the entire cooling season temperature range.

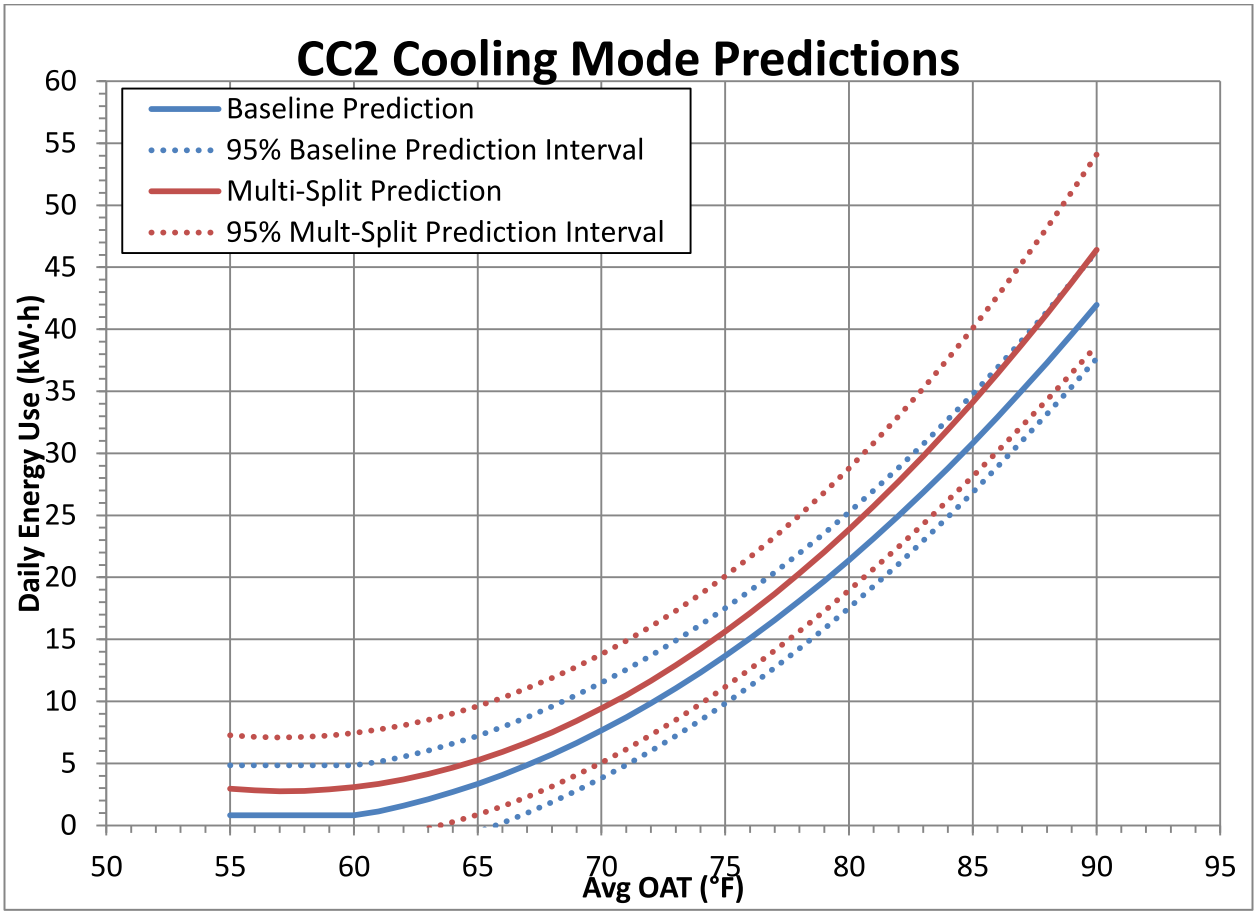

Figure 18: CC2 Cooling Mode Predictions

When the linear regression models for the baseline and multi-split systems are applied to the TMY3 cooling season data, the predicted energy use between the two systems can be determined. Figure 19 shows the running total of the energy use of both systems as well as the average OAT from the TMY3 data for Knoxville, TN. The baseline system was predicted to use 1907 ±45 kW·h, while the multi-split system was predicted to use 2204 ±55 kW·h with 95% confidence. This results in 298 ±71 kW·h, or ~16%, more energy use by the multi-split system. The AHRI SEER rating indicates that the multi-split system would use ~7% more energy than the baseline system during the cooling season.

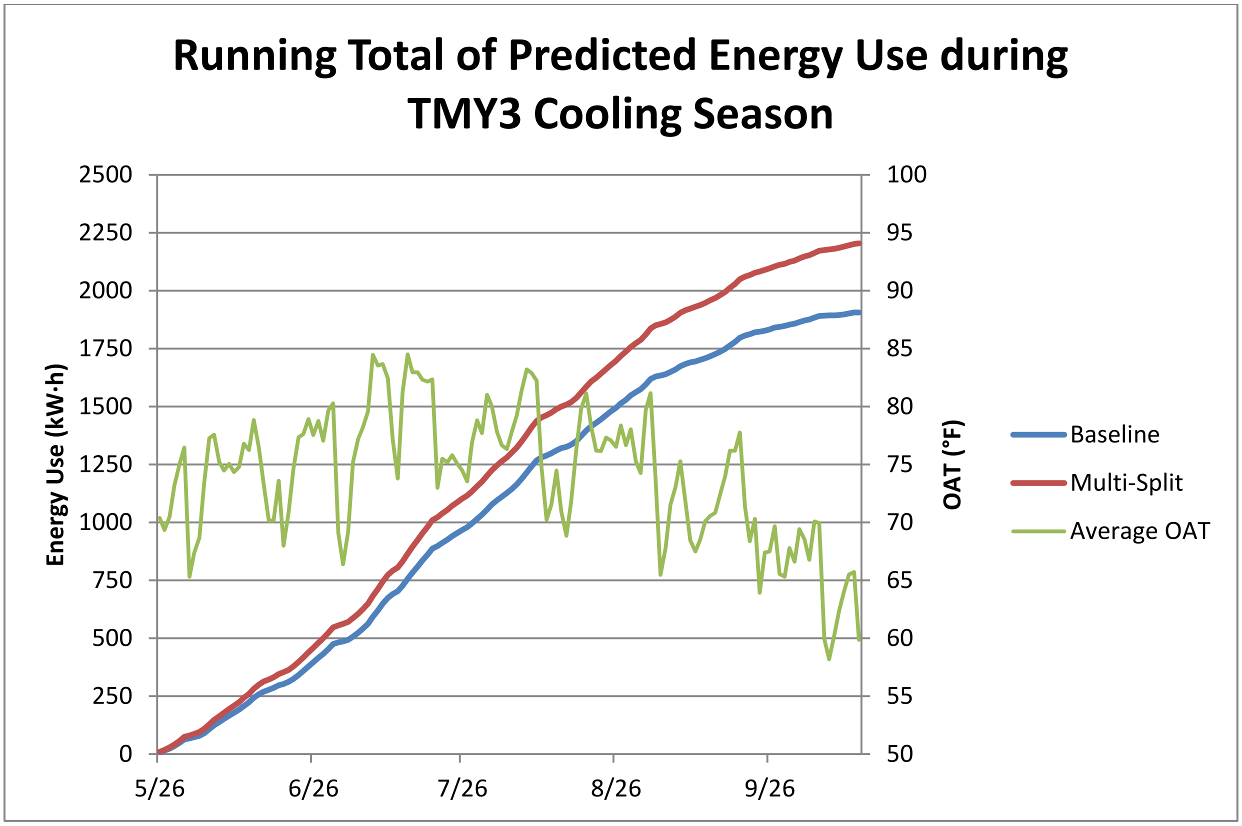

Figure 19: Running Total of Predicted Energy Use during TMY3 Cooling Season

As mentioned earlier, the relative humidity in the house was ~5% higher during the cooling season while the multi-split system was in service as opposed to the baseline system. Figure 20, also shows that the mutli-split system kept the inside air temperature of the house about 2°F warmer than the baseline system. This fact should reduce the load on the multi-split system when compared to the baseline system resulting in the linear regression model underestimating the energy use of the multi-split system.

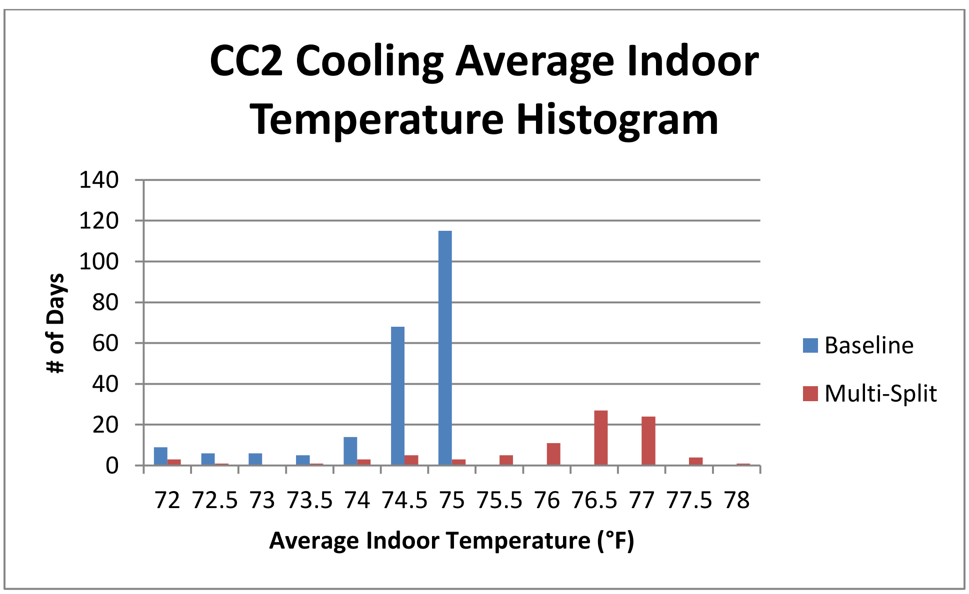

Figure 20: CC2 Cooling Average Indoor Temperature Histogram

4.6 ANNUAL PERFORMANCE

When both the heating and cooling season TMY3 predictions are combined, the baseline system is predicted to use 6660 ±125 kW·h and the multi-split system is predicted to use 8837 ±116 kW·h with 95% confidence. This results in an annual energy savings of 2177 ±171 kW·h for the baseline system.

4.7 DISCUSSION

The multi-split system was able to maintain the indoor air temperature setpoint of 71°F during the heating season without the use of auxiliary resistance heat. This potentially reduces the peak loads experienced by electric utilities in the winter, however the multi-split system used more energy in this study throughout the entire heating season than the baseline system. It did however provide a tighter temperature control. This would likely translate into better comfort year round for the homeowner, however the higher humidity in the summer would detract from this benefit. A major concern with the multi-split system is the amount of energy that the 8 indoor units consumed while the system was not actively heating or cooling. This is of particular concern for well insulated and sealed homes that will experience long periods of time without a need for heating or cooling. This could be addressed if the units were, when feasible, turned off when a room was unoccupied.

The commercial application controller installed with the multi-split system provided many issues with the installation and commissioning of the system. The manufacturer has reported that they are currently working on a central controller suitable for residential applications for their system, which should resolve the issues we experienced. The multi-split system still offers more zoning flexibility than a traditional system, which, if used properly, could provide a larger savings than traditionally zoned systems.

In this study, the multi-split system did not provide energy savings over a single stage 16 SEER HP. Due to the issues we experienced with over-conditioning in some rooms, it seems apparent that sizing the indoor units appropriately is still a major concern for this multi-split system despite its variable speed compressor.

5. CC3 DUCTED INVERTER SYSTEM

5.1 BASELINE SYSTEM

The baseline system in CC3 was a two stage, 16 SEER, 9.5 HSPF HP with an ECM blower motor that was connected to a zoning system with the upstairs and downstairs serving as the two zones. Due to an installation error with the zoning control board, the unit was not able to go into high stage for the entire heating season and part of the cooling season. The unit was able to meet the cooling load in the summer, but in the winter the use of auxiliary heat was required. It is not known how much of this could have been avoided if the unit was able to operate in the high stage.

5.2 INSTALLATION

Installation of the ducted inverter system began in November of 2010. The installation was relatively straight forward. The system was originally paired with 5 kW of auxiliary heat, however the unit would not allow the resistance heat to run simultaneously with the HP. When it got colder, this resulted in the system switching back and forth between HP heating and resistance heating. A service technician came out to check the unit, which involved connecting a service checker interface to collect data as the unit ran. This data indicated that the unit was low on charge, so an additional 1.5lbs of refrigerant was added. The 5 kW resistance heater was also replaced with a 3 kW unit, that could be run simultaneously with the HP as supplemental heat.

