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Variable Rate Rooftop Unit Test (VRTUT) Report

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Executive Summary

The Northwest Energy Efficiency Alliance (NEEA), the Bonneville Power Administration (BPA), New Buildings Institute (NBI), and the University of Idaho Integrated Design Laboratory (IDL) collaborated to measure the energy performance of higher-efficiency-rated products in the unitary, vapor-compression, direct-expansion (DX) package rooftop unit (RTU) class of commercial building heating, ventilation, and air-conditioning (HVAC) equipment.

Estimates in the Northwest Power and Conservation Council's (NWPCC's) Sixth Power Plan (NWPCC 2010) indicate that DX package RTUs are found on over forty percent of rooftops of the 1.3 billion square feet of commercial buildings in the NEEA and BPA Pacific Northwest (PNW) four-state (including western Montana) service area. NBI estimates the existing RTU population in the region at over 400,000 units, with nearly sixty percent sized at six tons or less.

The project sought to use experimental data to evaluate the operational capabilities of three RTUs, first in a relatively controlled lab setting and then in a field operating setting. The primary goal of the study was to analyze a typical (or baseline federal minimum efficiency rated) RTU alongside two examples of an emerging class of advanced-performance RTUs. In addition to accumulating formal project results, the team also wanted to study the interplay between operational modes, control approaches, and sensible loading while using an energy signature-based approach to summarize energy performance. Since the projection of lab results is limited to the conditions under which the RTU is tested, the secondary goal of the project was to create and test an initial Physical Model of the RTU. The RTU Physical Model would facilitate the projection of performance results under varying conditions. Overall, the team hoped to create a useful modeling approach for comparing RTU performance that includes economizer operation and advanced fan modes, both of which can have dramatic impacts on energy usage, but which are not included in standardized RTU testing for performance metrics.

NBI chose a Trane Precedent ("Trane") as the baseline to represent a code-level RTU because its nameplate specifications are in line with current (2011-2012) national energy code requirements for HVAC equipment in this product class. The AAON RQ series ("AAON") and Daikin McQuay Rebel ("Daikin") served as the high-performance models. All three units were five-ton electric direct-expansion (DX) units. This project did not measure heating performance. In 2012, a ten-ton Rebel RTU met the US Department of Energy (DOE)/ Commercial Building Energy Alliances (CBEA) High Performance Rooftop Unit Challenge with an integrated energy efficiency ratio (IEER) of 18.0 or higher. NBI tested each unit in its lab (NBIL) under two internal sensible loading conditions with constant ventilation air fraction, schedules, setpoints, and economizer settings.

NBI compared the performance of each unit using an "energy signature," which is a standardized plot of daily energy usage versus average daily outdoor dry-bulb air temperature. Using this standard enabled NBI to project the energy usage in different locations using the corresponding Typical Meteorological Year (TMY3) data. This report limits projections to the "Cooling Season," defined as May 1 to October 31, and excludes the heating season.

The lab results showed that the energy signature of each unit responded predictably to the increasing sensible internal loading. The Trane unit operated as a normal on/off DX unit. The projected cooling season energy usage for Portland, OR ranged from 4,130 kWh to 3,340 kWh depending on sensible internal loading, with 2,109 kWh of that projection estimated to be fan energy. The economizer did not function during testing.

The Daikin unit operated with a continuously varying inverter-based compressor that resulted in a much smoother power profile over the course of a test day. The projected cooling season energy usage for Portland also ranged from 3,196 kWh to 2,792 kWh depending on sensible internal loading, with 972 kWh estimated to be fan energy. The economizer operated with integrated compressor operation at low ambient temperatures; this feature allows for control over the supply air temperature and for humidity control in more humid climates. Researchers noted that in the test climate, the integrated economizer might have resulted in increased energy use over an economizer-only approach.

The AAON unit operated in a manner more similar to the Trane than to the Daikin, switching fan and compressor as a simple on/off unit, despite its advertised “Digital Scroll” capability as a variable capacity compressor, though notably the unit was configured with a control board specified by Fred Meyer rather than using the AAON controls. Its projected cooling season energy usage for Portland ranged from 5,105 kWh to 3,174 kWh depending on sensible internal loading and fan control settings, with 1,862 to 1,012 kWh of that estimated to be fan energy. The economizer did not function during the first two control modes, but was enabled for the third mode.

Overall, the projections of cooling energy usage suffered due to smaller-than-desired numbers of daily points comprising each energy signature; however, the team found the results to be somewhat useful. The projected cooling season energy usage did correlate to some extent with published Seasonal Energy Efficiency Ratio (SEER) values.

The initial RTU Physical Model proved very effective at fitting the data points of the energy signature for each unit from the lab testing; the data from the lab testing correlated well with the tuned parameters of the model. These positive results suggested the model is robust and reinforced the conclusions of the field comparison between the Daikin and Trane units.

NBI and IDL personnel installed and measured the Daikin unit in the field at a Fred Meyer (Kroger Corporation) box retail store in Nampa, Idaho (in the Boise metro area). Installation delays left the researchers unable to collect data from the AAON unit in time for the 2012 cooling season analysis. As expected, the store’s operating schedule, setpoints, and outside air fraction all differed at the field location. The Daikin unit operated in two control modes, one with a constant fan and one with a variable speed fan. Using the energy signature method for analysis, NBI projected the cooling season energy usage for the first mode in Boise to be 3,932 kWh, including estimated fan energy of 1,192 kWh. For the second (variable speed fan) mode, NBI projected the cooling season energy usage for Boise to be 4,025 kWh, including estimated fan energy of 1,546 kWh. In both modes, the economizer also operated with the integrated compressor.

Researchers fit the RTU Physical Model to the data points from the first (constant fan) mode of field operation only. They applied the resulting parameters of the model to an RTU Physical Model of the Trane from laboratory testing to establish an energy signature for the hypothetical Trane performance. During this exercise, researchers also took into account the impact of the non-functional economizer on the Trane, using the model to examine the results. The results showed that, in the Boise location, the Daikin would save 2,442 kWh, or thirty-seven percent, over the Trane unit with a non-functioning economizer; by contrast, it would save 1,915 kWh, or thirty-one percent, over the Trane with a functional economizer. Again, these data represent the cooling season results; the heating season findings may present additional fan energy savings. Since the Energy Efficiency Ratio (EER) and SEER of the Trane are near the levels for some code baselines, the team considered the Physical Model as a potential alternative method for code comparison.

Overall, the team found that the three RTUs varied in performance with sensible internal loading, as expected. The controls configuration, especially the fan energy usage, greatly affected the projected cooling season energy usage.

Additionally, NBI found that the RTU Physical Model approach (summarized in Appendix A) for projecting code level energy use – including economizer and fan energy effects – was possible, and that initial data fit projections well. NBI looks forward to further validation of the Physical Model in 2013.

1. Introduction

The Northwest Energy Efficiency Alliance (NEEA), the Bonneville Power Administration (BPA), New Buildings Institute (NBI), and the University of Idaho's Integrated Design Laboratory (IDL) collaborated to measure the energy performance of higher-efficiency-rated products in the unitary, vapor-compression, direct-expansion (DX) package rooftop unit (RTU) class of commercial building heating, ventilation, and air-conditioning (HVAC) equipment. The new products have combinations of controls and components that provide variable rates of speed on fan motors and that can modulate compressor output.

BPA and NEEA selected NBI to propose the research design, manage and implement the study, and analyze project results in conjunction with IDL staff in Boise, Idaho. An expanded project advisory team reviewed the project research results described in this report. NEEA had initially created the team for the indirect/direct evaporative DX hybrid RTU project (now indirect/direct evaporative only) in Idaho. The team includes staff from BPA, National Renewable Energy Laboratory, the Southwest Energy Efficiency Project, the Western Cooling Efficiency Center, and Western Environmental Services Corporation (Wescor). In addition, the manufacturers of the test units participated in the performance reviews.

1.1. Background

Newer, more advanced package RTU products have recently become available in the commercial market. The Air-Conditioning, Heating, and Refrigeration Institute (AHRI) provides energy efficiency ratings for these RTU products in accordance with its rating standards. The products include the AAON RQ Series, which has been available since 2010, and the Daikin McQuay Rebel, which was more recently released. Both are package units with natural gas heating options. Both units have DX coils for cooling; economizers; electronically commutated fan motors; variable speed controls on fans; and incorporate direct drives on evaporator and condenser fans. The RQ unit has a Copeland Scroll Digital™ compressor and the Rebel unit has an inverter scroll compressor.

1.1.1. Rooftop Units (RTUs) in the Pacific Northwest

Estimates in the Northwest Power and Conservation Council's (NWPCC's) Sixth Power Plan (NWPCC 2010) indicate that DX package RTUs are found on over forty percent of the rooftops of the 1.3 billion square feet of commercial buildings in the NEEA and BPA Pacific Northwest (PNW) four-state (including western Montana) service area, or just under half of all the commercial building stock. NBI estimates the existing RTU population in the region over 400,000 units, with nearly sixty percent sized at six tons or less. An estimated thirty-five percent of the units are between ten and twenty years old, with another estimated sixteen percent more than twenty years old; these aging units will increasingly drive RTU repair, retrofit, and replacement opportunities.

Under current building energy codes and utility incentive programs, RTU performance is designated by an efficiency rating. Common ratings include the Energy Efficiency Rating (EER), the Seasonal Energy Efficiency Ratio rating (SEER), and the Integrated Energy Efficiency Rating (IEER), all of which are specified by AHRI. However, AHRI does not require EER or

IEER ratings on equipment sizes less than five tons in capacity. These ratings provide a consistent measure of RTU performance under a given set of laboratory conditions; however, studies show that real-world RTU performance varies significantly due to many factors in addition to rated efficiency. Sources of variation include economizer settings; building loading conditions; installation practices; maintenance frequency; manufacturing variability; scheduling; fan settings; thermostat setpoints; and other factors.

In recognition of these variations, NBI began working with the Northwest Regional Technical Forum to understand how real-world RTU energy data could be used to establish performance expectations.

1.1.2. Regional Technical Forum Protocol

Given the large potential for divergence between the anticipated efficiency suggested by AHRI ratings and that seen in the field, the Northwest Regional Technical Forum (RTF) approved a Standard Protocol for Measurement of Fan and Cooling Savings from Commercial-Sector Packaged and Split System HVAC Units (Regional Technical Forum 2012). The primary goal of the savings protocol is to define a methodology for estimating annual electrical energy use of an existing RTU, based on a relatively short period (three to four weeks) of field monitoring of that existing unit using one-minute-interval data. The estimated annual energy use can then be compared after a second monitoring period (three to four weeks) following repair or retrofit of that same unit.

This approach is aligned with the guidance provided in the International Protocol for Measurement and Verification (IPMVP) (Efficiency Valuation Organization 2013), under Option A: Retrofit Isolation. Specifically, Option A treats each RTU as an isolated system that is individually metered.

The RTF RTU protocol provided valuable background for the present study, and NBI has drawn upon it extensively. The accelerated timeline of the project meant that NBI wasn't able to explicitly follow the protocol, but NBI used the basic analytical methodology for this report.

The RTF protocol uses an energy signature relationship to compare RTUs. The energy signature consists of an X-Y axis graphic plot of energy usage versus dry bulb outdoor air temperature on a daily interval.[1]

Specifically, the RTF protocol provides a basis for establishing the energy usage characteristics for a certain RTU operating at particular conditions, which may vary as discussed above. In the NBI laboratory measurement portion of the RTU tests, the RTF protocol proved effective for comparing the units with controlled conditions. In addition, the RTF protocol, using minute-by-minute data, reveals nuances of the control system. It also allows for separation of the energy used for primary ventilation air movement with the compressor from the remainder of the plant's energy use.

[1] The RTF protocol requires more detailed analysis using the one-minute interval data to supplement the daily energy signature comparison.

1.1.3. Need for a Model and General Project Approach to Assessing RTUs

The RTF protocol and the controlled laboratory test conditions established by NBI allow for a good comparison between a federal minimum energy code RTU and advanced higher-efficiency RTUs. However, in field conditions where fewer data points are available, and when assessing comparisons between units under other conditions, a model that can be calibrated to those conditions is necessary in order to determine the savings of an advanced unit over a standard one.

For example, the variation in the amount of outdoor ventilation air will greatly influence the energy usage of the RTU due to greater loading across the evaporator coil. A properly-constructed model will allow the analyst to account for this variation in data that can be observed in one-minute intervals.

The amount of outdoor ventilation air is one of several relatively easy-to-establish independent variables for determining RTU loading. Internal loading, which is comprised of many effects, is a more complicated variable that is relatively difficult to establish. In order for a model approach using limited field data to be effective, using field temperature and energy measurements to establish a methodology for assessing the thermal loading of the space would be helpful. NBI used both laboratory and field testing to examine this approach.

Ideally, the general use of an energy signature-based model for code projection provides a real-world, temperature normalized-based energy signature projection for code level “typical” RTU performance that can be used to establish more accurate assessments of energy savings for advanced RTUs. This approach can then form the basis of utility programs directed toward advanced RTUs.

This project used an initial model created and calibrated using the laboratory test data. The field data provided a chance to test the model by projecting how the typical unit would use energy given the field conditions.

Enhancement of this model is the subject of ongoing research, but this report presents some initial results. Appendix A describes the model in detail.

1.2. Project Goals and Scope

The research team sought to use experimental data to evaluate the operational capabilities of three RTUs in a relatively climate-controlled conditioned space setting. In particular, the team wanted to study the interactions among operational modes, control approaches, and sensible loading while using an energy signature-based approach to summarize energy usage. These data, along with the known conditions, would serve as a basis to create and test an initial Physical Model of an RTU. The team also sought to use the field testing to test the model as a way to project a code minimum level unit operating in the same conditions. The field data would aid in exploring a fundamental modeled approach to comparing RTU performance that includes economizer operation and advanced fan modes.