5.3 CONTROLS

Unlike the baseline system, the ducted inverter unit had no options available for zoning the system. As a result of this, the ducted inverter unit had to be run with a single thermostat located on the 1st floor. Throughout the heating and cooling season adjustments were made to the supply registers in an attempt to balance the system so that both floors would be maintained at a constant temperature. This was a challenging task and only achieved limited success as will be noted in the Performance Comparison sections.

5.4 HEATING SEASON

5.4.1 Temperature Control

The baseline system was zoned with one thermostat located between the kitchen and the dining room on the first floor, and the other located in the upstairs hallway. The temperature was recorded near the thermostats in order to observe how they were operating. Extreme cold days of similar average OAT were used in order to compare the operation and cycling of the HPs.

Figure 17 shows the indoor air temperatures at the thermostat locations for the baseline system during a day with an average OAT of 23.0°F. The temperatures are very constant, except for during the daytime hours when the home is being heated by solar radiation through the windows and in the late evening as the HP is no longer able to keep up with the load. It is clear that when the unit is not running a defrost cycle, that the system is using around 1300 W, which based on the manufacturers data, is indicative of the unit operating in the low stage. When the indoor temperature drops to ~68.5°F, the auxiliary heat is brought on to supplement the HP capacity, but the compressor power still indicates low stage operation. The frequent spikes on the HP power trace are times when the unit is going into the defrost mode and the auxiliary heat is energized. These are occurring at approximately 30 minute intervals and lasting around 4 minutes each time. This is the shortest defrost interval available on the unit and is most likely excessive and detrimental to the overall system performance.

The ducted inverter system did not have any zoning options available to use with our system, so the thermostat was placed on the first floor on the dining room wall, a few feet from the baseline system's thermostat. Initially the upstairs was under-conditioned and the temperature dropped below the set point, likely due to undersized ductwork in comparison to the downstairs ductwork and the lack of zoning. In order to compensate for this, the registers for the downstairs vents were significantly closed in order to force more conditioned air upstairs. Figure 18 shows the plot of the temperature near the thermostat as well as the temperature in the upstairs hallway where the baseline unit's thermostat was located during a day with an average OAT of 24.3°F and similar insolation. The thermostat for the ducted inverter system appears to be calling for heat when the temperature drops just below 71°F, and turning off the heat after the temperature is about 2.5°F above the set point. This is a fairly large overshoot, particularly for a system with a variable speed compressor. Figure 19 shows a single heating cycle with 1 minute resolution from the same day. It takes about 10 minutes for the unit to reach full power, and the unit maintains this level of power use even as the indoor air temperature at the thermostat increases up to 73.5°F. This indicates that the ducted inverter system is not controlling the compressor speed based on feedback from the thermostat, and could indicate that the unit is not operating properly. This issue is currently under investigation and the results will be documented in a future report.

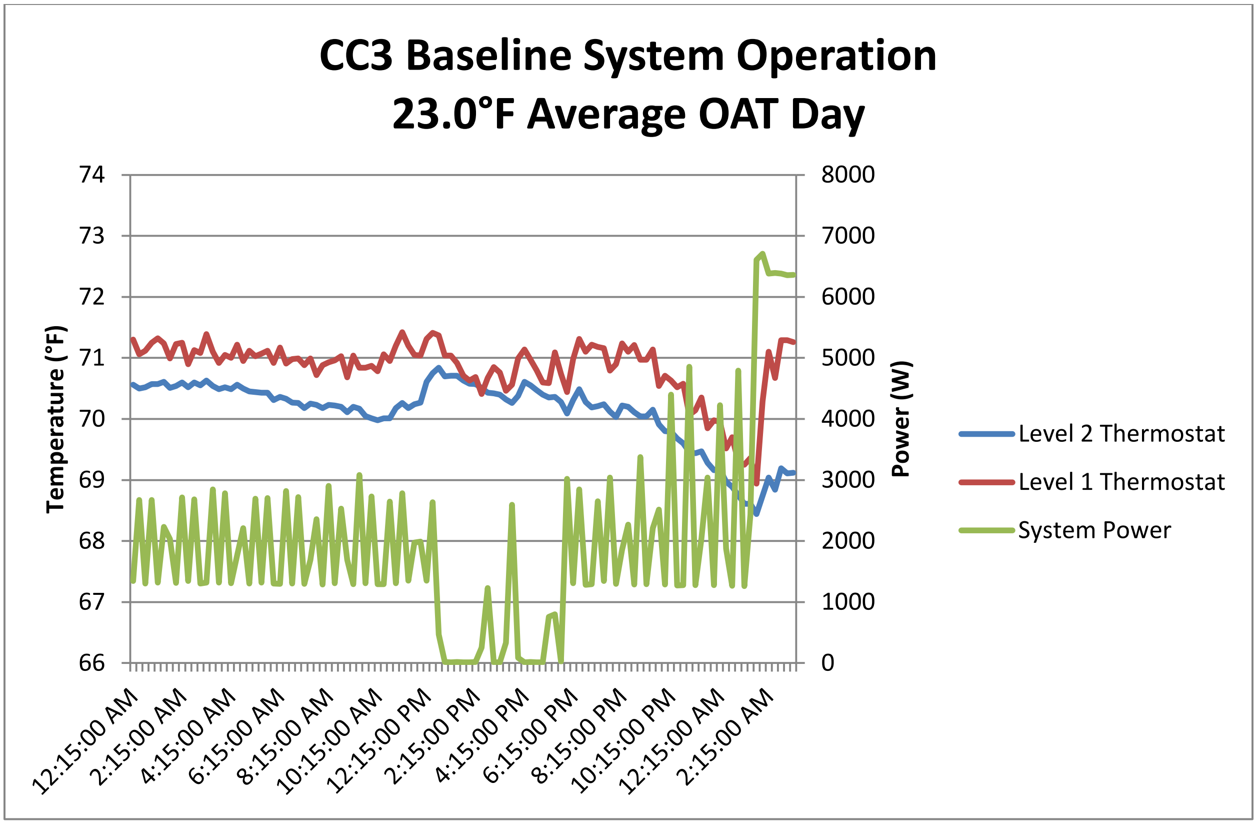

Figure 21: CC3 Baseline System Operation - 23.0°F Average OAT Day

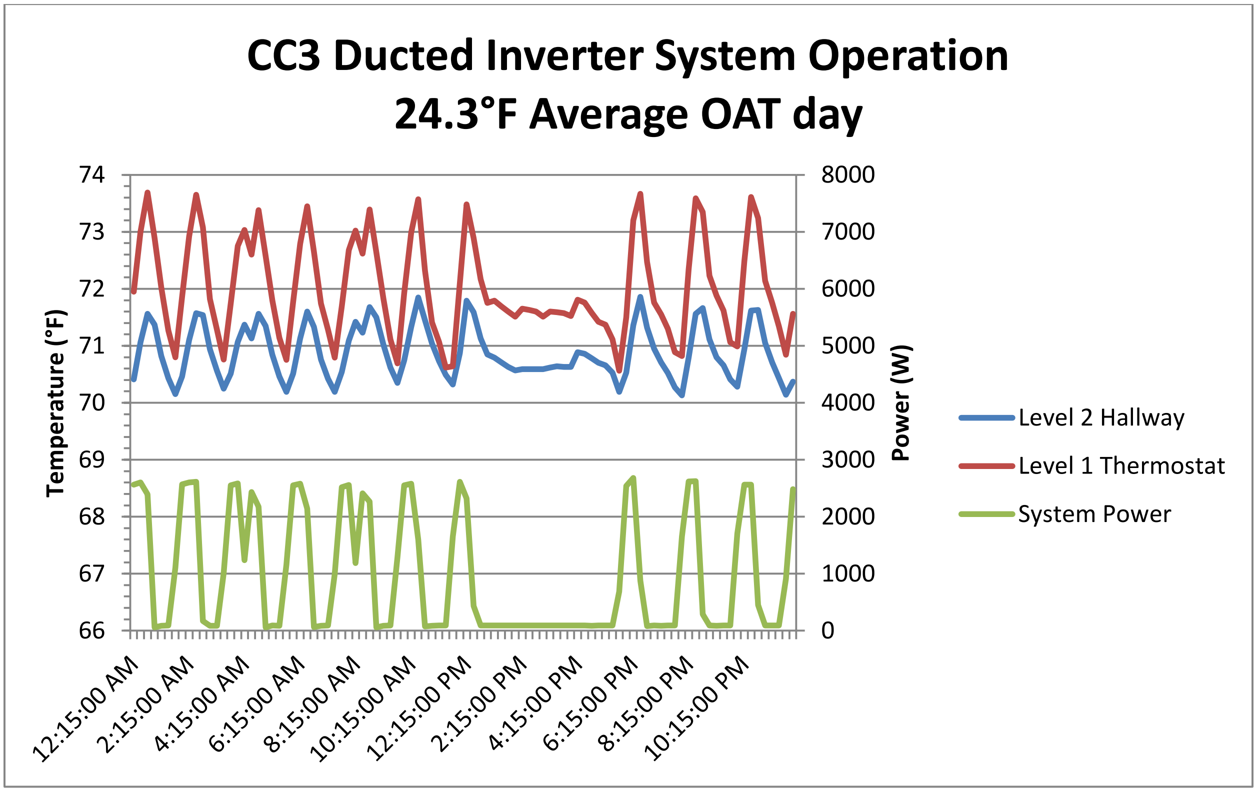

Figure 22: CC3 Ducted Inverter System Operation - 24.3°F Average OAT Day

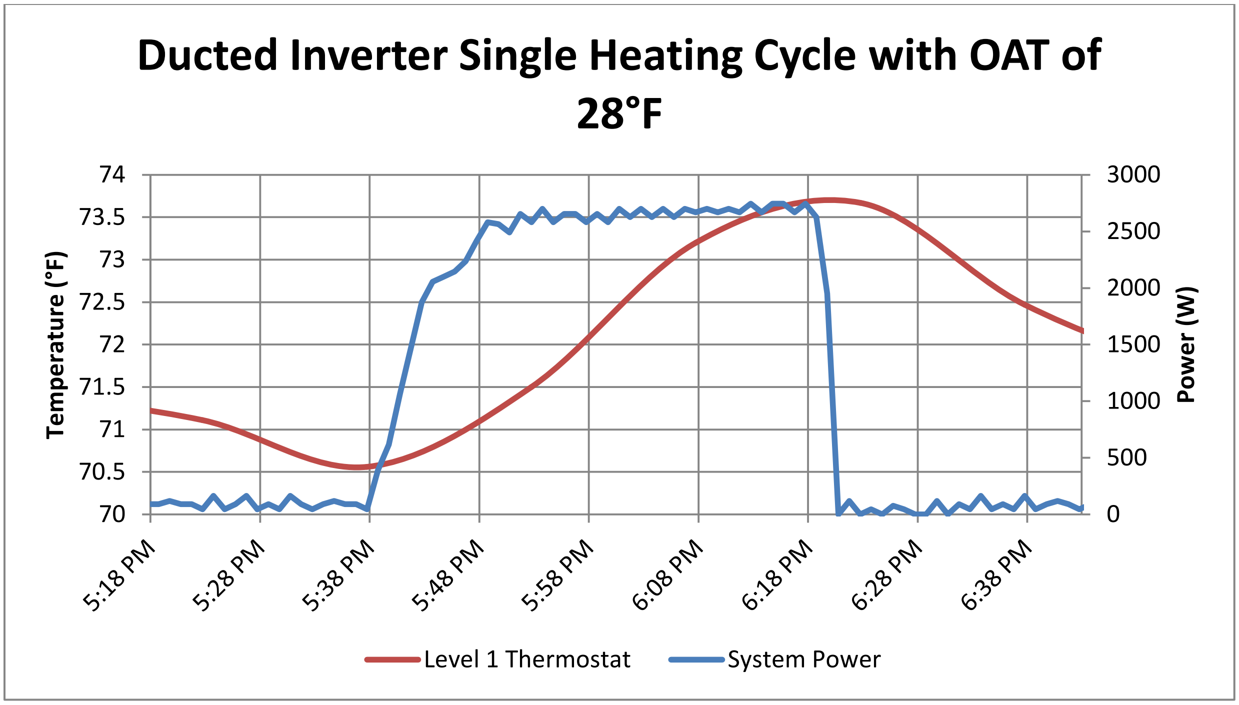

Figure 23: Ducted Inverter Single Heating Cycle with OAT of 28°F

While both the baseline system and the ducted inverter system at CC3 utilized the same ducts for distributing conditioned air, the baseline system was able to control dampers on the main supply lines to the upstairs and downstairs via its zone control system. The ducted inverter system required the individual register vents to be manually adjusted in order to try and balance out the temperature of the upstairs and downstairs. Figures 20 and 21 show plots of the mean radiant temperatures in each of the rooms of the house during a very cold day. The average mean radiant temperature of the rooms over the entire day was about 0.4°F higher while the ducted inverter system was operating. However, the ducted inverter system has very large, ~2.5°F, temperature swings each time the unit cycles.

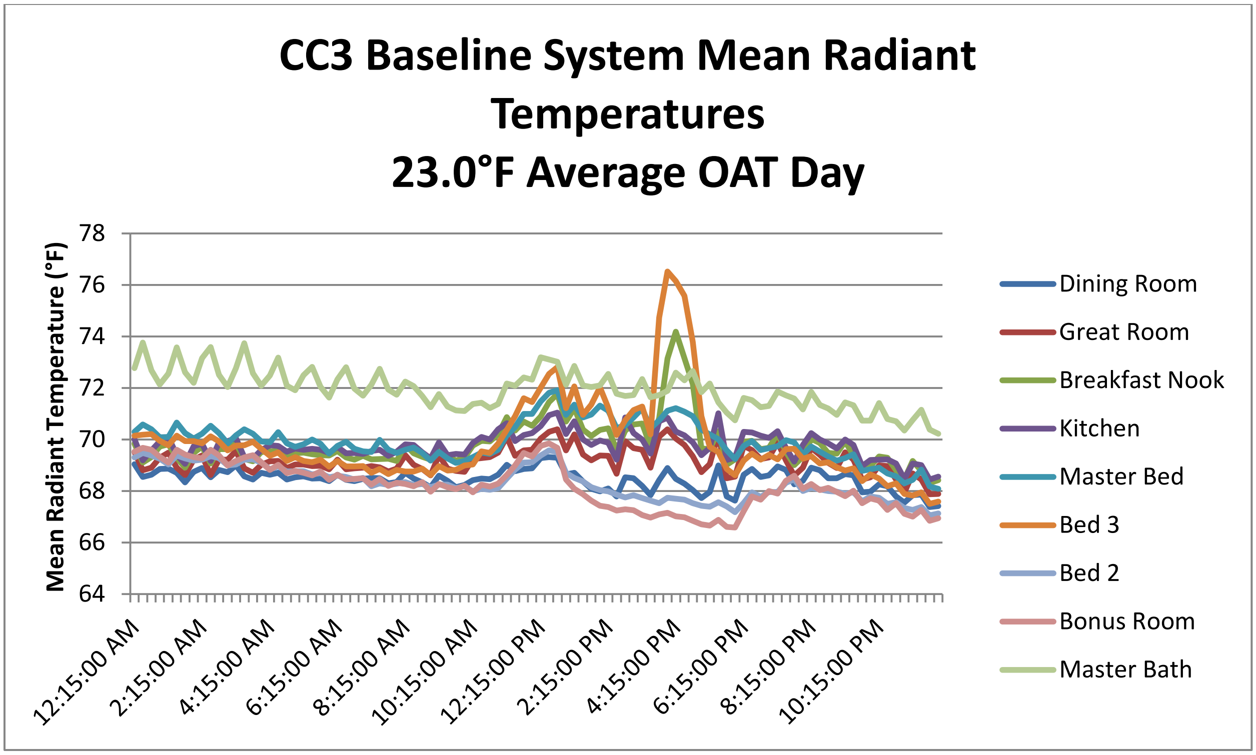

Figure 24: CC3 Baseline System Mean Radiant Temperatures - 23.0°F Average OAT Day

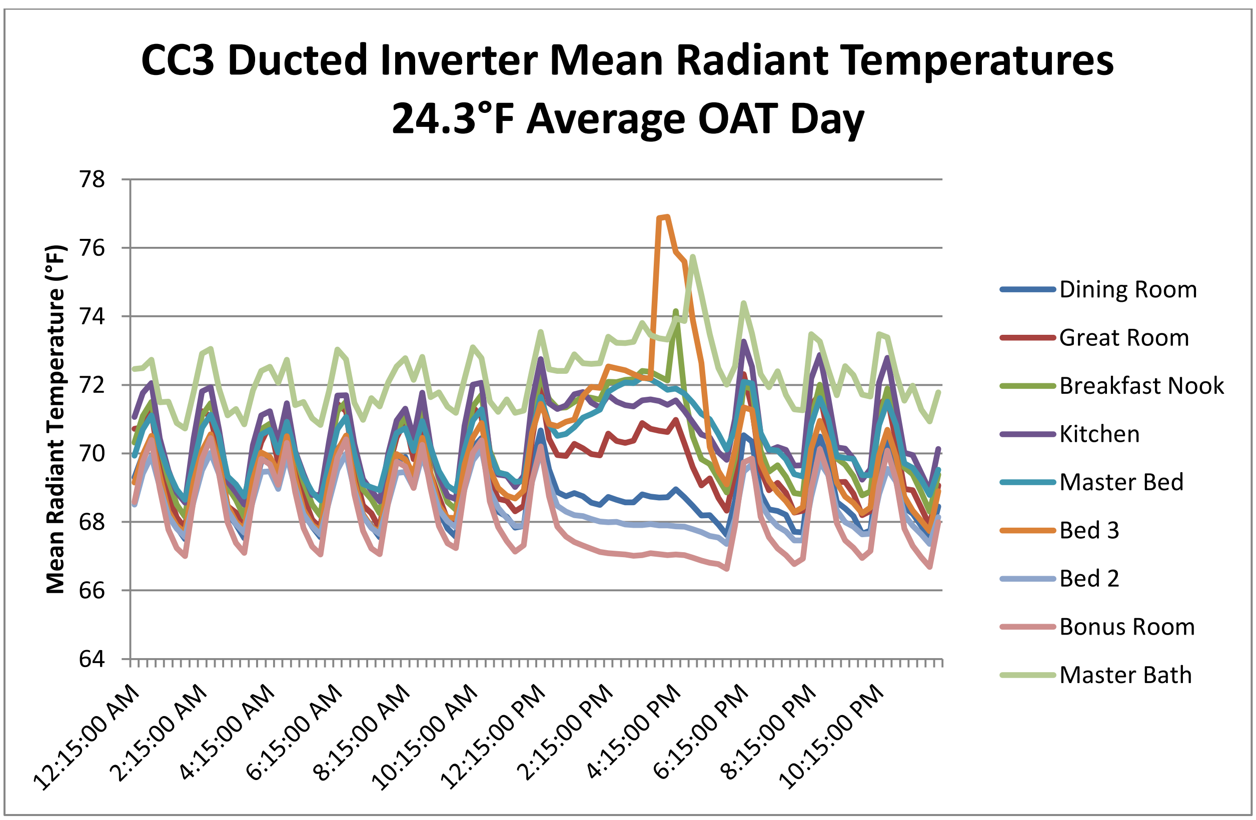

Figure 25: CC3 Ducted Inverter Mean Radiant Temperatures - 24.3°F OAT Day

5.4.2 Linear Regression Models

Figure 22 shows the measured energy use data and the linear regression model with 95% confidence prediction intervals for the baseline system during the heating season. The ability of the building envelope at CC3 to retain heat is readily apparent by how low the average outdoor air temperature was before heating was required.

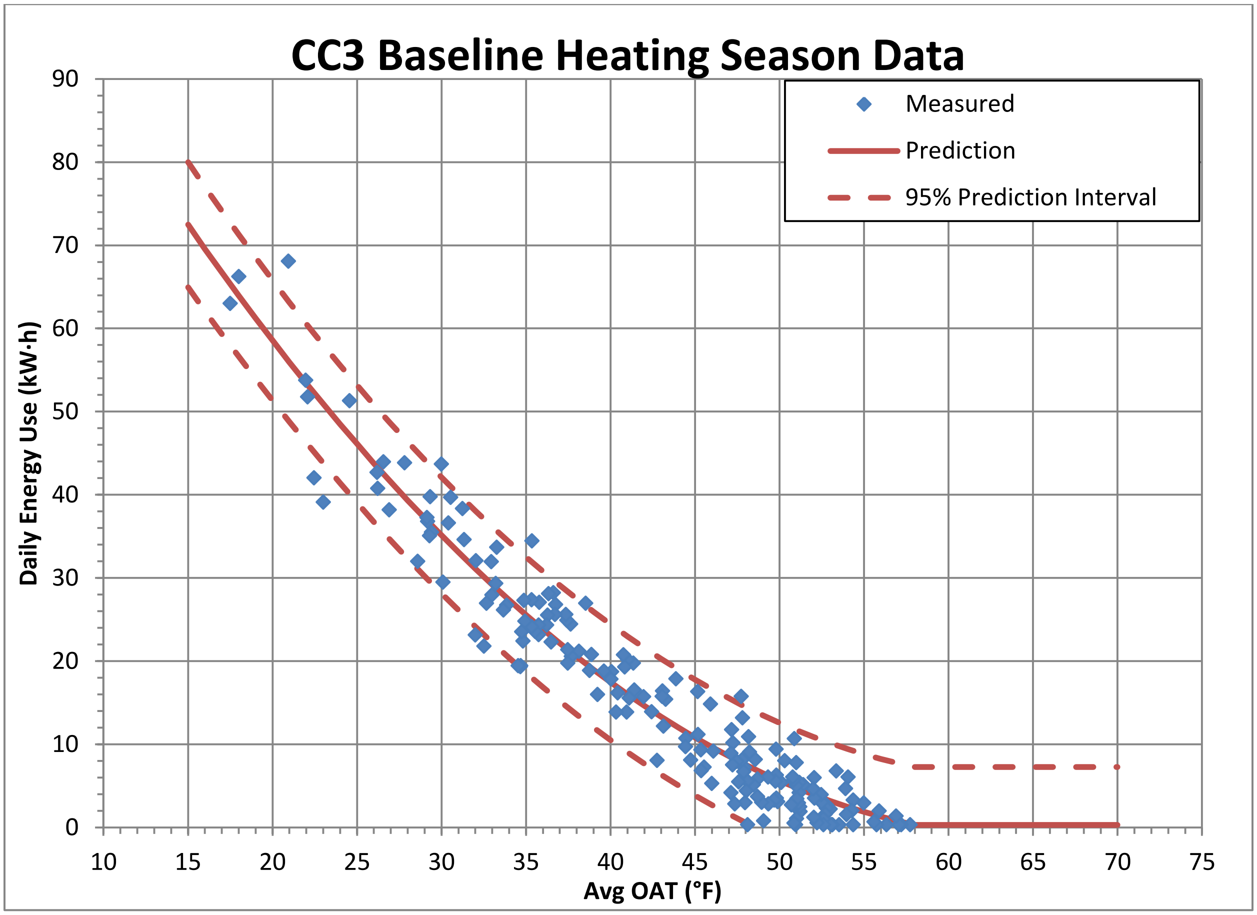

Figure 26: CC3 Baseline Heating Season Data

Figure 23 shows the measured energy use and linear regression model for the ducted inverter system during the heating season. Additional charge was added to the unit during January of 2011, so data points with large residuals before this date were removed.