Specific research questions included:

- How does the energy signature change in response to changes in thermal loading (kWh/day of internal building loads) for each of the three units?
- Which control nuances typify high performance units available today, and what are the performance implications of setup control choices?
- What are the estimated cooling season (May 1 to October 31) energy savings for an advanced unit at the laboratory conditions using a basic energy signature projection?
- Can the researchers derive a functional RTU model; what is the minimum dataset needed; and how does it clarify the expected savings of the higher efficiency units in the lab and in the field?

Another research question follows:

- How can published SEER ratings compare to projections of cooling season energy use with energy signatures and the RTU Physical Model?

1.3. Relevant RTU Research in Literature

The research described in this report focused on the interactions of energy used by the primary components of the RTU: the supply fan, condenser fan, compressor, and the actuation of the economizer.

Recent research at Pacific Northwest National Laboratory (PNNL) examined the effectiveness of several control approaches using the US Department of Energy's EnergyPlus modeling software (Pacific Northwest National Laboratory 2011). Although it did not use an energy signature approach, the research findings showed that significant savings for RTUs could be found in two areas: supply fan controls and economizer usage. PNNL researchers are conducting a large-scale follow-on field trial that will have results in 2013 that may greatly inform the use of the RTF protocol and the RTU Physical Model.

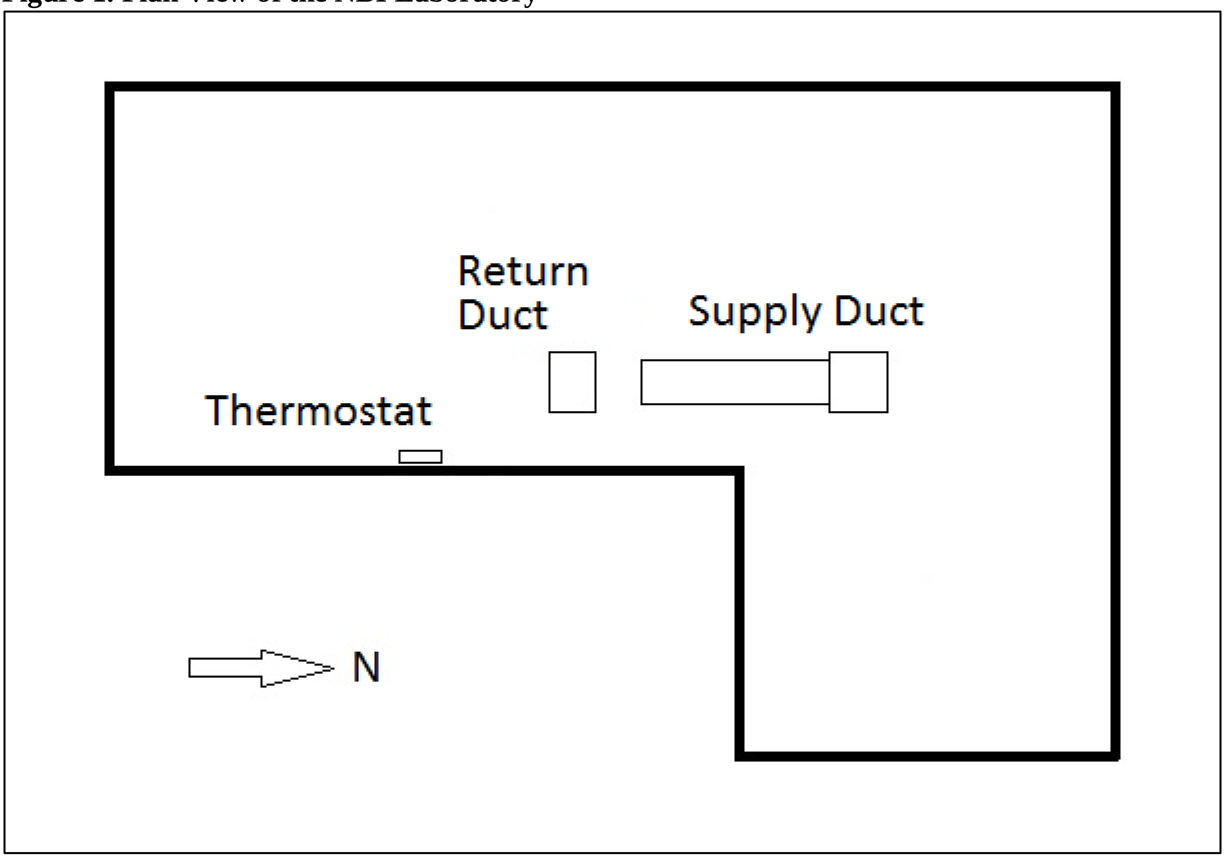
2. Methodology

2.1. Test Sites and Tools

2.1.1. New Buildings Institute Laboratory

The team conducted the first phase of testing at New Buildings Institute's Laboratory (NBIL), which is co-located with NBI's Vancouver, Washington headquarters. The lab consists of a 956-square-foot indoor test space that is isolated from the rest of the office facility with modest insulation and an air barrier to reduce infiltration. A dedicated federal minimum (EER 11.0) RTU serves this single zone, which was swapped out prior to testing each of the two advanced RTUs in its place. RTU loading can be controlled by adjusting the output of heat sources inside the zone that the lab uses to simulate varying levels of occupancy, as well as light and plug load density. The lab is extensively instrumented with a suite of data-logging temperature, airflow, and power sensors.

Figure 1. Plan View of the NBI Laboratory



2.1.2. Field Sites

In addition to the testing it conducted at the NBIL, the team conducted additional testing at a retail box store (Fred Meyer/Kroger Corporation) near Boise, Idaho. NBI chose the store because of an existing testing history with RTUs at the facility; a cooperative relationship with the regional corporate facilities manager; and the proximity of the Integrated Design Lab (IDL) at the University of Idaho with its expert engineering and research staff.

The team used the daytime and nighttime setpoints (schedules/temperatures) from the field site as the laboratory testing setpoint conditions for all testing.

Initially, NBI planned two field measurement installations, one each for the AAON and Daikin units; however, due to unexpected installation scheduling issues, it ultimately tested only the Daikin in the field. NBI will issue follow-on results for the AAON in the fall of 2013.

2.1.3. ClimaCheck System

ClimaCheck, a Swedish-based refrigeration cycle performance monitoring/fault detection and diagnostic tool, was applied to the RTUs (and is also used in industrial/commercial refrigeration systems). The ClimaCheck analysis provided data from the refrigeration side of the RTU operation, while NBI's data was focused on using air side instrumentation for project results.

Data from the ClimaCheck system is included in NBI's analysis where appropriate. A complete report on each RTU is included in Appendix B.

2.2. Test Units and Test Protocols

The following sections describe NBI's testing of the three RTUs during this study, outline the test protocols for the laboratory and the field, and discuss significant deviations from the original proposed test protocols.

2.2.1. RTU Specifications

The goal of this study was to analyze a typical (code baseline) RTU alongside two examples of an emerging class of advanced performance RTUs. NBI chose a Trane Precedent ("Trane") as the baseline to represent a code baseline RTU because its nameplate EER specifications are in line with current code requirements in many regional jurisdictions. NBI selected the AAON RQ series ("AAON") and Daikin McQuay Rebel ("Daikin") as the high-performance models based on their specifications. Table 1 shows the published specifications for the three, five-ton RTUs tested during the course of this study. Notably, the Daikin recently met the DOE High Performance Rooftop Unit Challenge (Energy Efficiency & Renewable Energy 2012).

Table 1. High-Level Feature Summary of the Test RTUs

	Trane Precedent	AAON RQ Series	Daikin McQuay Rebel
Short name used in report	Trane	AAON	Daikin
Nominal Size, Tons	5	5	5
SEER, AHRI	13.0	14.8	18.0
EER, AHRI	11.0	12.7	12.7
Cooling Capacity kBtu/hour, AHRI	62.27	63.50	61.75
Economizer Control	Dry bulb	Comparative	Comparative

		Enthalpy	Enthalpy
Supply/Condenser Fans	Direct drives	ECM/VFD Direct drives	ECM/VFD Direct drives
Compressor	Scroll	Scroll Digital™	Inverter scroll
Refrigeration Metering Device	TXV	TXV	EXV
Exhaust Air	Barometric	Barometric	Barometric
Model Year	2011	2012	2012
Web Link	Trane Precedent™	AAON® RQ Series	Daikin McQuay™ Rebel

Notable differences among the RTUs include the use of electronically commutated motors for the direct-drive supply fans in the advanced units, as well as modulating capacity compressors: an inverter-based scroll compressor in the Daikin unit and a Digital Scroll compressor in the AAON unit. In addition, the Trane unit utilized only a dry-bulb, non-integrated economizer, while the advanced units employed an option for a comparative enthalpy economizer control.

NBI conducted a combination of lab and field testing from July 1 to November 11, 2012. It conducted lab testing at the NBIL and field testing at a box retail store in the Boise, Idaho area. The test protocol initially called for field testing at two box stores, but installation cranes were not available to complete the AAON field installation in time for the team to gather meaningful data during the 2012 cooling season. The Daikin field measurements took place from September 15 to October 16, 2012. Automated data acquisition will remain in place until August 2013. At this time, no support is available for performance data analysis for the 2013 heating or cooling seasons.

Figure 2. The Three Test RTUs: Trane (left), AAON (center), and Daikin (right)

This series of testing produced eight datasets, shown in Table 2. Researchers tested each of the three units at the NBIL at two internal loading conditions (three internal loadings for the Trane); as mentioned above, they field tested the Daikin at only one Idaho retail box store where the sensible internal loading was not known.

Table 2. Test Datasets and Sensible Loading Descriptions for the Laboratory and Field Tests

RTU Unit	NBIL	Field Site
	Sensible Loading Description	

	Low Loading	High Loading	Ultra High Loading	As-Occupied Loading
Trane	x	x	x	
AAON		x	x	
Daikin		x	x	x

NBI derived the sensible internal loading (also referred to as “gain”) profiles from office space lighting and plug load data collected in previous projects. It based the Low and High Loading profiles on the low and high cases for sensible heating in watts per square foot (W/SF) and assumed additional sensible heating from occupants. The Ultra High Loading added additional sensible heating during the occupied period. The VRTUT Laboratory Test Protocol in the following section describes these loadings in detail.

The team conducted ongoing data analysis in conjunction with testing. In general, tests in the lab consisted of daily runs at a certain sensible loading condition with a fixed schedule, outside ventilation air fraction, setpoints, and test space orientation. In both the lab and the field, NBI made slight changes to the control settings of the units, primarily in fan controls. The following section outlines these changes and their impacts on daily energy usage.

2.2.2. Laboratory Test Protocol

The team installed and operated each RTU with standard configuration settings. Each RTU served an artificially loaded zone (empty office/storage space) measuring 956 square feet.

Entek mechanical contractors installed each unit at the same location on the roof of the NBIL, with one unit replacing another as the testing progressed. As necessary, Entek fabricated and installed a curb adapter to ensure proper alignment of each unit with the supply and return penetrations. With each installation, NBI installed airside sensors (described below) at equivalent points inside each unit and used a consistent location for outdoor air temperature sensing. The Trane unit, which was tested first, was already in place at the NBIL.

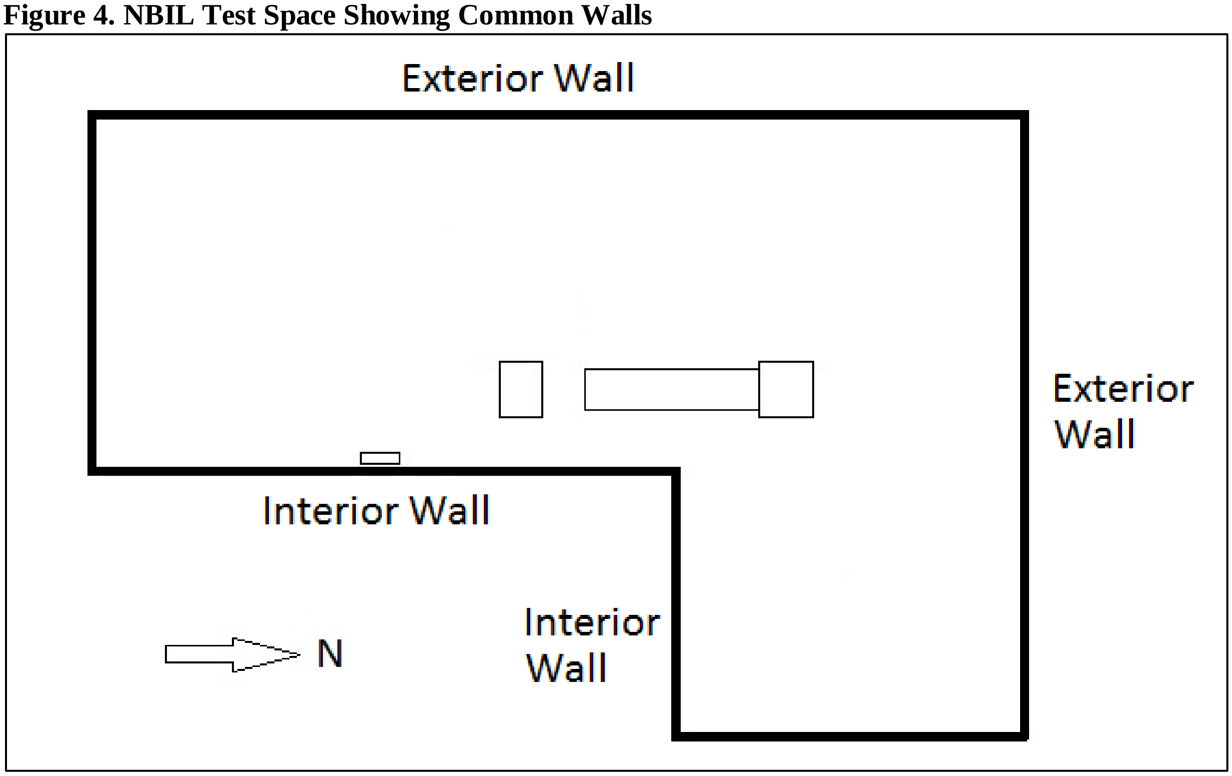
The ClimaCheck system provided detailed information on the refrigeration component of each unit. ClimaCheck personnel installed the instrumentation for each unit in equivalent locations and collected data remotely via a cellular connection.