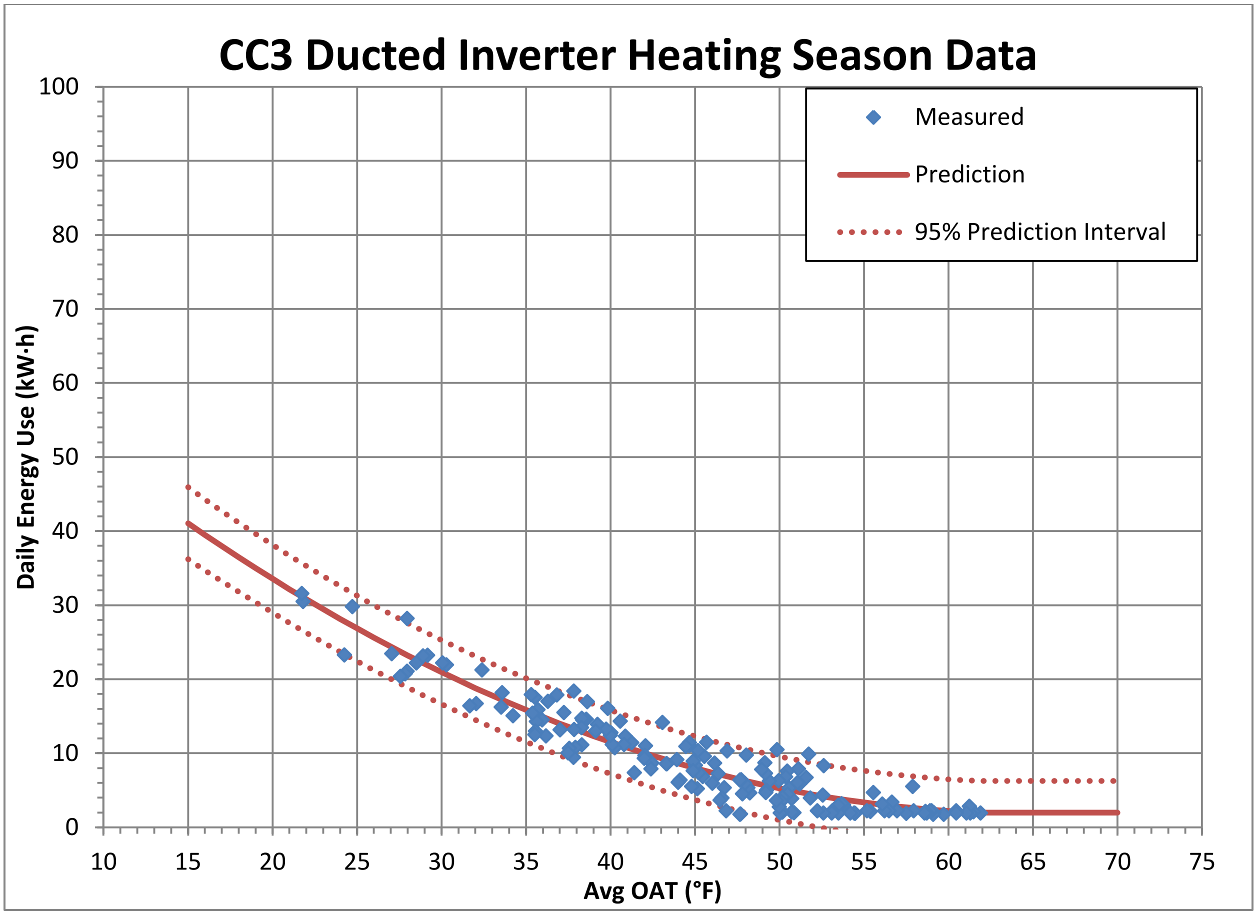

Figure 27: CC3 Ducted Inverter Heating Season Data

5.4.3 Performance Comparison

Figure 24 shows the linear regression models for both the baseline system and the ducted inverter system plotted against the daily average OAT. The ducted inverter system's performance is substantially better at average OATs less than 51°F. Similar to the multi-split system, the ducted inverter system consumed considerably more power while in standby compared to the baseline system. The standby power consumption of the ducted inverter was ~80 W, while the baseline unit consumed ~13 W. Due to the issue of the baseline system not being able to run in the high stage though, the performance difference of these systems in the heating mode must be discounted some.

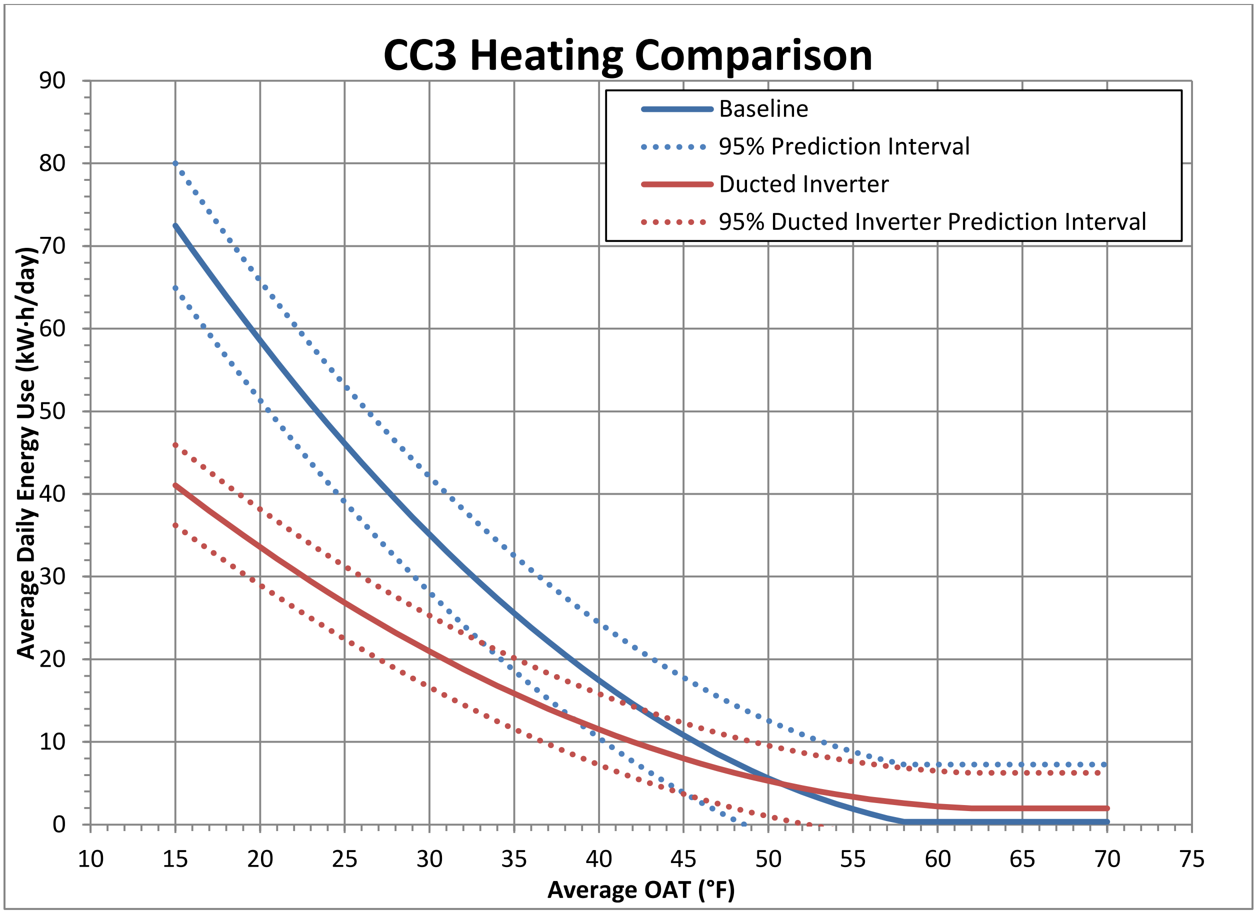

Figure 28: CC3 Heating Comparison

Applying the linear regression models to the TMY3 heating season data for Knoxville, TN predicts that the baseline unit would use 2885 ±104 kW·h and the ducted inverter system would use 2025 ±64 kW·h with 95% confidence. The running total of the two systems' energy use is plotted in Figure 25 along with the average OAT from the TMY3 data file. This results in the ducted inverter system being predicted to use 860 ±122 kW·h, ~30%, less energy than the baseline system. The AHRI HSPF ratings indicate that the ducted inverter system would use ~7% more energy than the baseline system.

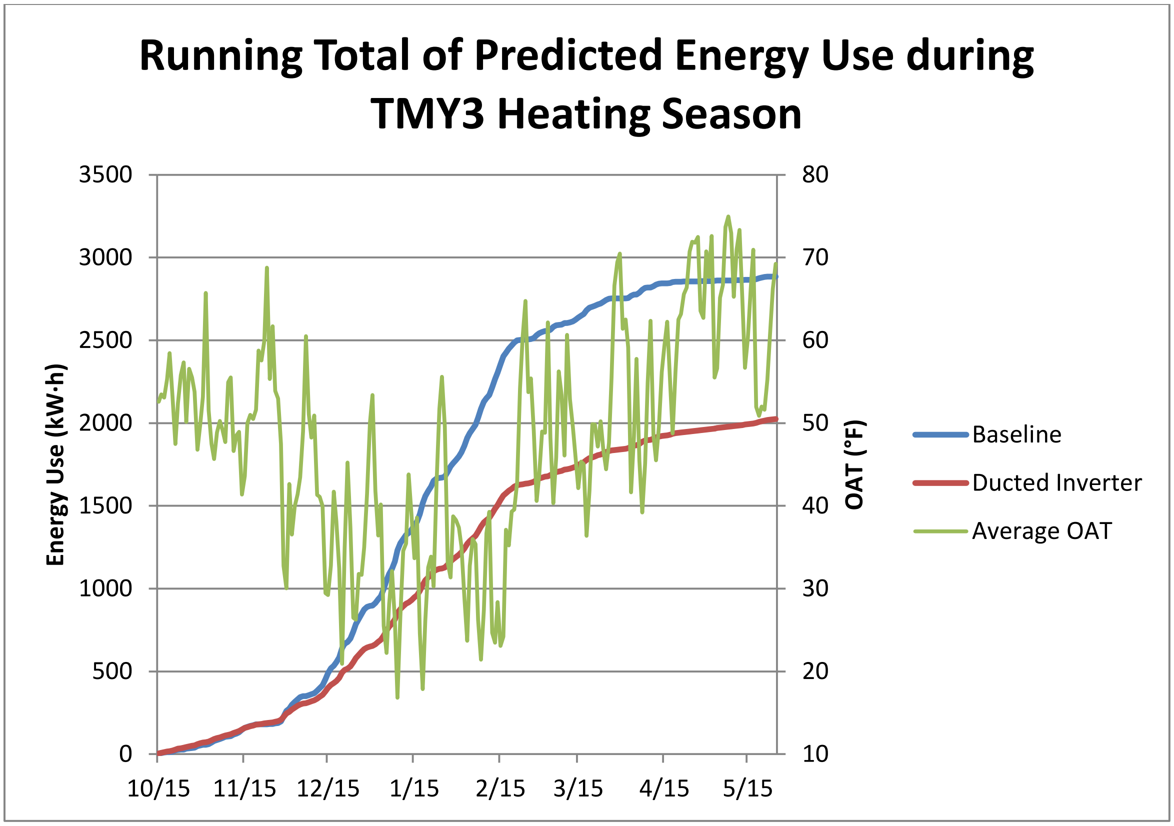

Figure 29: Running Total of Predicted Energy Use during TMY3 Heating Season

5.5 COOLING SEASON

While the baseline system was not able to operate in the high stage for a portion of the cooling season data, this was corrected for the latter half. Even without the ability to operate in high stage the unit was capable of maintaining the set point with just low stage operation.

5.5.1 Temperature Control

Figure 26 shows the two temperatures at the zone thermostats for the baseline system, as well as the total power use of the system on a hot day. The temperature variation is small, which is expected of a zoned system. Figure 27 shows the same measurements for the ducted inverter system on a day with a similar average OAT and insolation. As with the heating mode, there are large swings in the level 1 thermostat temperature, which are uncharacteristic of a variable capacity unit. The ducted inverter system also exhibits shorter and more frequent cycles, which is also unexpected in a variable capacity HP.

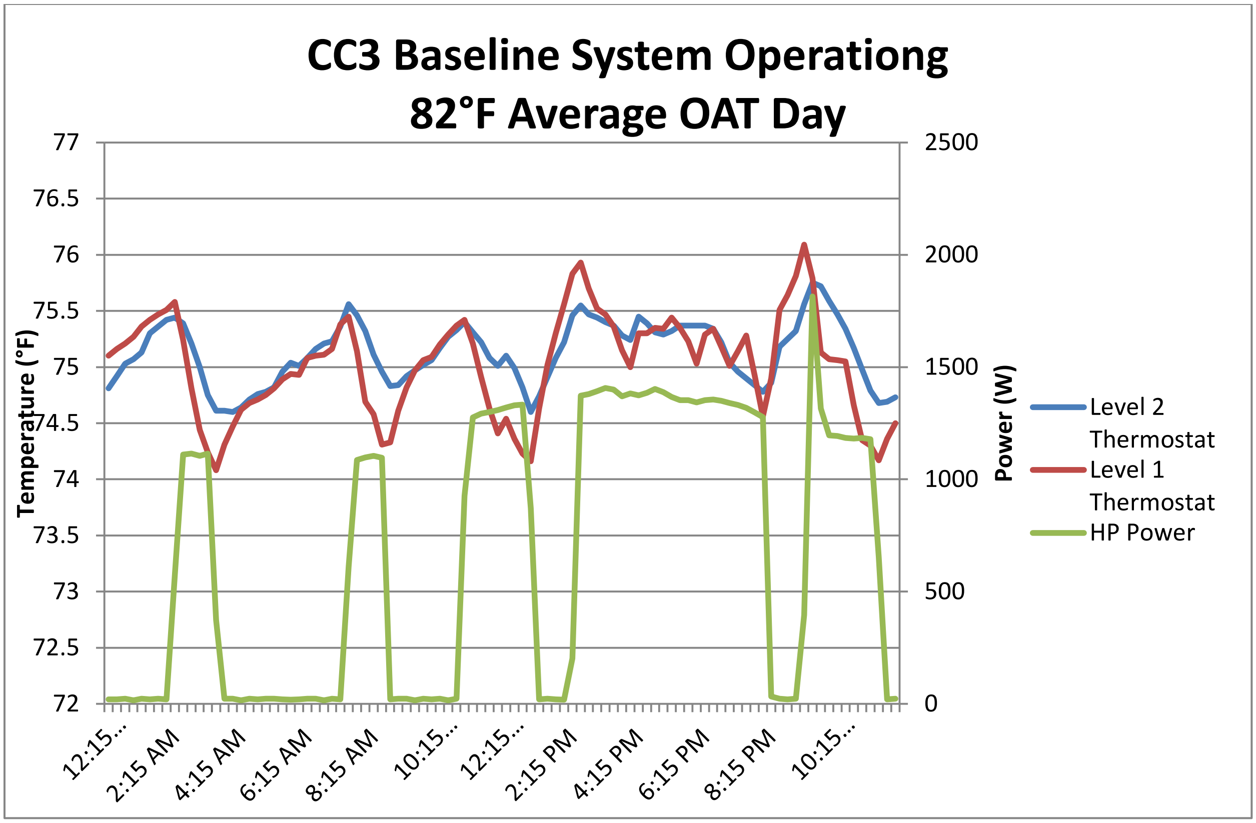

Figure 30: CC3 Baseline System Operating - 82°F Average OAT Day

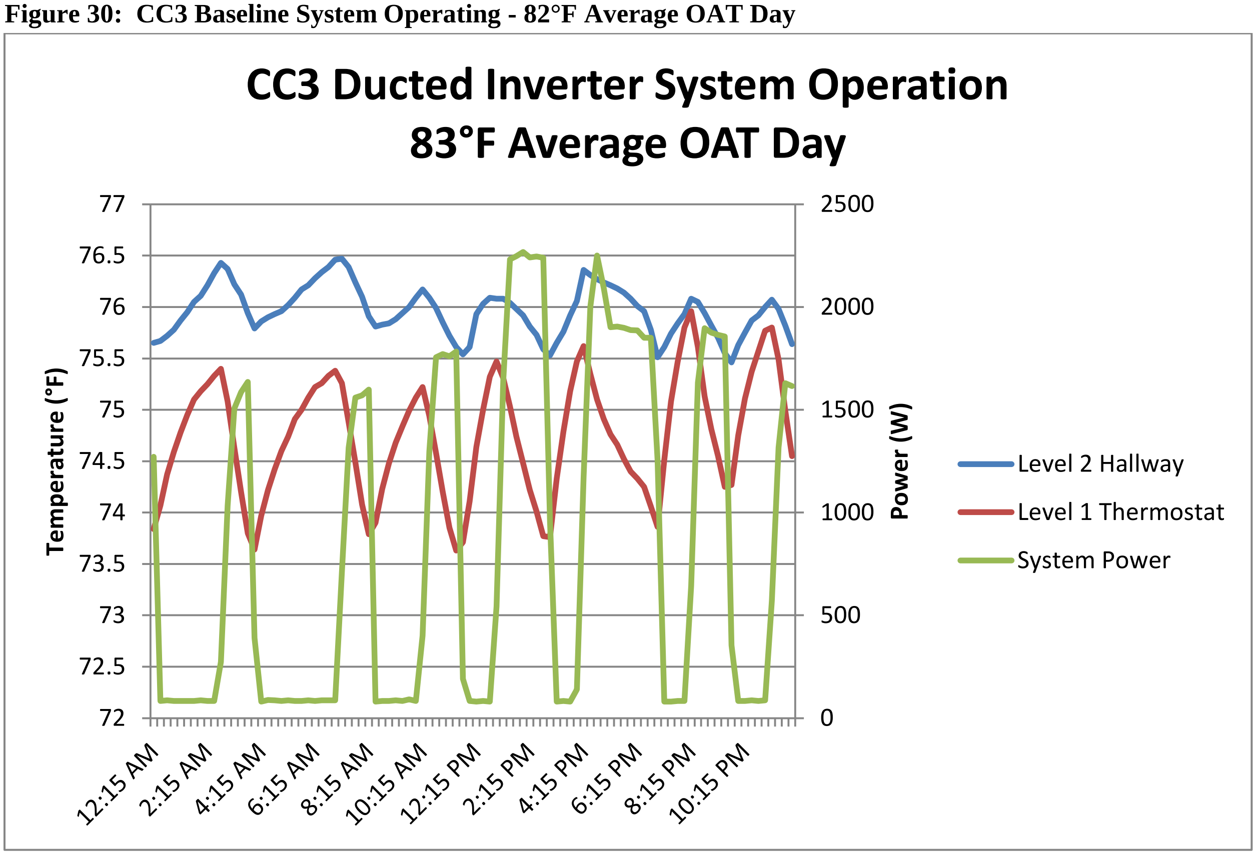

Figure 31: CC3 Ducted Inverter system Operation – 83°F Average OAT Day

Figures 28 and 29 show the mean radiant temperatures in the individual rooms for the same days as in Figures 26 and 27. Both units display reasonably large temperature swings in the rooms. While the two thermostat temperatures for the baseline system tracked very closely to each other, the individual room temperatures were not as close. This is likely due to a duct system that was not properly balanced for the cooling season. The baseline system maintained a slightly cooler mean radiant temperature averaged over all of the rooms of 73.9°F compared to 74.5°F for the ducted inverter system during their respective "hot" days.

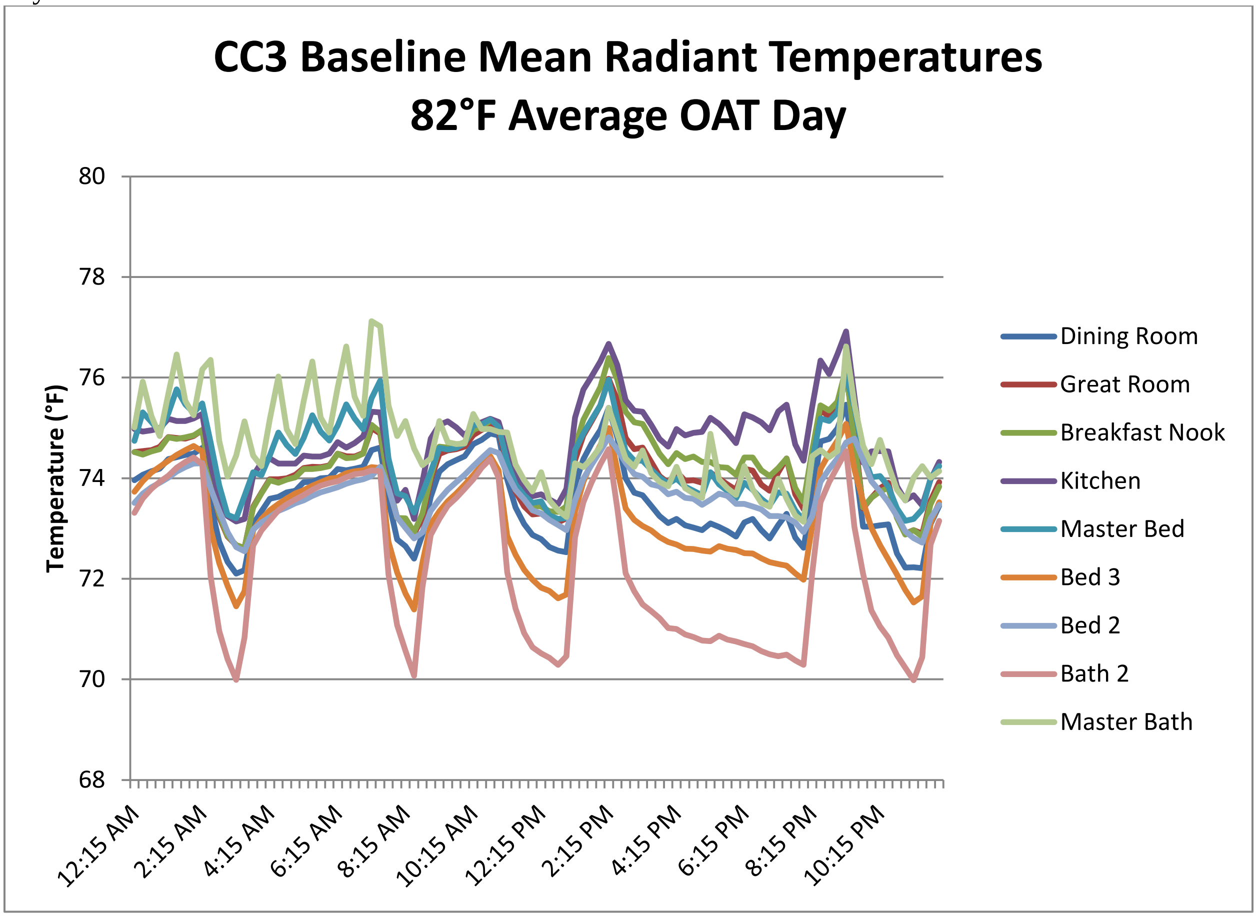

Figure 32: CC3 Baseline Mean Radiant Temperatures - 82°F Average OAT Day

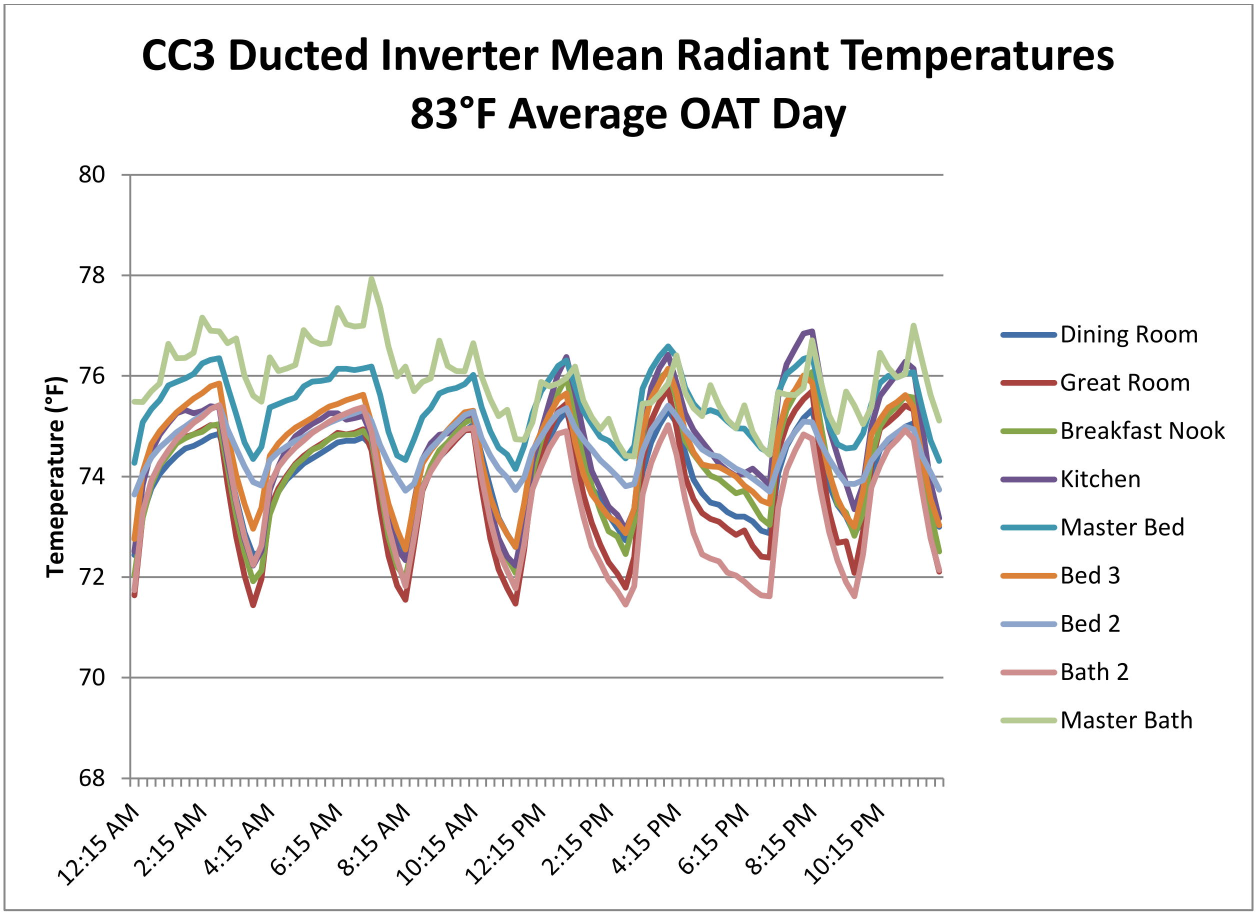

Figure 33: CC3 Ducted Mean Radiant Temperatures - 83°F Average OAT Day

5.5.2 Linear Regression Models

Figure 30 shows the measured cooling season data and the linear regression model for the baseline unit in CC3.