Figure 3. The ClimaCheck System

Appendix B includes details regarding the measurement points and sensor accuracy for ClimaCheck.

The ductwork and airside instrumentation remained the same for each unit. The space served by the test RTUs shared two common walls sealed internally with clear plastic to reduce air infiltration. NBI maintained the spaces with shared walls at the same thermal conditions for all tests to minimize conductive or infiltrative contributions through the walls.

Figure 4. NBIL Test Space Showing Common Walls



Each RTU included a method for sensing room temperature, and controls for maintaining the temperature to the specified setpoints and schedule. While each unit had specific control equipment, all indoor temperature sensors used the same sensing location to ensure that each unit saw the same airflow dynamics in the zone.

2.2.2.1. Lab Testing: Sensible Internal Loading

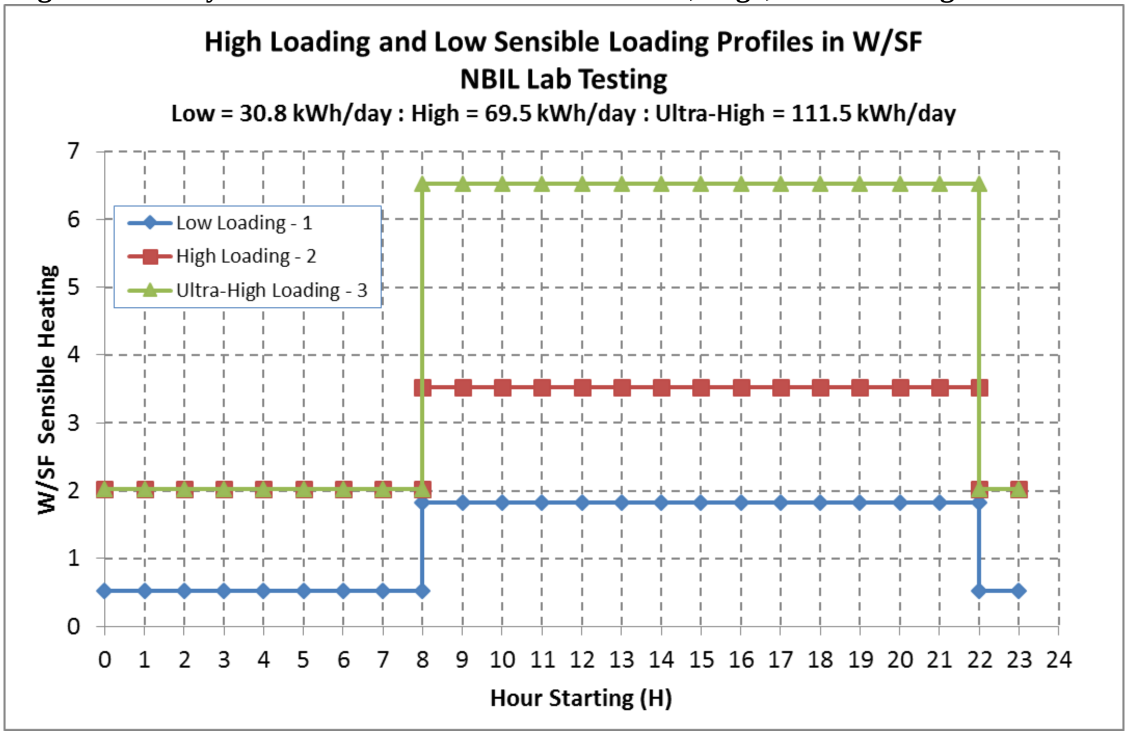
Researchers used two internal loading levels ("High" and "Ultra High") to represent the extremes of typical loading expected in an office setting. Figure 5 shows the daily load profile for each. Researchers tested the Trane unit at an additional level ("Low") that was not repeated for the other two units. NBI used detailed data for typical lighting and plug load usage profiles in office spaces, as well as assumptions of occupant metabolic heating, to construct each profile.

The Low loading profile elicited insufficient response from the first test unit (shown as the "Low Loading" profile in Figure 5). Researchers used this profile only for the first (Trane) unit, as they determined that this level of sensible internal gain was too low to provide useful results. Thus, the remainder of the laboratory testing proceeded with "High Loading" as the lowest loading case and "Ultra High Loading" as the highest loading case for the remaining test datasets.

Researchers controlled sensible loading using 1500W electric resistance heaters switched by a Reliable Mach-Pro building automation system. The tests included no latent loading through humidification or other means.

Researchers conducted tests with the specified loading repeated daily for the duration of each loading dataset. The loading profiles simulated an occupied schedule of 8:00 a.m. to 10:00 p.m. and an unoccupied schedule of 10:00 p.m. to 8:00 a.m. No weekends were simulated in order to achieve a valid signature in a short period of time.

Figure 5. Hourly Sensible Internal Load Profiles: Low, High, and Ultra High



2.2.2.2. Lab Testing: System Configuration and Controls

Each unit used a specific set of equipment to operate as discussed below. Notably, none of the units were programmed with demand-controlled ventilation. Fixed minimum position damper controls regulated ventilation air. The team calculated the fraction of outdoor air using the calibrated flow rate of each unit and a dry-bulb temperature mixing calculation when the outside air and return air were significantly different. The team adjusted damper settings to ensure achievement of the desired percentage of ten to fifteen percent.

Trane

The Trane RTU represents a typical unit with a very simple outdoor control system. This unit performs no indoor control functions, but relies on a standard five-pin thermostat to call for cooling, heating, and ventilation. The compressor, supply fan, and condenser fan are all single-speed and operate simultaneously when a call for cooling occurs. A separate sensor/controller actuates the economizer to allow the supply fan to turn on at a single speed while opening the outdoor air damper to the one hundred percent setting. The RTU maintains ventilation air by fixing the damper in a minimum position using a potentiometer.

The Trane unit operated as designed at an airflow of 1,588 CFM and an approximate ten percent outdoor air fraction. The outdoor control settings permitted no variation other than the setting of the outdoor air fraction and the economizer changeover temperature (placed on setting “C”).

Daikin McQuay

The Daikin had a more sophisticated outdoor controller that modulated all the components and also managed indoor controller functions. The outdoor controller was capable of network communication, but the team made all setting changes for this project using the manual user interface on the controller. A single temperature sensor monitored indoor conditions (at the location shown in Figure 6), although the team implemented setpoints, deadband, and other control settings in the outdoor controller. The compressor, supply fan, and condenser fan are all variable speed. The control system integrated economizer and compressor operation and ventilation airflow.

Notably, the installer set up the Daikin in a control mode called Discharge Set Point (DSP) so that the unit primarily maintained a supply air discharge setpoint (55 degrees F) and allowed the space temperature to float in a wider deadband. This inadvertent control setup may have caused some variation in results, although the indoor temperature did not stray far outside the test protocol of one degree F. The RTU operated the supply fan in a fixed speed mode (840 CFM) with a fixed outdoor air fraction of about twelve percent. The compressor capacity modulated to maintain the DSP and the indoor setpoints.

During the field testing, the field team set the Daikin to a more standard Zone Set Point control approach. Field testing included a period of constant supply fan operation and a period of continuously variable supply fan operation.

Figure 6. Outdoor Controller for the Daikin McQuay Rebel

AAON

The research team ordered the AAON RQ Series RTU for eventual installation in the Nampa Fred Meyer. Given these RTUs do not have factory-default controls, Fred Meyer works with Wytek Controls to retrofit AAON units with custom outside controllers compliant with the Fred Meyer building automation system standard. Figure 7 shows the outdoor controller used for this test unit. A single temperature sensor monitored conditions in the zone. The outdoor controller communicated with a building automation system (BAS) control unit, an Emerson E2 system, which controls multiple RTUs and allows detailed changes to the outdoor control system parameters as well as to the indoor schedule, setpoints, deadband, and minimum damper position.

Researchers set the maximum fan speed through a single manually-controlled variable speed drive (VSD), which allows the AAON product line to use the same supply fan for many different-sized models. The E2 allows control that sets fan speeds (as the percentage of maximum) for different modes of operation (ventilation only, economizer, DX cooling). The ventilation rate of air varied from 1,240 CFM to 840 CFM during three modes of operation, with an approximate five percent outdoor air fraction for each mode. The scroll compressor responds to requests from the controller to modulate using its unloading capacity modulation.

Figure 7. AAON RQ Outdoor Controls and Refrigeration Panel

2.2.2.3. <u>NBIL Test Control Settings Details</u>

NBI held the fan and setpoint schedules constant for all units shown in Table 3. This defines the hours that the fan ran to satisfy minimum outside air requirements, regardless of whether a call for cooling was present. The fan power in this mode differed from unit to unit.

NBI held the induced outdoor air (OA), as a percentage of supply air at the minimum outside airflow rate, constant for all units at approximately fifteen percent. This minimum outside air mass flow per unit time applied during all operating modes except “Off.” A unit may deliver flows above this minimum outside airflow in other modes such as the economizer mode, or second stage compressor, as a consequence of the unit control strategy.

As described above, NBI calibrated the outside air fraction for each unit using a dry-bulb temperature mixing comparison.

Researchers attempted to leave the onboard control logic of each RTU in its “out-of-box” mode, aside from changes necessary to meet the test conditions and selection of the proper control mode. The units with advanced capabilities (such as fan or compressor modulation, higher-order temperature optimization, or adjustable gains that tune these processes) operated per manufacturer programming to provide the energy performance intended.

Table 3. Controls Summary – NBIL Testing

<table>
<tr><th>RTU</th><th>Indoor Control Type</th><th>Setpoints Occupied/Unoccupied /Deadband [°F]</th><th>Occupied Schedule</th><th>Target OA Fraction [%]</th></tr>
<tr><td>Trane</td><td>Standard five-pin thermostat indoor controller.</td><td rowspan="3">73 / 85 / 1</td><td rowspan="3">8:00 a.m. – 10:00 p.m.</td><td rowspan="3">15%</td></tr>
<tr><td>Daikin</td><td>Temperature probe in test space. Conditions controlled by outdoor controller with manual user interface.</td></tr>
<tr><td>AAON</td><td>Temperature probe in test space. Conditions controlled by outdoor controller overseen by Emerson E2.</td></tr>
</table>

During the course of testing, researchers changed some details of the control settings to examine the impact on the energy signature and to facilitate tuning of the Physical Model.

Table 4 summarizes specific control settings.

Table 4. RTU Lab Control Settings Details

<table>
<tr><th>RTU</th><th>Control Settings Label</th><th>Description</th></tr>
<tr><td>Trane</td><td>T-1</td><td>Factory default single fan speed for ventilation, compressor, and economizer modes. Economizer did not operate due to unknown error.</td></tr>
<tr><td>Daikin</td><td>DM-1</td><td>Discharge Set Point (DSP) control mode used with supply air dry-bulb temperature held at 55 degrees F. Single fan speed for ventilation, economizer, and compressor, but continuously varying inverter-based compressor. Economizer operated with integrated compressor operation.</td></tr>
<tr><td rowspan="3">AAON</td><td>A-1</td><td>Factory default single fan speed for ventilation, compressor, and economizer modes. The economizer did not initially operate due to unknown error (later determined to be an improperly-set economizer lockout temperature).</td></tr>
<tr><td>A-2</td><td>Service contractor modification to explore modally-varying constant fan speeds. The fan was configured with two-speed control, running at a lower speed for ventilation mode, and a very high speed during compressor and economizer modes. Schedule and setpoints were unchanged. The economizer still did not operate.</td></tr>
<tr><td>A-3</td><td>Additional service contractor modification to explore changes to fan speed modes. The fan speeds during compressor and economizer modes were reduced. The economizer lockout was adjusted and the economizer was enabled, which was confirmed by data inspection.</td></tr>
</table>

2.2.2.4. Lab Testing: Schedule and Test Runs

NBI conducted daily tests in runs of multiple days at the same control settings and sensible loading, with continuous collection at one-minute intervals. Table 5 shows the datasets and control setting configurations.

Table 5. NBIL Testing Timeline

RTU	Start Date	End Date	# of Days	Loading Condition	Control Settings
Trane	7/9/2012	7/19/2012	11	Low	T-1
	7/20/2012	8/1/2012	13	High	T-1
	8/2/2012	8/5/2012	4	Ultra-High	T-1
Daikin	8/10/2012	8/20/2012	11	High	DM-1
	8/21/2012	8/28/2012	8	Ultra-High	DM-1
AAON	8/30/2012	9/4/2012	6	High	A-1
	9/6/2012	9/14/2012	9	High	A-2
	9/15/2012	9/19/2012	5	Ultra-High	A-2
	9/21/2012	9/30/2012	10	Ultra-High	A-3

2.2.2.5. Lab Testing: Instrumentation and Data Collection

The NBIL data acquisition system collected the data points presented in Table 6 and Table 8 at one-minute intervals throughout the test period.

Table 6. Airside Measurement Points – Lab Testing

Point Name	Measured Values	Measurement Interval
Supply Air	T_{db}, T_{dp}, RH	1 minute
Outdoor Air	T_{db}, T_{dp}, RH	
Return Air	T_{db}, T_{dp}, RH	
Mixed Air 1	T_{db}	
Mixed Air 2	T_{db}	
Mixed Air 3	T_{db}	
Mixed Air 4	T_{db}, T_{dp}, RH	
Supply Air Duct Differential Pressure	air pressure (Pascals) calibrated flow rate/CFM	
Indoor Room Air	T_{db}, T_{dp}, RH	

Note: Refer to Appendix C – Definitions of Key Terms for Measured Values definition

An Onset HOBO® U30 data logger with Internet connectivity collected air-side sensor data. Sensor types are detailed in Table 7 below.

Table 7. Airside Sensor List – Lab Testing

Airside Sensors	Description	Accuracy/Range
12-bit Temperature	High accuracy sensor; duct mounted where applicable	±0.2°C/±0.36°F or better Range: 40°C to 100°C (-40°F to 212°F)
12-bit Temperature and RH	High accuracy sensors, relative	±0.2°C/±0.36° F,

	humidity, dew point	RH: +/- 2.5% Range: -40°C to 75°C (40°F to 167°F) at 0 - 100% RH
Differential Pressure Sensor	Duct mounted	+/- 1% full scale Range: 0 – 500 Pascal (Pa)

NBI calibrated airstream flow rates using Energy Conservatory TrueFlow® plates to correlate duct differential pressure to airflow measured by the plates. The resulting relationship between airflow (CFM) and duct differential pressure (p) in Pascal is shown below:

$$CFM = 111.74 * p^{0.521} \ \ with \ \ R^2 = 0.9992$$

Appendix D – Flow Calibration Curves shows the plotted data for this relationship.

NBI specified a DENT PowerScout 18 for collecting power and energy measurements. Table 8 shows the points monitored during lab testing and Table 9 summarizes the power side sensors.

Table 8. Power Side Measurement Points – Lab Testing

Point Name	Point Value	Measurement Interval
RTU Power	Average 3-phase true power in the interval	1 minute
RTU Energy	Average 3-phase energy in the interval	
RTU Power Factor	Power Factor in the interval	

Table 9. Power Side Sensor List – Lab Testing

Sensor	Description	Accuracy
30 Amp Current Transformers	Continental Control Systems - CTT Series, solid-core, high accuracy, Toroidal current transformers	±1% from 10% - 130% of range

Researchers verified the measurement accuracy of the 30A current transformers at low currents using independent measurement equipment with high accuracy 5A current transformers.

2.2.2.6. Lab Testing: Additional Instrumentation -- ClimaCheck

ClimaCheck installed instrumentation on the refrigerant side of the RTU to examine the operating characteristics (detailed in the report in Appendix B) of the vapor-compression cycle at one-minute intervals. The system also measures compressor power and uses a proprietary method to calculate the compressor coefficient of performance (COP) among other refrigeration cycle parameters. This report occasionally includes relevant data from the ClimaCheck instrumentation to supplement the analysis.

2.3. Field Test Protocol

IDL research staff used a reduced set of instrumentations in the field. The sensible and latent loads in the field were unknown independent variables; the setpoints and occupied/unoccupied schedules, fan run time, and temperature setpoints were known, as was the approximate outdoor air fraction shown in Table 10.

Table 10. General Daikin Field Settings

Days of Testing	Setpoints Occupied/Unoccupied/Deadband [°F]	Schedule	Approximate Outside Air Fraction
32	73/76/1	7:30 to 23:00	40%

2.3.1. Field Testing: System Configuration and Controls

NBI selected two modes of control for the field testing, as Table 11 shows.

Table 11. Daikin Field Testing Control Settings

Control Settings Label	Fan Control	Economizer Active	Minimum Fan Power (kW)	Standby Power (kW)	Economizer Fan Power (kW)
ID-1	Constant Speed – 24-hour schedule	Yes	0.13	0.14	Not available
ID-2	Variable Speed – 24-hour schedule	Yes	0.23	0.12	0.35

Table 12 shows the timeline of field testing.

Table 12. Field Testing Timeline

Start Date	End Date	# of Days	RTU	Loading Condition	Control Settings
9/15/2012	10/10/2012	26	Daikin	As Found	ID-1
10/11/2012	10/16/2012	6	Daikin	As Found	ID-2

2.3.2. Field Testing: Instrumentation and Data Collection

NBI specified the field instrumentation and data collection systems, which were installed and calibrated by the Integrated Design Lab. Table 13 shows the data points.

Researchers conducted airside data acquisition using an Onset HOBO U30 data logger with Internet connectivity. All sensors are Onset smart sensors sampled at one-minute intervals. Table 7 shows the sensor specifications.

Table 13. Airside Measurement Points – Field Testing

Point Name	Measured Values	Measurement Interval
Supply Air	T_{db}, T_{dp}, RH	1 minute

Outdoor Air	T_{db}, T_{dp}, RH
Return Air	T_{db}, T_{dp}, RH
Mixed Air 1	T_{db}
Mixed Air 2	T_{db}
Mixed Air 3	T_{db}
Mixed Air 4	T_{db}, T_{dp}, RH

Since the team field-tested only the Daikin McQuay unit, IDL personnel added a power meter on the supply fan that was calibrated with the flow rate to better understand the dynamics during test run ID-2. Test run ID-2 used a continuously variable fan (as opposed to a control mode in which different constant speeds are used). Table 14 shows the power and energy measurement points.

Table 14. Power Side Measurement Points – Field Testing

Point Name	**Point Value**	**Measurement Interval**
RTU Power	Average three-phase true power in the interval	1 minute
RTU Energy	Average three-phase energy in the interval	
RTU Power Factor	Power Factor in the interval	
Supply Fan Power	Average three-phase true power in the interval	
Supply Fan Energy	Average three-phase energy in the interval	
Supply Fan Power Factor	Power Factor in the interval	

IDL staff collected field data for power and energy measurements using the U30 data logger with Continental Control Systems WattNode power transformers. WattNode accuracy is published at 1.25% of reading given line voltage of -20% to +15% of nominal, measured current between 5%-100% of current transformer full scale, and possible high ambient temperatures, up to 130 degrees F.

Field site metering included separate metering of fan power; this permitted an assessment of the role of the supply fan in the overall RTU operation and allowed for a cross-check of the standby power measurement. Since the field site had fixed ductwork (with no automatic terminal dampers), IDL used Energy Conservatory TrueFlow plates to calibrate airflow rate (CFM) to fan power (kW_F), as shown below:

$$CFM = 1372.8 * KW_F^{\;0.3791}$$

Note that the field site was located at an elevation of 2,500 feet, which reduced the air density from approximately 15 CF/lb to approximately 13.5 CF/lb. Researchers adjusted the calibration equation and all airside calculations to account for the density change.

2.4. Analysis Methods

2.4.1. Data Analysis

NBI's analysis draws from the RTF protocol for Measurement of Fan and Cooling Savings from Commercial-Sector Packaged and Split System HVAC Units (Regional Technical Forum 2012). The RTF protocol establishes a methodology for predicting annual electric energy use and savings for an upgrade to an RTU, based on a short period of measurement using a combination of daily data and one-minute interval data.

NBI did not use the full RTF protocol due to insufficient test days to meet its requirements. However, the RTF protocol is based on using daily energy use correlated to average outdoor dry-bulb air temperature, called the "energy signature" – and NBI used the energy signature to examine a projection of cooling season energy usage and to form the basis for the initial working model.

The energy signature method is limited in that it is valid for each RTU only under the same highly-specific conditions, thus necessitating a model with the flexibility to modify the energy signature for an RTU to account for changes in conditions.

2.4.1.1. Daily Energy Signature

Researchers first developed a daily energy signature, as illustrated in Figure 8, to analyze the energy use of each unit. The energy signature shows the total daily energy use in kWh/day plotted against average daily (twenty-four-hour) outdoor dry bulb air temperature. Measured data points and a linear trend line are shown for each loading condition. Each dataset yields a clear and expected pattern; as loading increases, so does daily energy use. The slope of each trend line is a result of the temperature response of the RTU as well as the response of the thermal zone to increased thermal loading. Increasing temperatures increase the thermal load in the test zone, which in turn requires the RTU to provide more cooling and use more energy. The increasing temperature also reduces the condenser efficiency of the RTU, which further increases energy use.

For each loading condition, the fitted trend line shows three key elements of each RTU's performance:

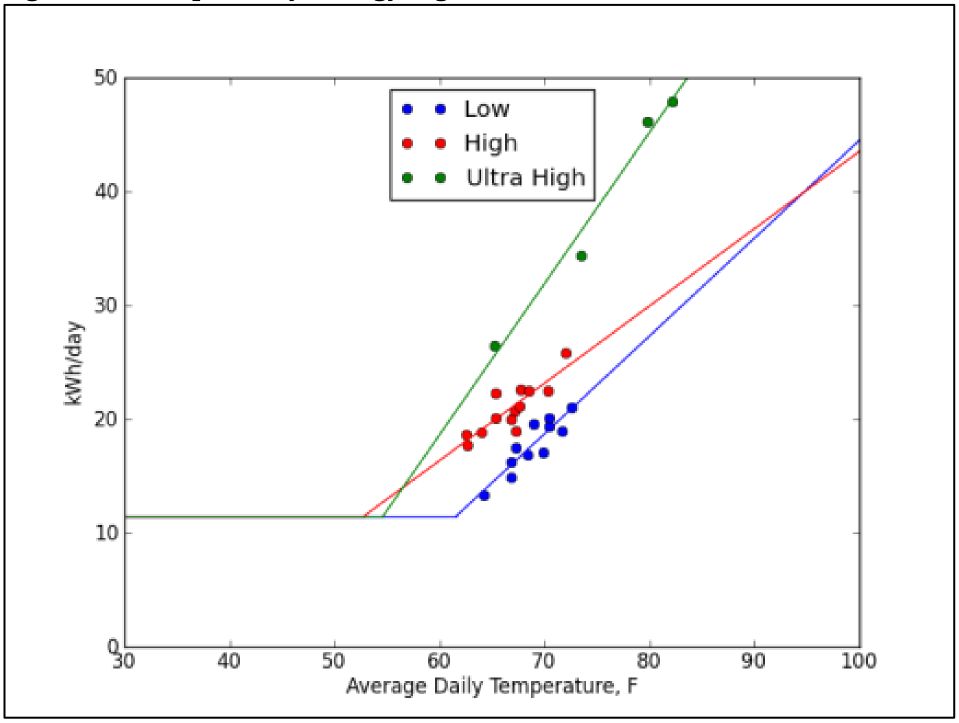
Base Load: This is the flat portion of the line (around 12 kWh/day in Figure 8) representing the daily energy used for ventilation and standby power (such as for transformers or other power electronics in the RTU).

Balance Point: This is the temperature at which the flat line ends and the sloped line begins. Below this temperature no cooling load exists, and daily energy use equals the base load kWh/day. As the temperature increases above the balance point, additional energy is needed for cooling. The balance point of a given RTU relates to economizer functionality, temperature setpoint, scheduling, and the level of thermal gain in a space. For example, if the economizer is functional, the balance point will move to the right of its expected position under the condition of the compressor providing all cooling.

<u>Slope</u>: The sloped line is an expression of how much additional energy is needed in response to an increase in daily average outdoor temperature. This provides some insight into the efficiency of a unit's refrigeration loop. However, the overall efficiency is impacted by base load and balance point as well.

These parameters form a basic representation of the energy response versus outdoor air temperature. Combining them with Typical Meteorological Year (TMY) data for May 1 to October 31 allows projection of the cooling season energy usage discussed below.

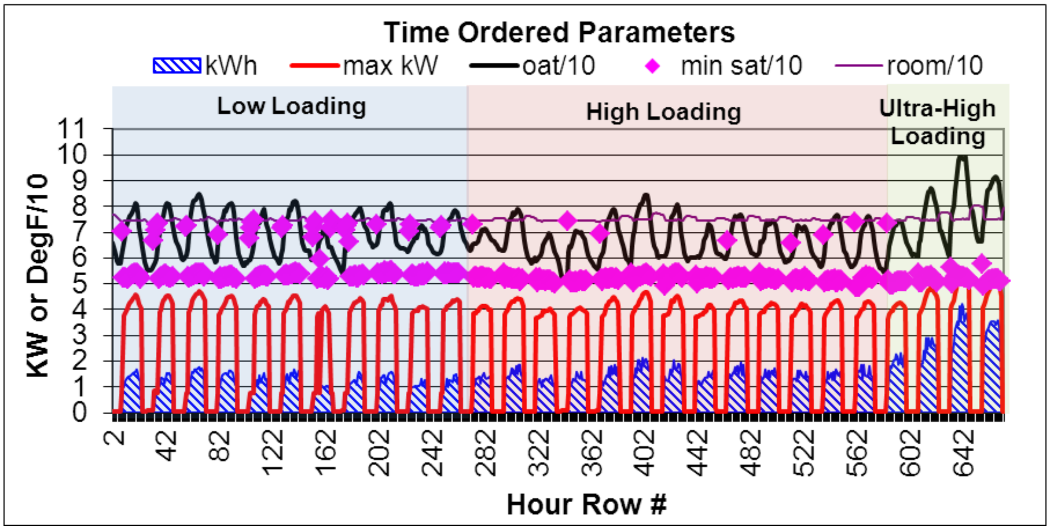
Figure 8. Example Daily Energy Signature



2.4.1.2. <u>Hourly Time Series</u>

The hourly time series shows the general day-to-day trend of an RTU, with an example shown in Figure 9. This resolution generally does not permit the viewing of individual compressor cycles, but rather only the aggregate impact of numerous cycles during an hour. This time series shows overnight performance, which can be an indicator of standby power or fan-only power. Inspection of the gap between the maximum kW and the average kWh during an hour indicates the loading of the RTU, assuming that it modulates output by cycling its compressor (as opposed to an advanced unit that can modulate compressor output). This plot also shows extended periods of economizer operation, indicated by the periods when the minimum supply air temperature is equal to the outdoor air temperature.