Thermostat setback testing was performed during several weeks in July and August of 2011. This data was not included in the analysis since the indoor air temperature was not maintained at a constant level. Figure 31 shows the measured cooling season data and the linear regression model for the ducted inverter system.

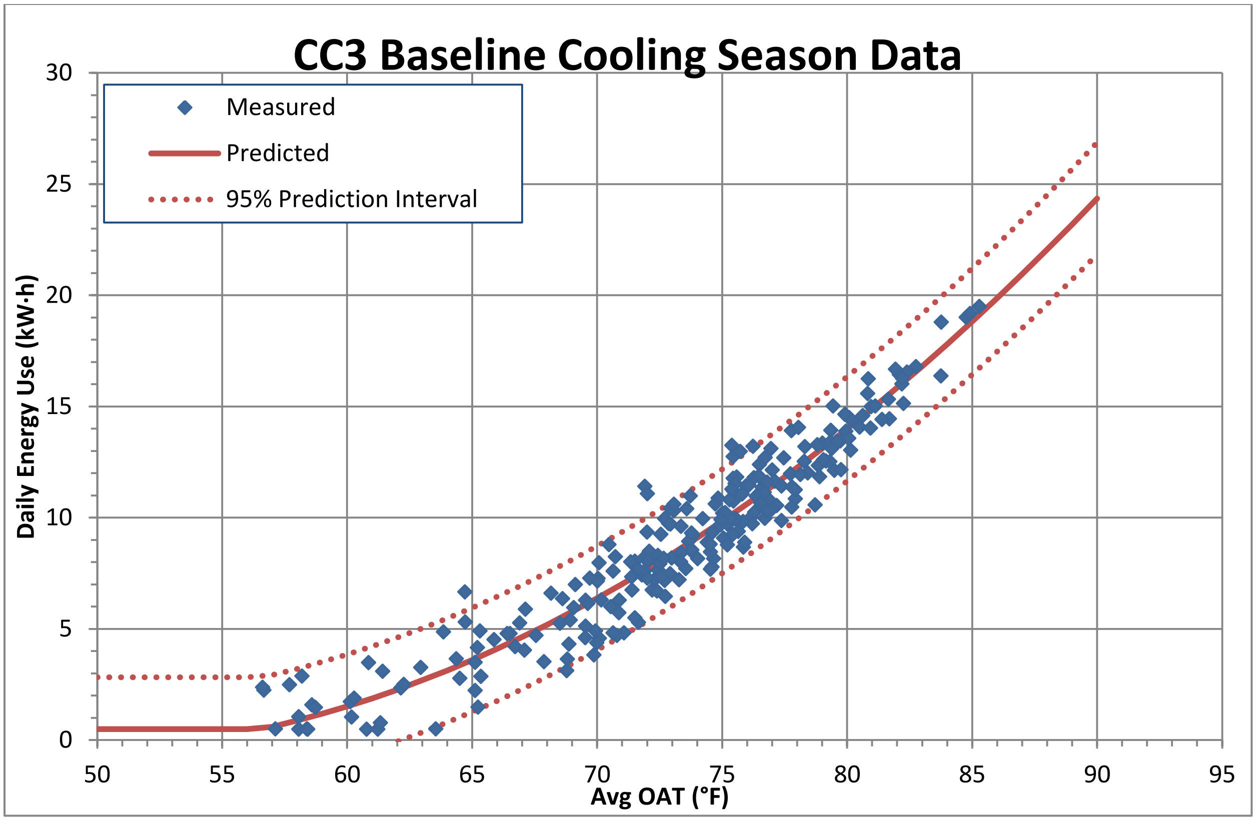

Figure 34: CC3 Baseline Cooling Season Data and Linear Regression

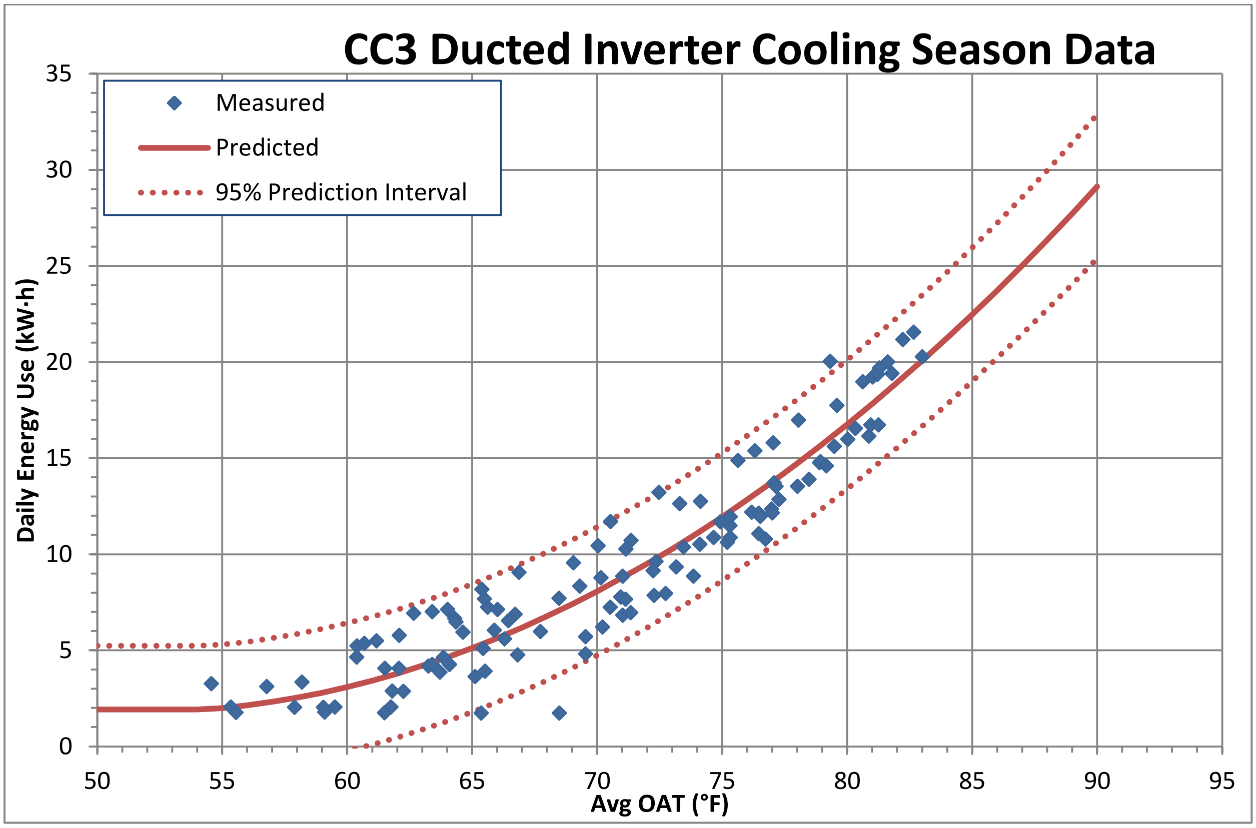

Figure 35: CC3 Ducted Inverter Cooling Season Data and Linear Regression

5.5.3 Performance Comparison

Figure 32 shows the linear regression models for both the baseline and ducted inverter systems plotted versus the average OAT. The baseline system used less energy across the entire temperature range.

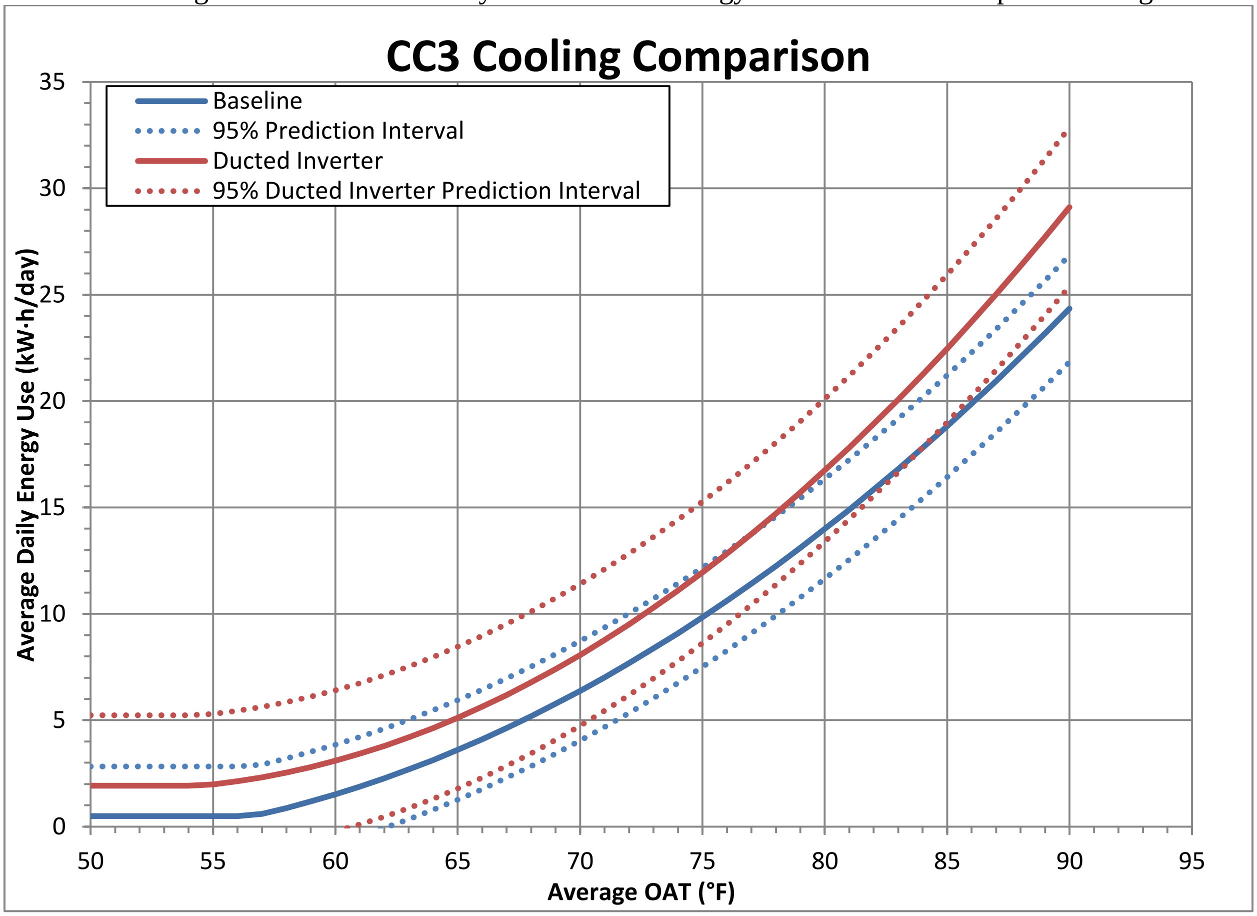

Figure 36: CC3 Cooling Comparison

When the linear regression models for the baseline system and the ducted inverter system are applied to the cooling season data from the Knoxville, TN TMY3 weather file, a prediction of the average cooling season energy use can be made. The running total of this energy use is shown in Figure 33 along with the average OAT from the TMY3 data file. The baseline system is predicted to use 1354 ±28 kW·h and the ducted inverter system is predicted to use 1660 ±38 kW·h with 95% confidence. This results in the ducted inverter system being predicted to use 306 ±49 kW·h, or 23%, more energy than the baseline system during a typical cooling season in Knoxville, TN.

The AHRI SEER ratings for the units would predict that the ducted inverter unit would use 11% less energy than the baseline system.