Figure 9. Example of an Hourly Time Series



Notes: The air temperature points are divided by 10 so that the points fit on a single Y-axis. "OAT" represents outdoor air temperature; "min SAT" is the lowest supply air dry bulb temperature seen in the hour; and "room" is the test space dry-bulb air temperature.

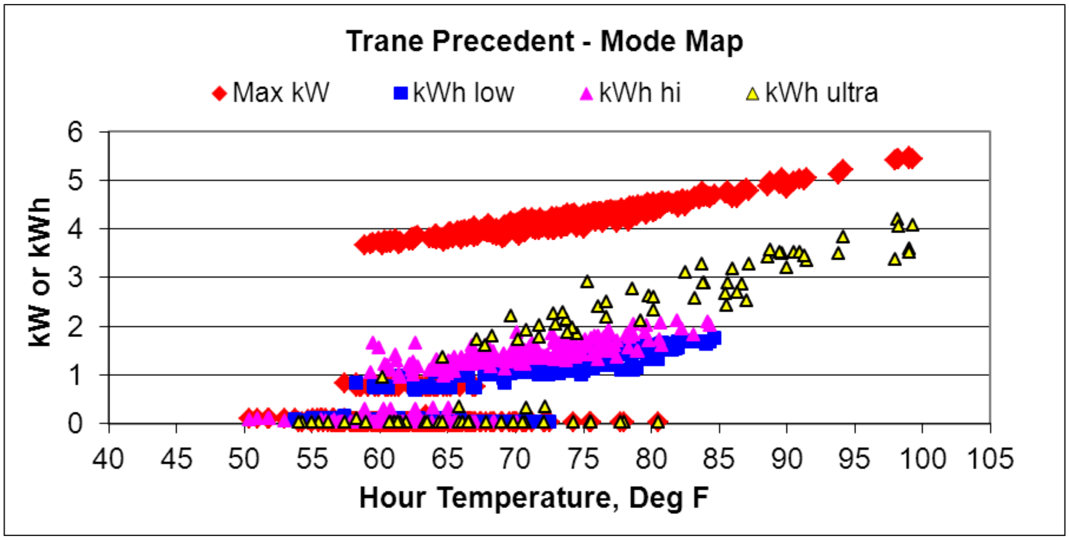
2.4.1.3. Hourly Mode Map

The hourly mode map, shown in Figure 10, better demonstrates the loading of each RTU. The mode map shows the hourly energy use (kWh) and hourly maximum power (kW) versus average hourly temperature. In Figure 10, the trend in maximum power indicates that despite loading conditions, the maximum power always falls on the same line. This line represents the power used when the compressor is running at full speed, the only option for a traditional RTU that cycles its compressor on and off in response to loading.

The Figure 10 example illustrates that most hours above 60 degrees F have at least a small amount of compressor operation. As the temperature increases, an increase in duty cycle is evident in the trend of gradually-increasing kWh with temperature. Increasing temperature also increases maximum power; the closer the hourly kWh is to the maximum hourly kW, the closer the unit is to running at a 100% duty cycle for that hour. The slope of the red "Max kW" points below is related to the efficiency of the condenser at rejecting heat, which in turn corresponds to a higher operating temperature at the compressor and to a greater power demand.

The mode map is well-suited for understanding the different operating modes of a traditional RTU that uses a duty cycle to meet cooling demand. RTUs that modulate compressor output in response to cooling demand will display a less regular pattern.

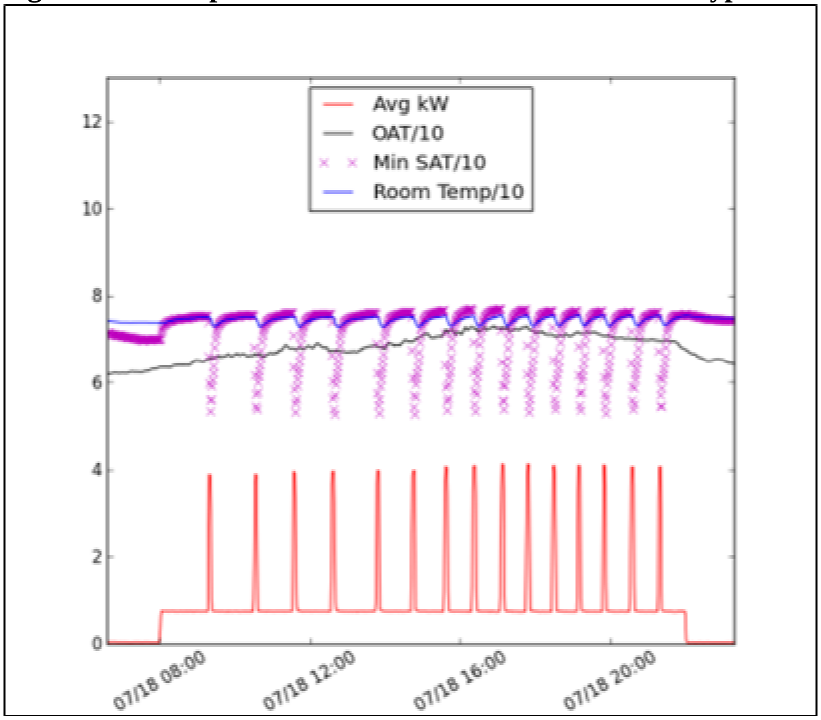
Figure 10. Example of an Hourly Mode Map



2.4.1.4. *Detailed Time Series*

Analysis also included detailed one-minute time series data for a given unit to investigate the behavior during each cycle of the compressor, or during each diurnal cycle. The example in Figure 11 shows data for the RTU average power, outdoor air temperature, minimum supply air temperature, and room temperature. The detailed level of data facilitates understanding of the operating mode and economizer functionality of each unit and infers measurements of standby power loss and fan power in different modes of operation.

Figure 11. Example of a One-Minute Time Series for a Typical Day



Note: The air temperature points are divided by 10 so that the points fit on a single Y-axis.

2.4.1.5. Normalized Cooling Season Energy

As described above, the researchers used the three elements of the energy signature to estimate normalized cooling season energy consumption. The researchers used the signature, combined with average air temperature data for May 1 to October 31, to estimate the total cooling season energy use. Researchers used this date range to capture the hottest summer months along with the mild shoulder seasons expected in most Northwest cities. Since researchers conducted no testing in heating mode, an estimate of wintertime energy use – and therefore annual energy use – is not included.

Researchers calculated the cooling season energy use twice for each unit; the first estimate assumed a seven-day schedule where every day of the year has the same 8:00 a.m. to 10:00 p.m. occupancy, and the second estimate assumed a five-day weekday schedule of 8:00 a.m. to 10:00 p.m. with no operation on weekends. This approach assumes that many buildings, especially offices, will likely fall between the two scenarios by using a significant weekend setback that allows some RTU operation, but reduces usage below weekday levels.

Each estimate also identifies the split between “cooling energy” – the energy use associated with the compressor and balance of plant operation excluding the supply fan, and “base load energy” – the energy used by the supply fan to provide ventilation air during the entire occupied schedule throughout the year (not just during the cooling season).

2.4.2. RTU Physical Model

The energy signature analysis and projection of cooling energy use, and therefore savings, is inherently limited by the specific nature of the response of each unit to the particular testing conditions. NBI sought to use an energy model under which the parameters of operation could be varied depending on the application to tune the model to a set of data points. This model then allows the projection of annual energy use for each unit, as well as a comparison to the annual energy use projected for other RTUs. This allows researchers a new way to compare energy projections of advanced performance RTUs against the code level units for the same conditions.

Appendix A provides details of the model. Subsequent sections will discuss in more detail the model structure and analysis.

Testing the three units at the NBIL under a structured sequence of known loading conditions led to a model of RTU energy use that is responsive to the most significant of the site conditions. NBI derived the resulting Physical Model by fitting the average day energy signature metered at the NBIL to an analytical model.

The calibrated model enables prediction of the performance of both the advanced and the code units anywhere in the Northwest region under a range of loading conditions.

3. Findings

3.1. Lab Test Results

This section summarizes the results from testing at the NBIL. Table 15 outlines the general observed data on the operating conditions, including specific control settings.

Table 15. Observed Data from NBIL Testing

	Trane – T1	Daikin – DM1	AAON – A1	AAON – A2	AAON – A3
Standby power, (kW)	0.35	0.145	0.13	0.13	0.13
Fan power above standby – Ventilation mode (kW)	0.75	0.13	0.5	0.17	0.17
Fan schedule (hours)	14	14	14	14	14
Percent outside air (%)	12%	10%	5%	5%	5%
Airflow– Ventilation mode (CFM)	1,588	840	1,237	825	840
Airflow (lb./min)	120	62	93	62	63
Economizer Operational?	No	Yes	No	No	Yes

3.1.1. Trane

Figure 12 shows the energy signature for the Trane unit, clearly illustrating the pattern of increasing energy use with increasing loading. The linear regressions suffered from the small number of data points and the lack of a wide band of average daily temperature days.

The linear nature of the points as they approach the minimum fan energy horizontal indicated that the economizer may have not been functional, and researchers confirmed this by observation and additional temperature data.

Figure 12. Trane Energy Signature

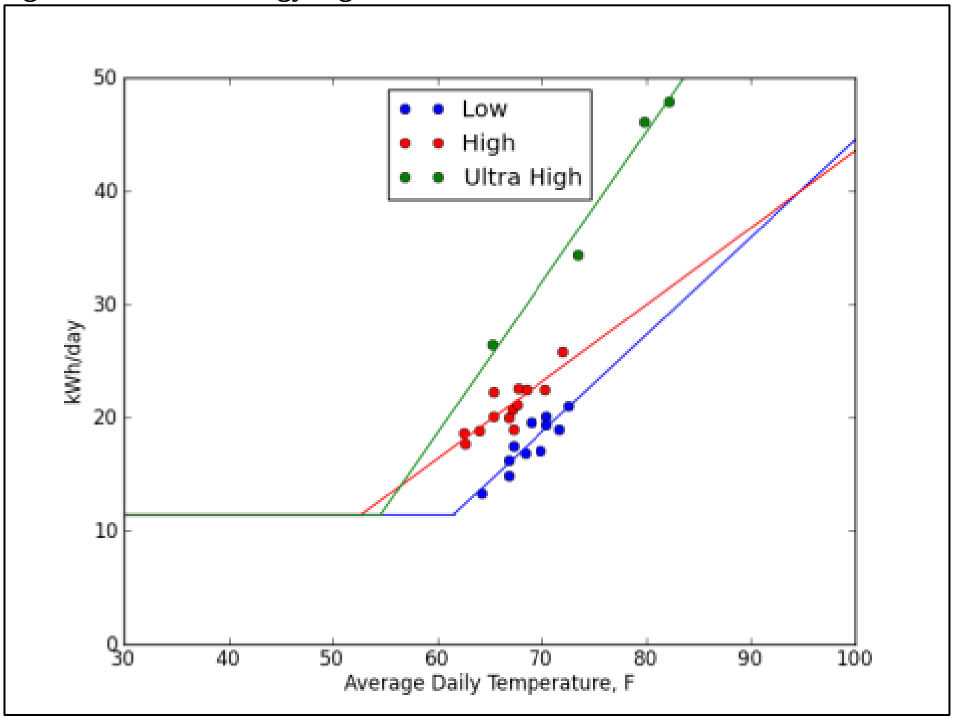


Table 16 shows the resulting energy signature parameters.

Table 16. Trane Energy Signature Parameters

Testing Regime	Slope [kWh/day*F]	Base load [kWh/day]	Balance Temp [°F]	R^2	Sample Size
Low T1	0.86	11.46	61.6	0.80	11
High T1	0.68	11.46	52.8	0.72	13
Ultra-High T1	1.33	11.46	54.6	0.98	4

Figure 13 shows the hourly time series for the Trane unit. Data from this plot confirms that the economizer was not operating when outdoor conditions met the published criteria for economizer operation.

Figure 13. Trane Hourly Time Series

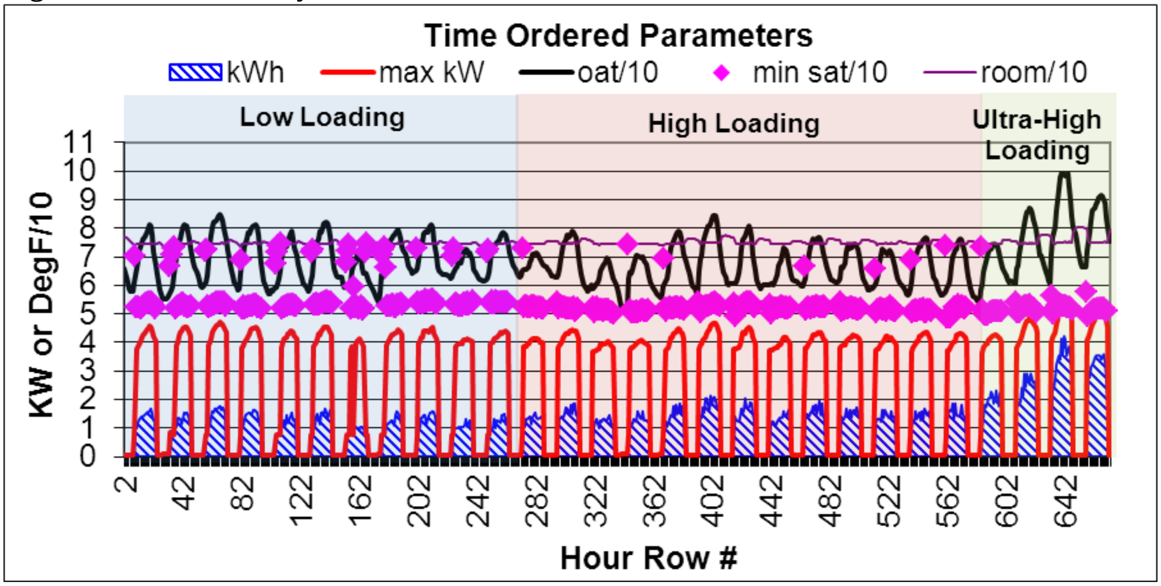
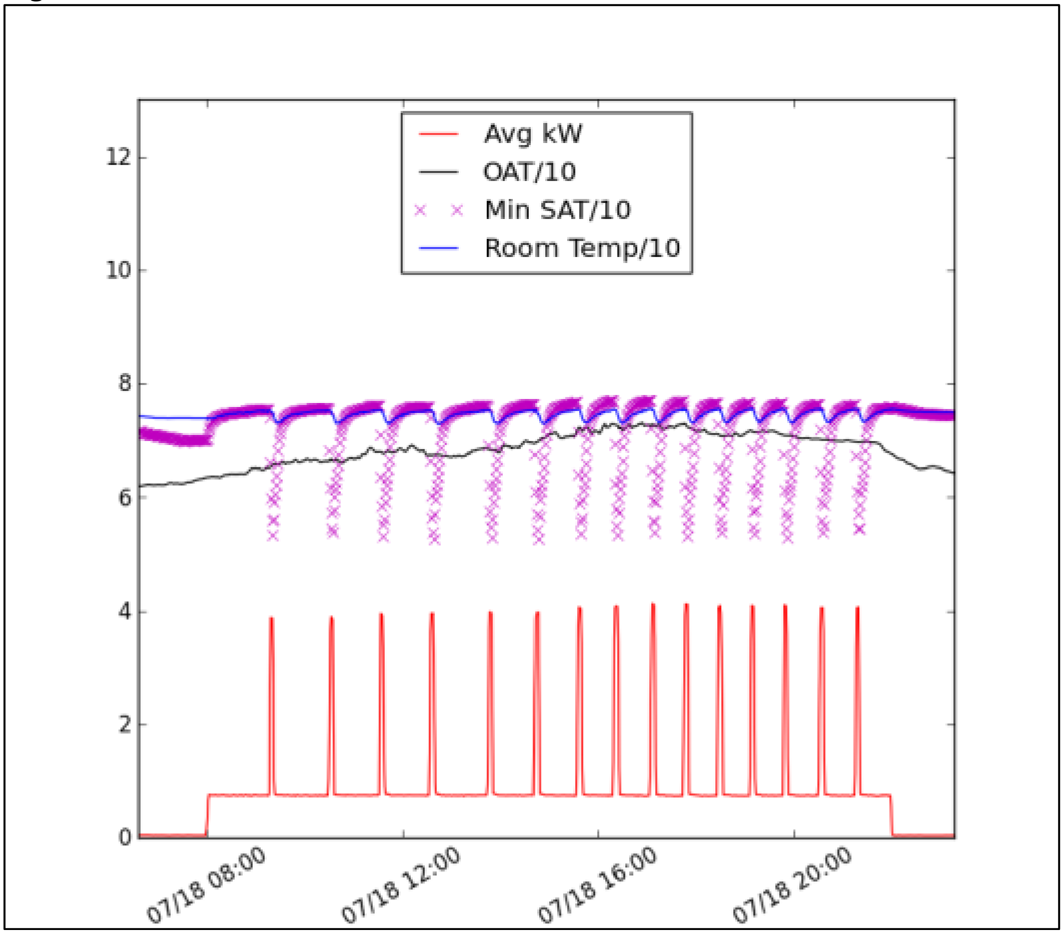


Figure 14 shows a Trane one-minute time series for a typical day with some parameters divided by 10 so they can be presented on the same Y-axis. The data show that the fan comes on as scheduled at 8:00 a.m. with a power draw of approximately 0.75kW. The fan runs all day until 10:00 p.m. (end of occupied period), with intermittent compressor cycles as dictated by the internal and thermal loading. A typical cycle for this unit ranges from approximately five to fifteen minutes. Although the unit was equipped with an economizer, the data indicates it was not functioning as expected. For example, during the first compressor cycle shown in Figure 14, the outdoor air temperature (OAT) was around 65 degrees F. With a functional economizer, the minimum supply air temperature (SAT) during this period would be expected to closely match the OAT, and the compressor would not be running. Instead, the Trane turned on the compressor and the minimum SAT was significantly below the OAT, approaching 50 degrees F.

NBI tested the economizer sensor with a blast of very cold air, which caused it to actuate. This test demonstrated that the economizer was functional, but that it did not operate as expected for an unknown reason.

Figure 14. Trane One-Minute Time Series



The hourly mode map in Figure 15 shows three distinct modes of operation: standby, fan only, and fan plus compressor. The periods of fan and compressor operation fall into the middle range of the chart and show the expected trend of increased hourly kWh as loading increases from Low to Ultra-High. The chart shows a small number of fan-only hours at 0.75 kWh. Standby hours are clustered just above zero, at 0.04 kW.

Figure 15. Trane Hourly Mode Map

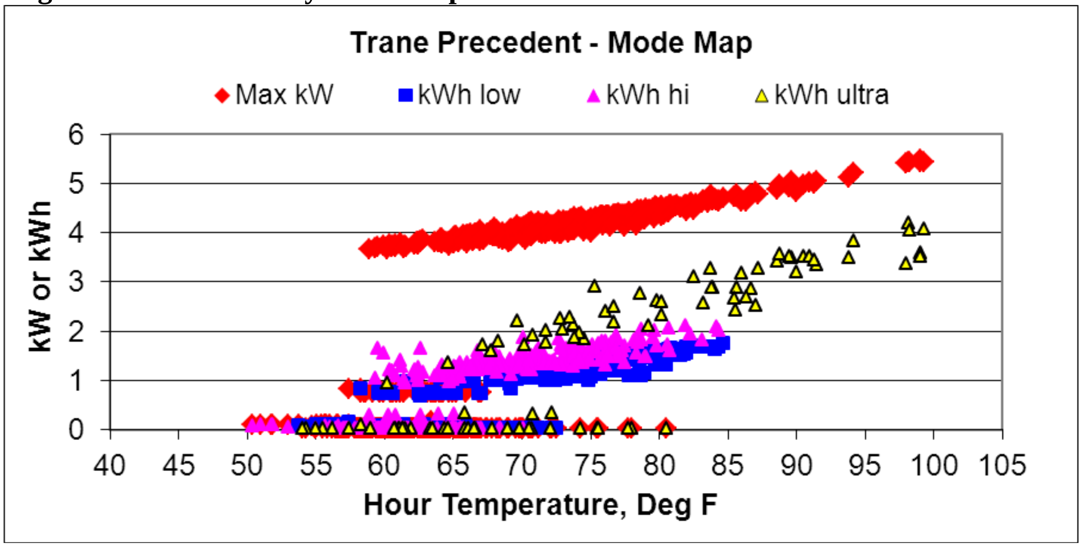


Table 17 summarizes the normalized cooling season energy use under each set of loading conditions for seven-day and five-day per week schedules.

Table 17. Trane Cooling Season Energy Use Summary

Loading/Control	Seven-Day (kWh)			Five-Day (kWh)		
	Total	Base	Cooling	Total	Base	Cooling
Low T1	2,644	2,109	535	1,912	1,501	411
High T1	3,340	2,109	1,231	2,388	1,501	887
Ultra-High T1	4,130	2,109	2,022	2,955	1,501	1,454

3.1.2. Daikin

Researchers tested the Daikin unit for a total of nineteen days, subjecting it to High and Ultra-High loading, as previously defined in Figure 5.

Figure 16 shows the daily energy signature for the Daikin. Unfortunately, the Ultra-High testing period did not include the same range of outdoor air temperatures experienced during the High period. The comparison between 65 and 70 degrees shows the anticipated increase in daily usage as the loading was present, but the separation between the two sets of data points appears very modest.

Figure 16. Daikin Energy Signature

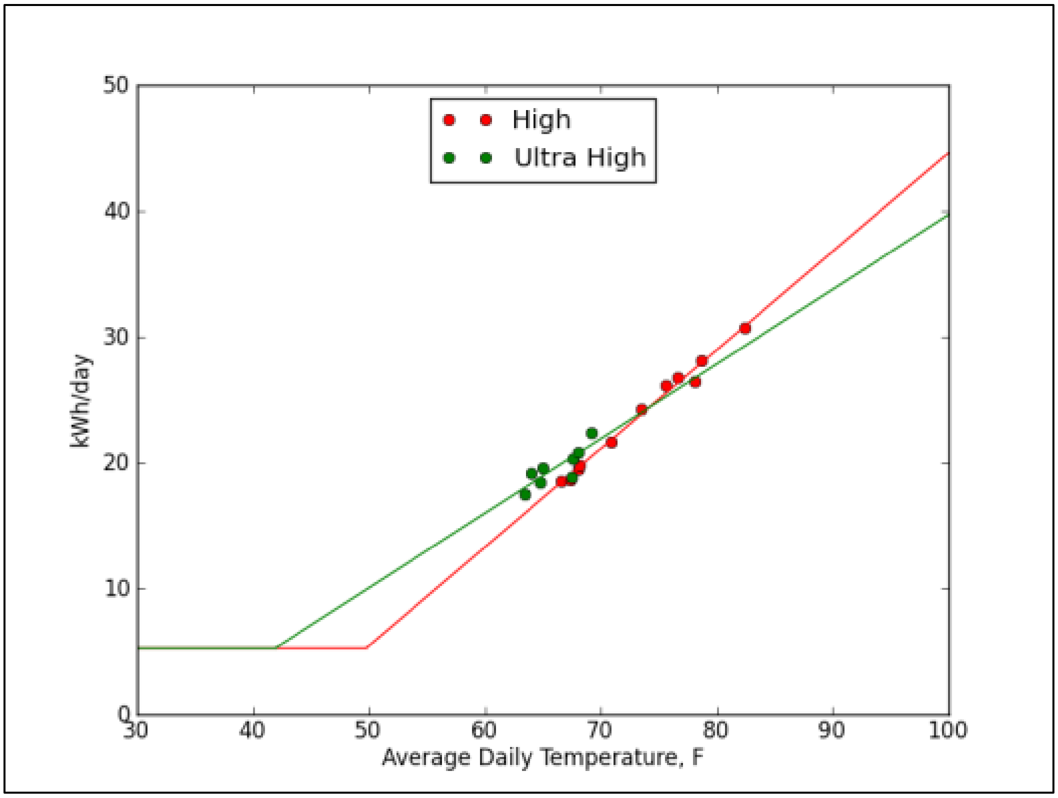


Table 18 shows the resulting energy signature parameters.

Table 18. Daikin Energy Signature Parameters

Loading/Control	Slope	Base load	Balance Temperature	R^2	Sample Size
High DM1	0.78	5.28	49.8	0.99	11
Ultra-High DM1	0.59	5.28	41.9	0.69	8

Figure 17 shows the hourly time series. The Daikin's operation is notably different from the Trane in that no cycling takes place. The unit turns on at 8:00 a.m. when the daytime setpoint takes effect and the internal loads begin to ramp up, and operation is relatively steady throughout the day. As the thermal loads increase, the average power also increases slightly. Although the power increases in response to loading, the SAT does not change.

Figure 17. Daikin Hourly Time Series

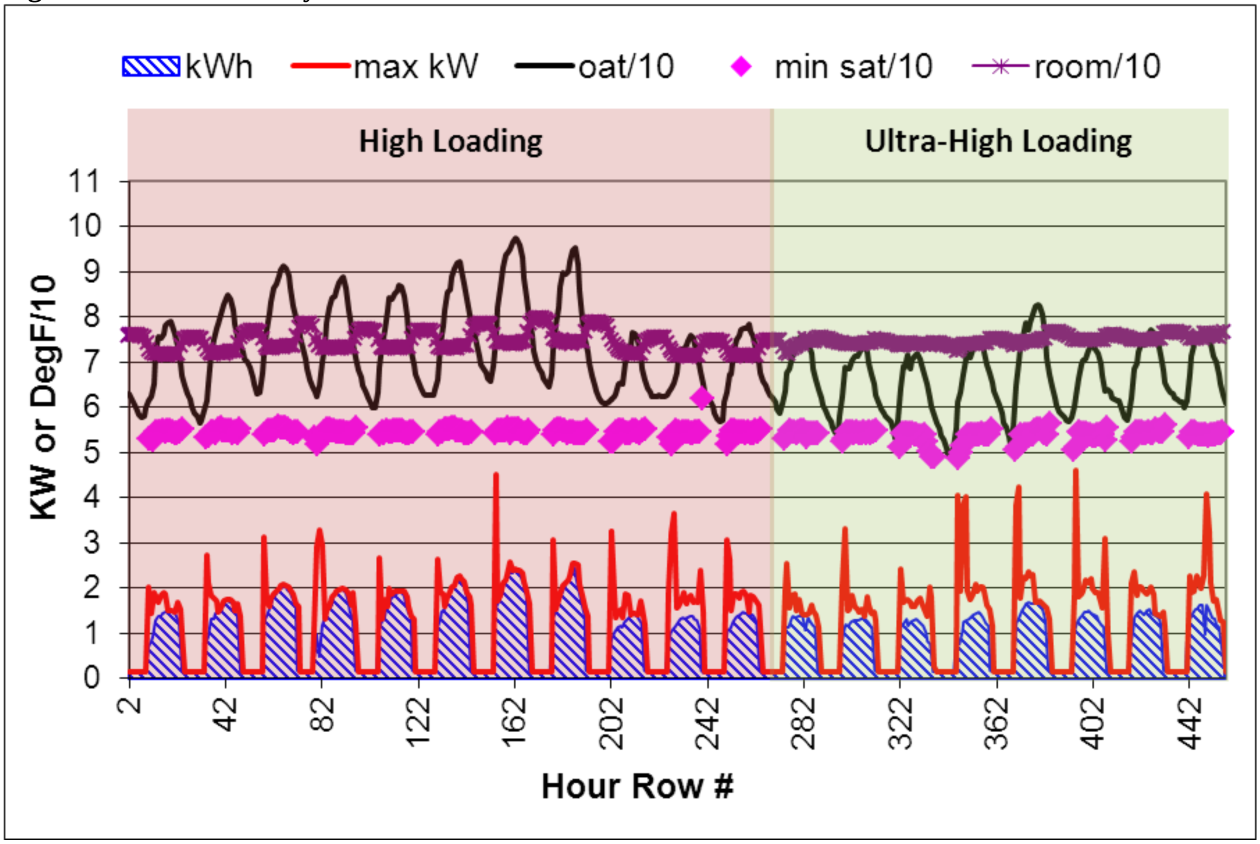


Figure 18 shows the one-minute time series for the Daikin unit. The modulation of the compressor is continuous throughout the cycle.