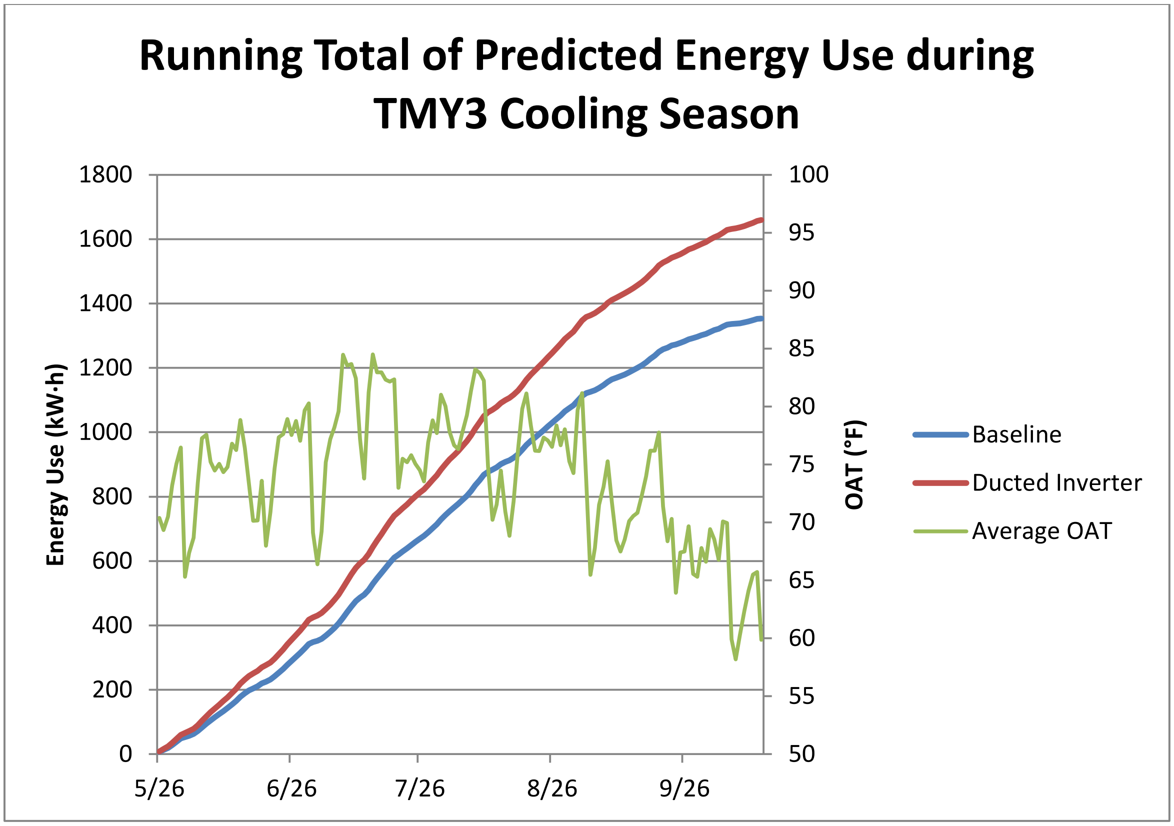

Figure 37: Running Total of Predicted Energy Use – TMY3 Cooling Season

5.6 ANNUAL PERFORMANCE

When both the heating and cooling season predictions are combined, the baseline system is predicted to use 4239±108 kW·h and the ducted inverter system is predicted to use 3684±76 kW·h over the course of an average year in Knoxville, TN with 95% confidence. This results in the ducted inverter system being predicted to use 554±132 kW·h less than the baseline system over the entire year.

5.7 DISCUSSION

While the heating season performance of the ducted inverter system appears very good when compared to the baseline system, the baseline system likely used much more energy than it should have due to it being limited to low stage operation. The operation of the ducted inverter unit did not display the expected characteristics of a unit with a variable speed compressor, and, instead, cycled frequently and operated at very constant power levels.

The performance of the ducted inverter system was lower than expected in the cooling season. This supports the notion that the ducted inverter unit is not functioning as intended. In contrast to the heating season, the baseline system did not appear to experience a performance penalty even though it was limited to low stage operation. It was also able to maintain the set point, which indicates that the cooling season performance of the baseline is representative of a properly operating unit. The AHRI ratings indicate that the ducted inverter system should have a clear advantage in cooling performance, but this was not seen in the measured data.

Recent data from an identical ducted inverter system at a retro-fit home indicates a much different pattern of power usage and cycling than the unit installed at CC3 exhibited. Upon investigation, it appears that a distinct change in the operation of the unit occurred during a service call soon after the installation of the unit. Power traces from before the service call were very similar to that of the other unit in the retro-fit house, while after the service call they appeared to be more akin to a single stage HP. The service call was scheduled after the unit was not keeping up with the load. Our logbook indicates that 1.5 lbs. of refrigerant were added to the system, but no other changes were communicated. We are currently investigating what other changes may have taken place during that service call.

6. CONCLUSIONS

The performance of the two inverter driven systems was not as expected. The multi-split system performed worse than a single stage 16 SEER HP in both the heating and cooling seasons. It did provide better temperature control than the baseline system during both heating and cooling, but did a poor job of controlling the relative humidity in the cooling mode. The ducted inverter system showed promising results in the heating mode, although the baseline system performance was degraded by its inability to run in high stage as well as apparently excessive defrost cycles. The cooling performance was worse than the baseline unit, however there is a very high likelihood that the ducted inverter system is not operating optimally due to a change in operating behavior that coincided with a service call.

It is difficult to draw conclusions regarding the achievable performance of these particular inverter driven systems. It is clear however that proper system design, installation, and service are crucial elements for any energy efficient HP installation. The increasing level of technology and complexity that accompanies higher efficiency systems increases the opportunity for errors during the commissioning of equipment that can have significant impacts on the installed efficiency.

Future reports will examine detailed air-side and refrigerant-side measurements in order to determine the absolute performance of a few inverter driven systems, while ensuring that the units are operating as the manufacturer intended.

7. REFERENCES

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Winkler, J., 2011. "Laboratory Test Report for Fujitsu 12RLS and Mitsubishi FE12NA Mini-Split Heat Pumps. National Renewable Energy Laboratory, September 2011.

8. APPENDIX A

Table 2: Linear Regression Model Coefficients and Statistics

	CC2				CC3			
	Heating		Cooling		Heating		Cooling	
	Baseline[1]	Multi-Split	Baseline	Multi-Split	Baseline	Ducted Inverter	Baseline	Ducted Inverter
Intercept	0.793	2.107	0.824	2.952	0.320	1.982	0.488	1.927
HDD	0.248	0.973			0.430	0.083		
HDD^2	0.035	0.016			0.029	0.016		
CDD			0.362	-0.176			0.255	0.110
CDD^2			0.034	0.040			0.014	0.018
RMSE	0.157(HDD+4)	3.576	1.943	2.171	3.513	2.163	1.183	1.669
R^2	0.977	0.988	0.922	0.905	0.973	0.953	0.984	0.968

1 Results after transforming the linear regression model back into standard units for this study

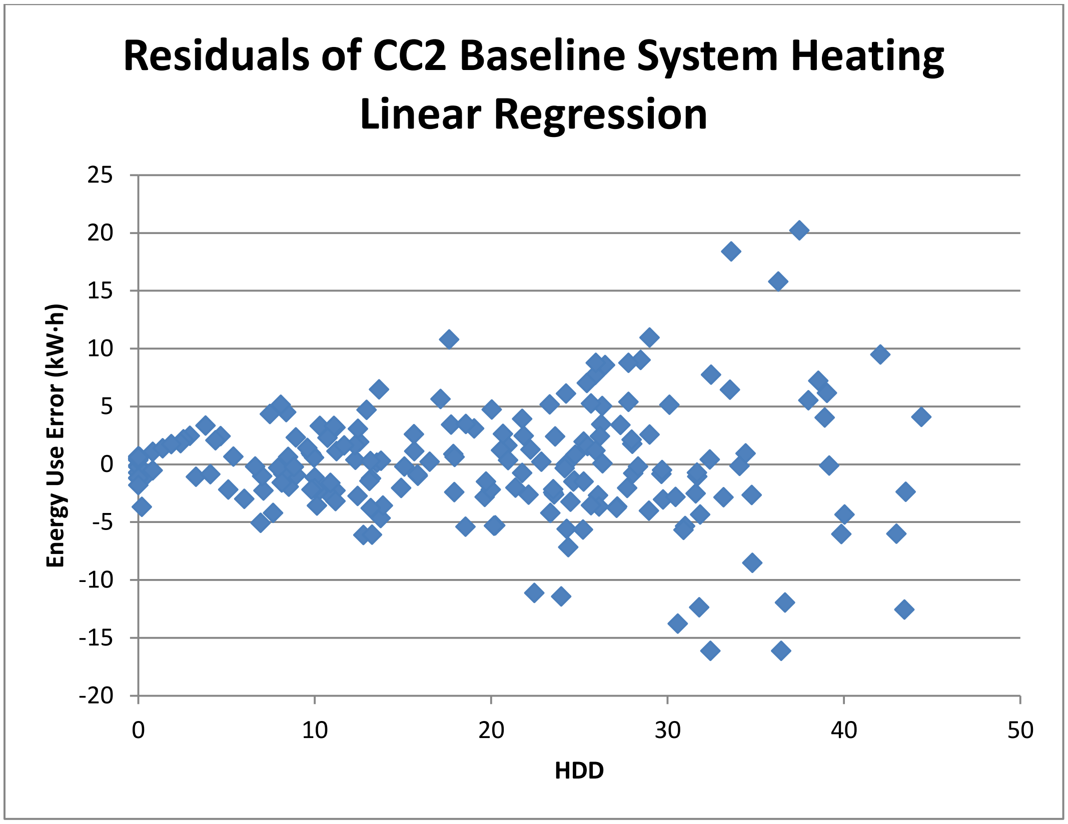

Figure 38: Residuals of CC2 Baseline System Heating Linear Regression

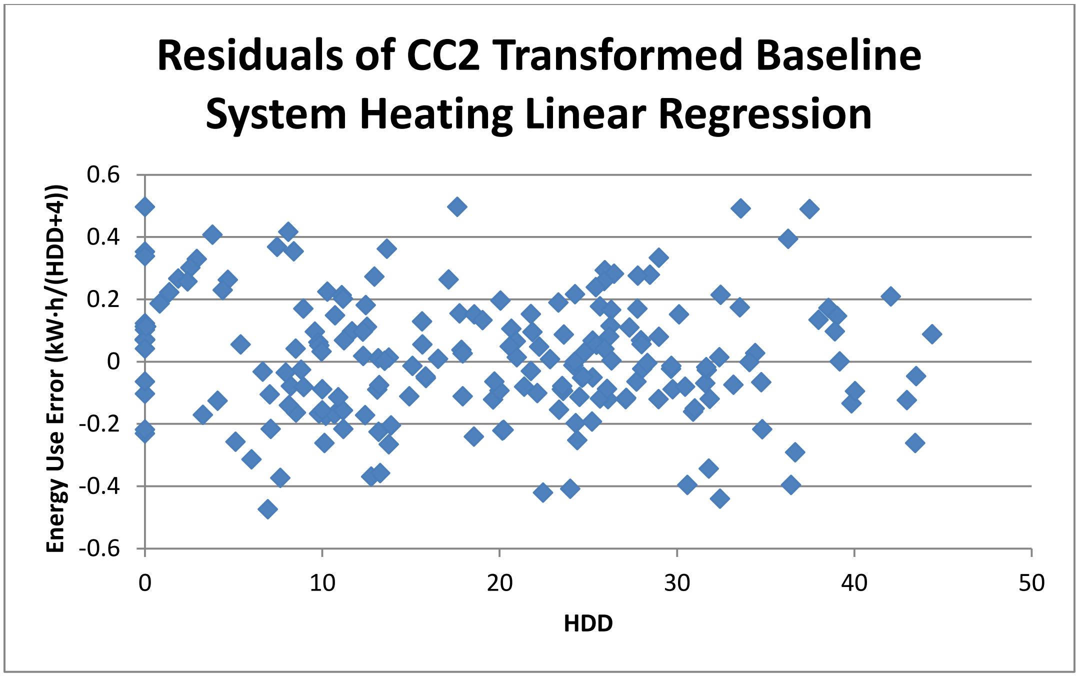

Figure 39: Residuals of CC2 Transformed Baseline System Heating Linear Regression