Figure 18. Daikin One-Minute Time Series for High Loading

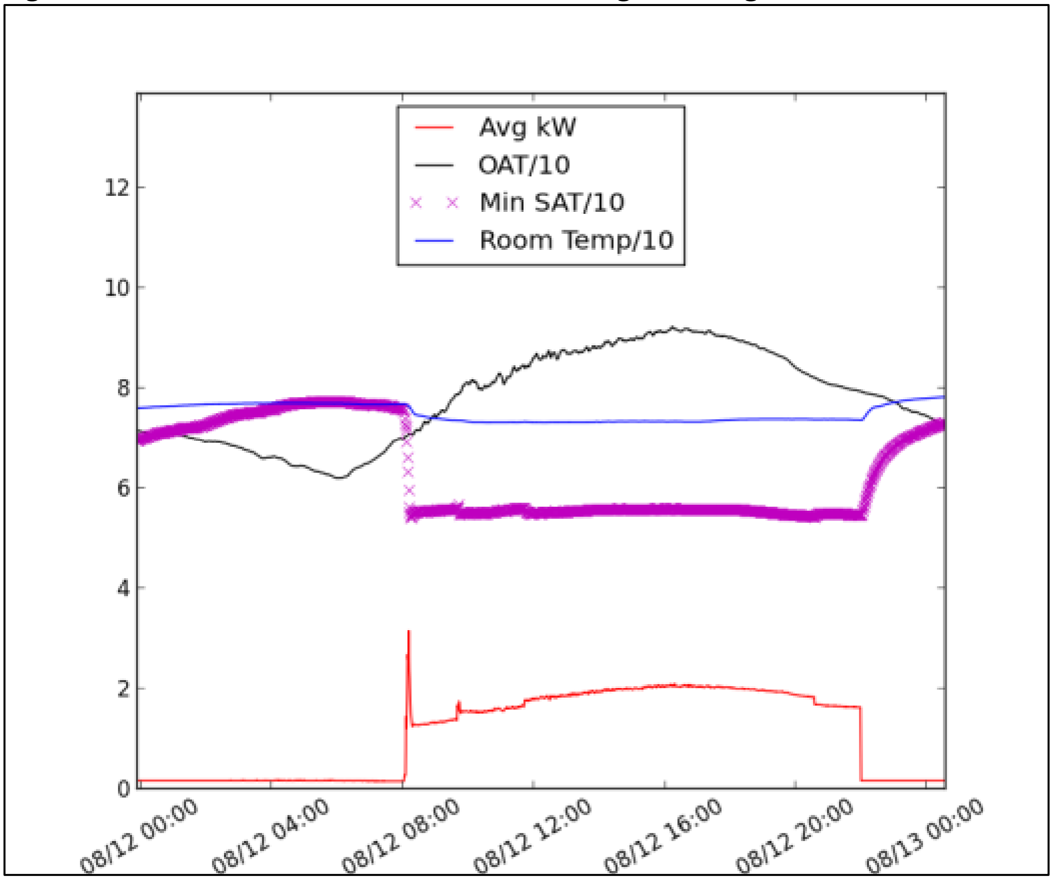


Figure 19 shows the mode map for the Daikin unit. Due to the highly modulated output of the inverter scroll compressor and the constant speed fan, the relationship between maximum kW and kWh is quite different from that of a typical RTU. Most RTUs exhibit a linear relationship between the maximum kW and OAT, with the hourly kWh forming a cloud of points beneath the slope; however, data in Figure 19 show a very consistent and linear relationship between the hourly kWh and OAT. The maximum kW shows an unusual trend; it is higher at lower temperatures. NBI believes this is the result of brief power spikes that tend to occur at the beginning of each day's loading cycle, but may sometimes occur mid-day. Subsequent discussions with Daikin McQuay engineers suggested that this short spike resulted from the oil management system in the compressor that runs on high speed to separate the oil from refrigerant in the system.

Figure 20 shows additional one-minute time series data on a day when the economizer on the Daikin was operating. It shows that the economizer and compressor are operating simultaneously and were likely employed to maintain the 55 degrees F SAT setpoint.

Figure 19. Daikin Hourly Mode Map

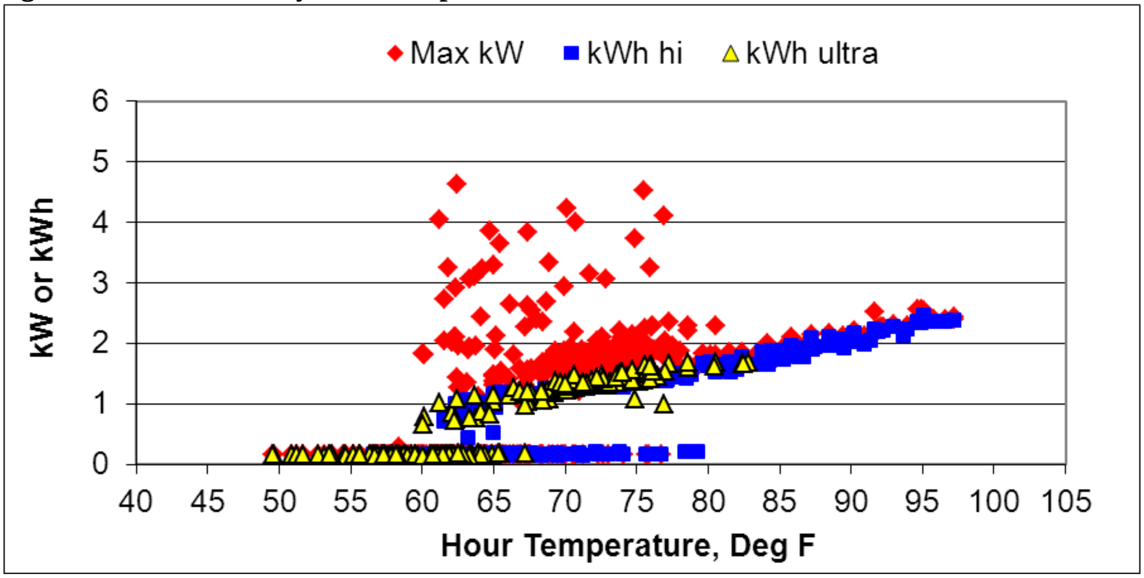
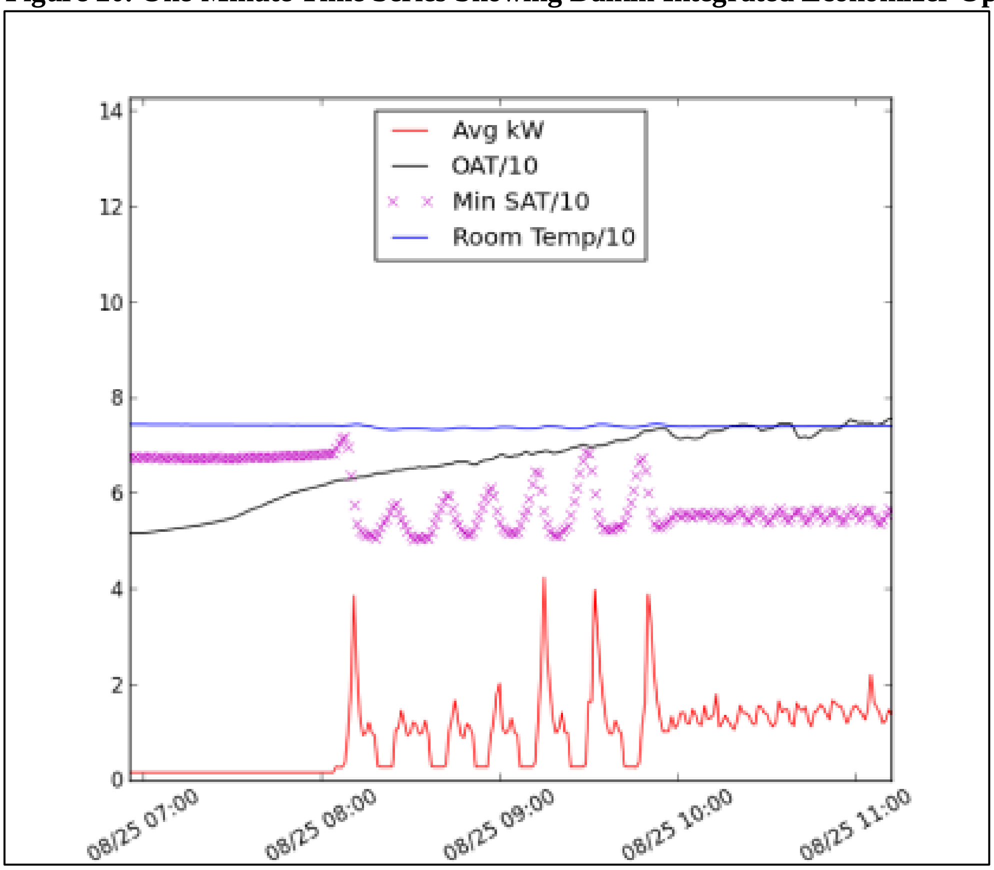


Figure 20. One-Minute Time Series Showing Daikin Integrated Economizer Operation



The argument for using the compressor during economizer cycles is rooted in the fan power laws. During the economizer interval, the airflow was low because the fan power was only about 120 W. At this low airflow, economizer operation is inherently limited. The RTU could have achieved more economizer-only cooling simply by increasing the airflow, but this would have increased the fan power significantly, probably to a level even higher than compressor energy.

Table 19 summarizes the normalized cooling season energy use under each set of loading conditions for seven-day and five-day per week schedules.

Table 19. Daikin Cooling Season Energy Use Summary

Loading/Control	Seven-Day (kWh)			Five-Day (kWh)		
	Total	Base	Cooling	Total	Base	Cooling
High DM1	2,792	972	1,821	1,990	692	1,298
Ultra-High DM1	3,196	972	2,224	2,281	692	1,589

3.1.3. AAON

As shown for the energy signature of the AAON in Figure 21, the two periods of High loading (A1 and A2) show no significant differences. It exhibited a lower minimum fan speed under A2 controls than under A1 controls, but the fan speed during compressor operation was much higher; however, the net effect appears to minimal. Table 4 summarizes the details of each control mode (A1, A2, and A3).

The impact of loading is apparent between “High A2” and “Ultra (High) A2.” The data show a vertical translation of the energy signature line that projects increased energy use at all temperatures. If this line were extrapolated to lower temperatures, a lower balance point would likely develop.

The difference between loading Ultra (High) A2 and Ultra (High) A3 allows a final comparison of controls. The Ultra A3 line shows better performance than Ultra A2 at all temperatures, with some evidence of effective economizing as daily temperatures fall below 65 degrees F.

Table 20 shows the resulting energy signature parameters.

Table 20. AAON Energy Signature Parameters

Loading/Control	Slope [kWh/day*F]	Base load [kWh/day]	Balance Temp [°F]	R^2	Sample Size
High A1	0.63	10.12	46.8	0.86	6
High A2	1.05	5.50	51.3	0.86	9
Ultra-High A2	0.95	5.50	38.9	0.96	5
Ultra-High A3	1.21	5.50	50.5	0.77	10

Figure 21. AAON Energy Signature

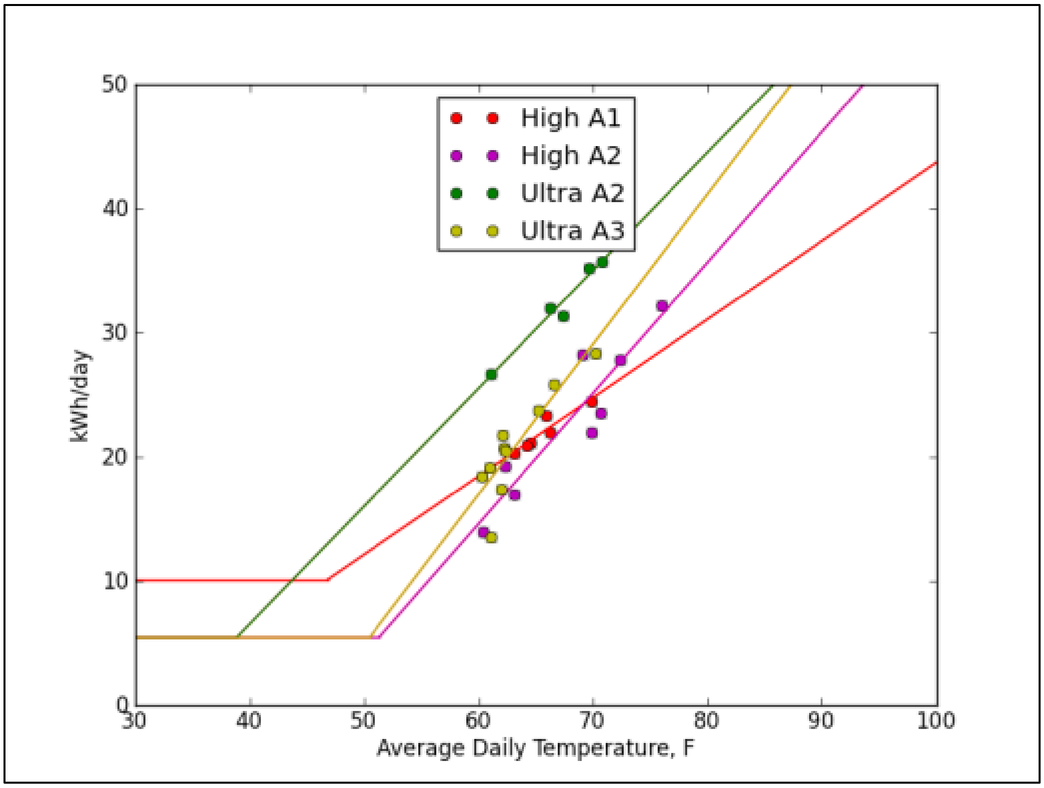


Figure 22 shows the hourly time series for all modes and loadings.