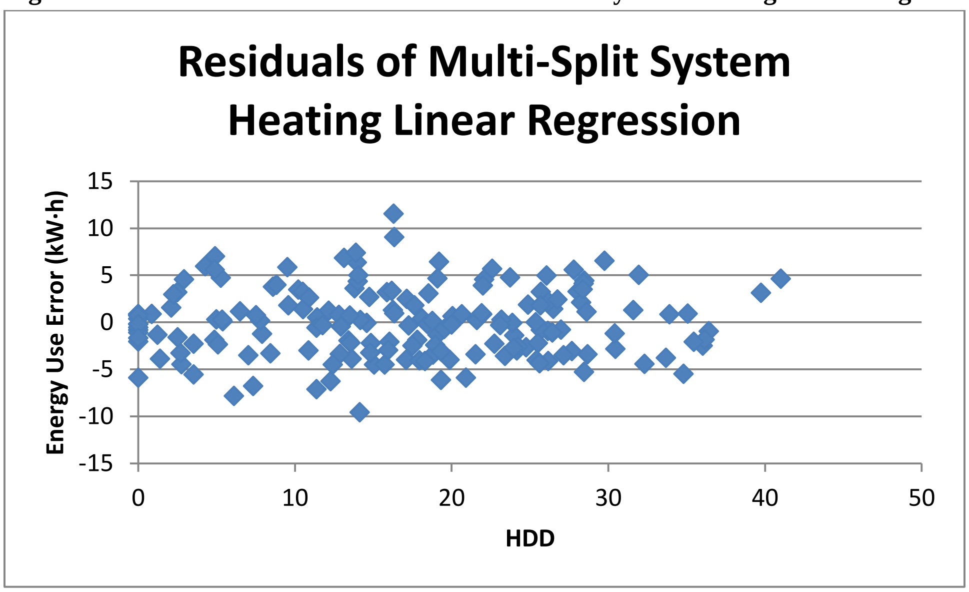

Figure 40: Residuals of Multi-Split System Heating Linear Regression

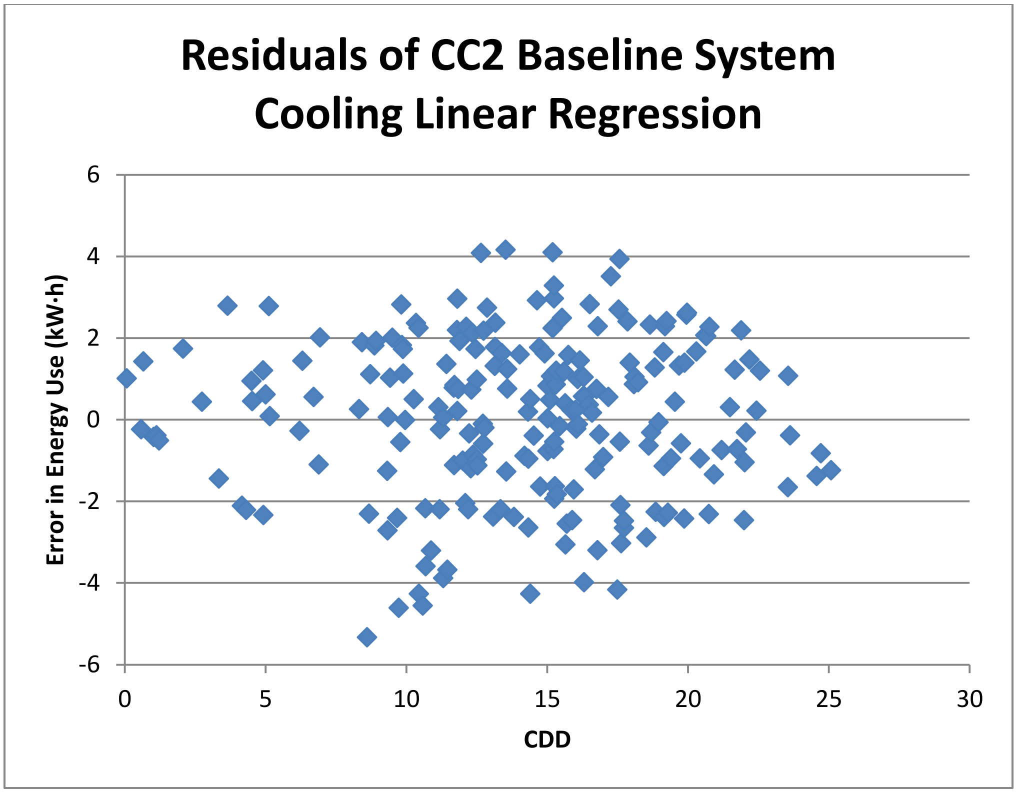

Figure 41: Residuals of CC2 Baseline System Cooling Linear Regression

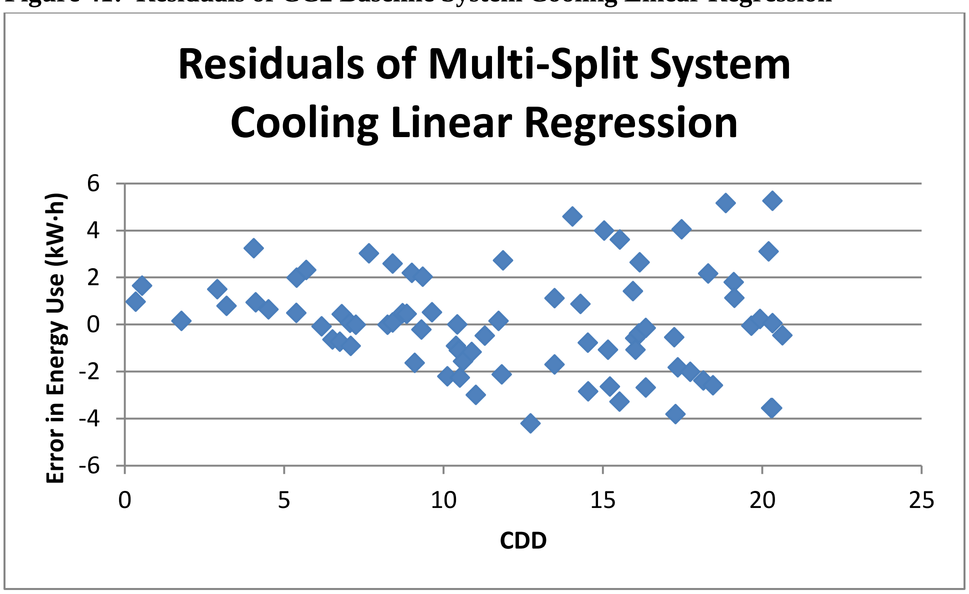

Figure 42: Residuals of Multi-Split System Cooling Linear Regression

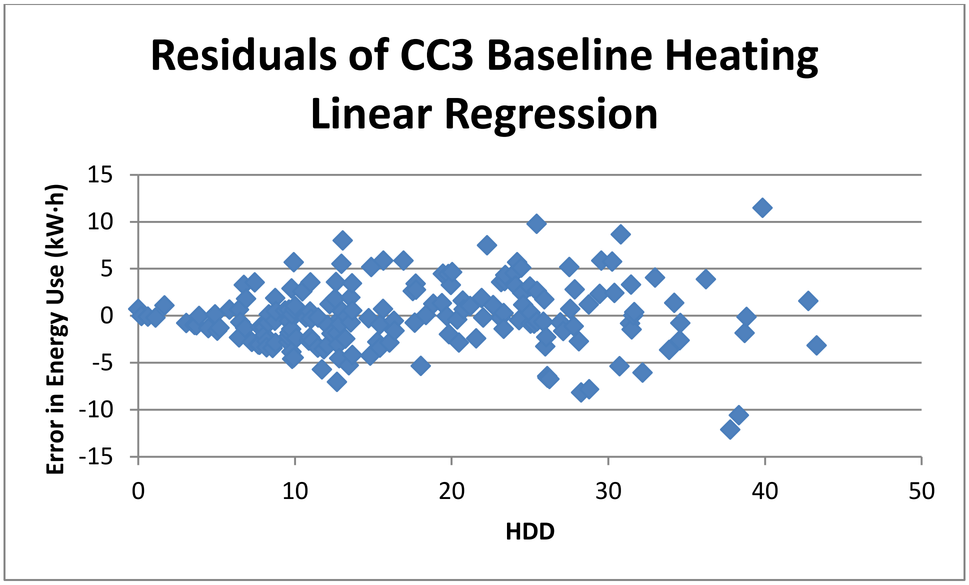

Figure 43: Residuals of CC3 Baseline Heating Linear Regression

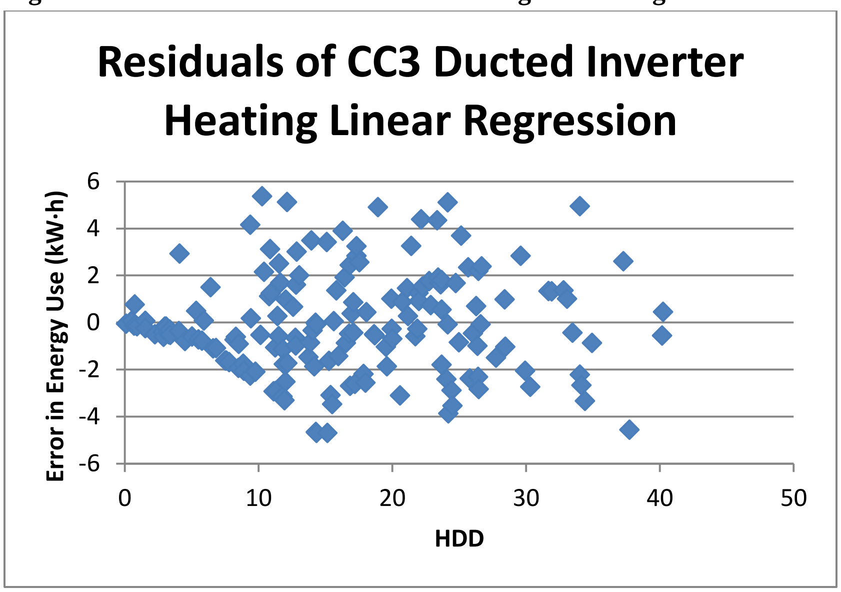

Figure 44: Residuals of CC3 Ducted Inverter Heating Linear Regression

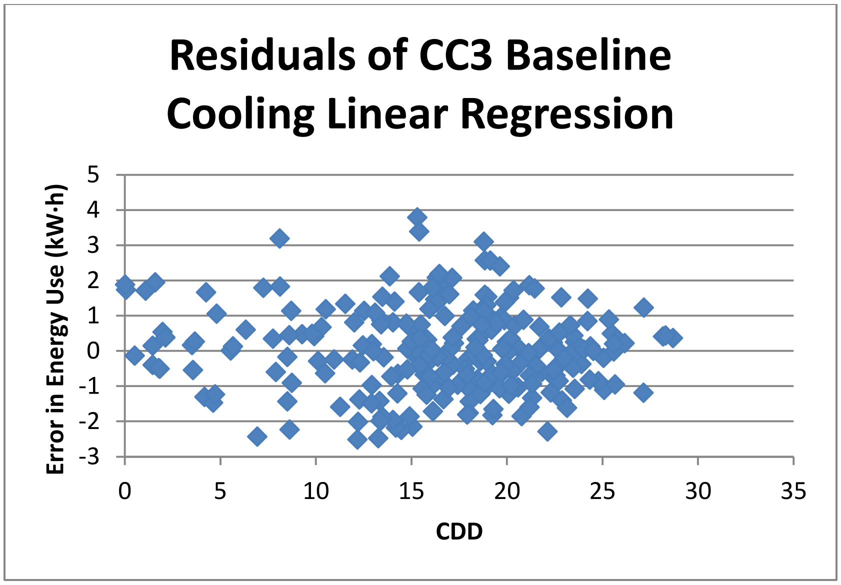

Figure 45: Residuals of CC3 Baseline Cooling Linear Regression

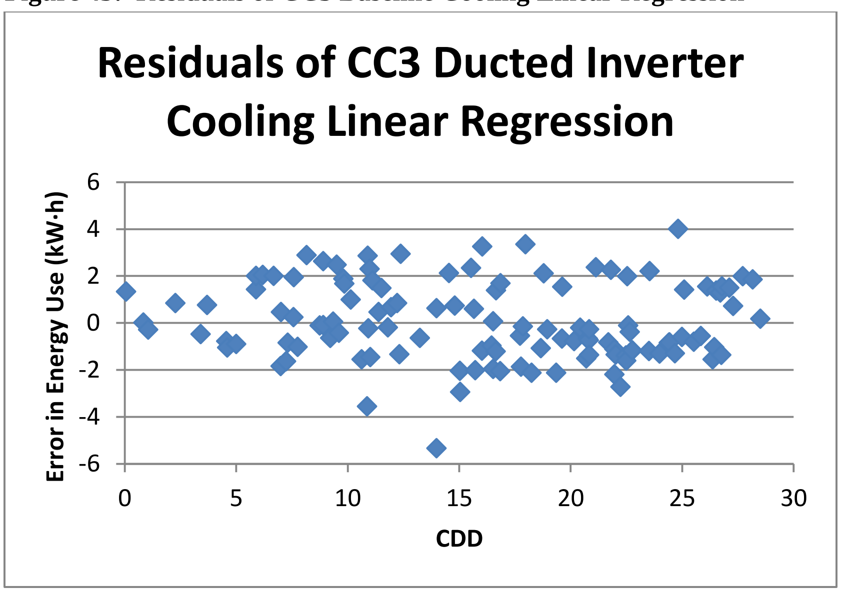

Figure 46: Residuals of CC3 Ducted Inverter Cooling Linear Regression