Figure 22. AAON Hourly Time Series

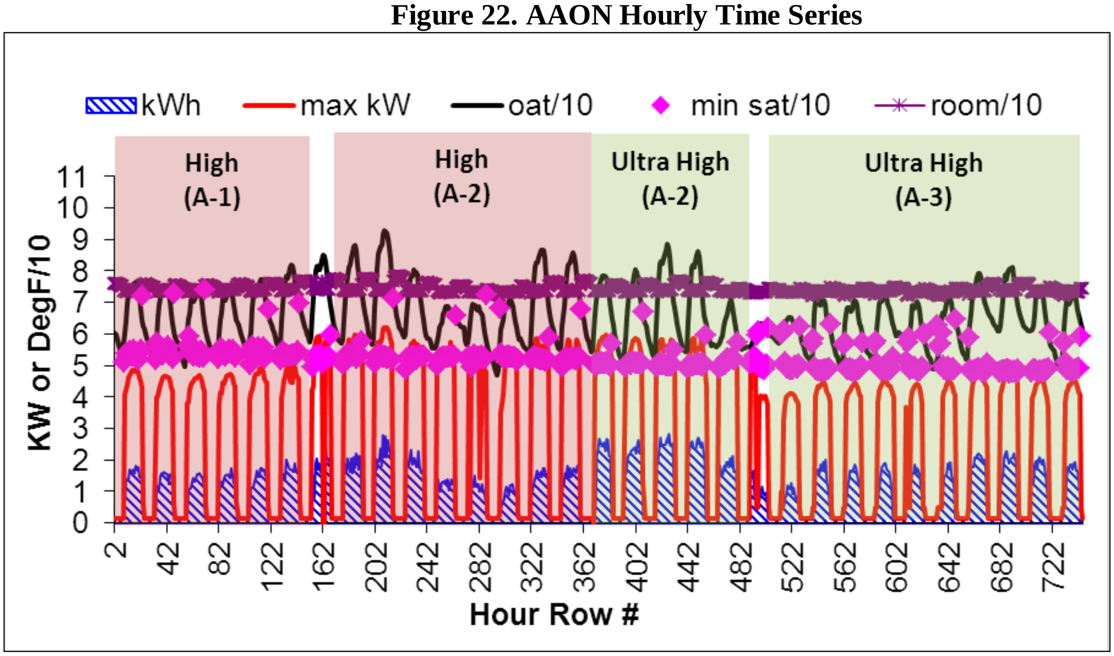
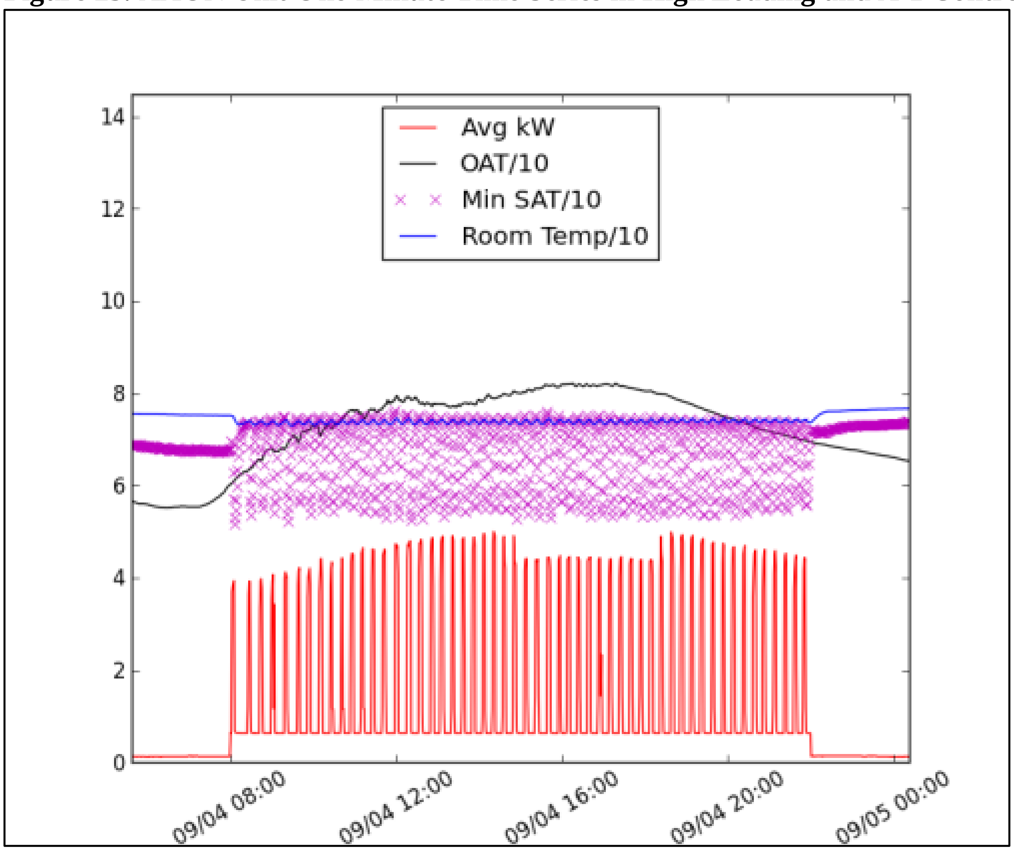


Figure 23 shows the one-minute time series for the AAON in control mode High A1. The unit is clearly doing a lot of cycling, with average power ranging from 0.6 kW to almost 5 kW. The cycle time is very short – rarely exceeding five minutes in duration.

The absence of economizer operation is also evident during this period. At 8:00 a.m. when the internal loading ramps up, the AAON's compressor immediately starts to cycle. If the economizer were active, a fan cycle would likely coincide with a period where the SAT closely matches the OAT. Researchers traced the inactive economizer to an outdoor lockout setting, and corrected this issue for control mode A3.

The presence of the Digital Scroll compressor would generally imply less cycling for this unit. Researchers began to investigate the operation of the outside controller and the Emerson E2 controller, including the ClimaCheck system, to examine data at more frequent intervals.

Figure 23. AAON Unit One-Minute Time Series in High Loading and A-1 Control Regime



The operating system of the E2 showed what appeared to be instructions to the compressor to modulate capacity. The ClimaCheck system provided one-second interval data (shown in Figure 24) during a typical cycle of the AAON compressor. The Digital Scroll uses continuously updated twenty-second windows during which the compressor is loaded (one hundred percent) for a portion of the twenty seconds, and unloaded (zero percent) for a portion, to reach an equivalent capacity modulation. Figure 24 confirms that the compressor unloads only at the beginning and end of the seven- to eight-minute cycle. Researchers suspect this “pre” and “post” modulation is a preprogrammed sequence of start-up and shutdown.

Figure 24. One-Second Power Measurements during a Typical Cycle of the AAON Compressor

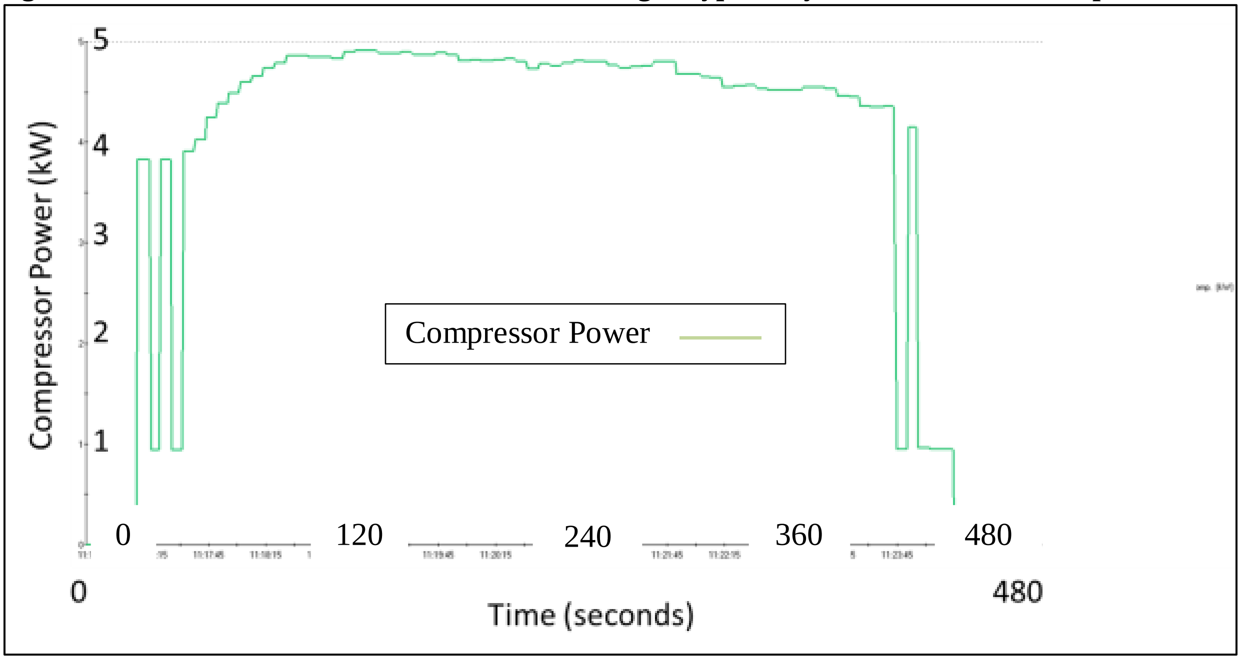
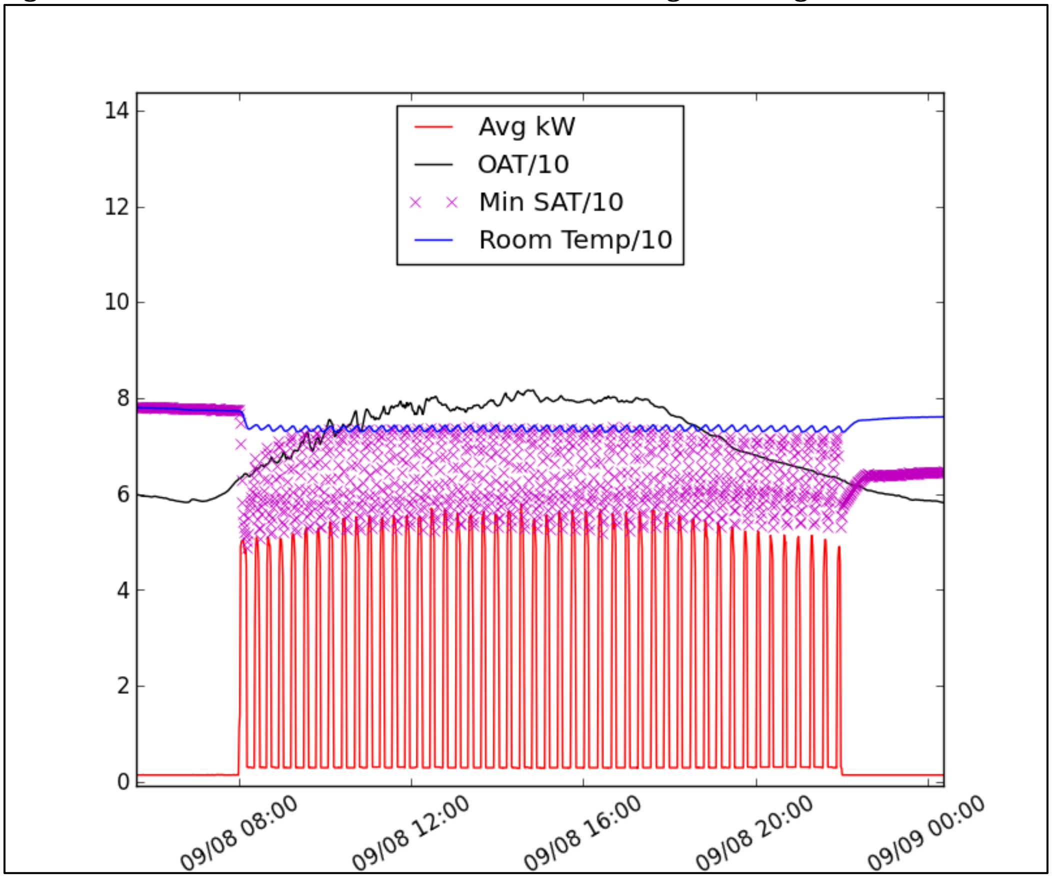


Figure 25 illustrates a similar day during the A-2 control regime. Here the fan speed during ventilation mode has been reduced, resulting in a minimum power of 0.3 kW. This corresponds to an approximate twenty-five percent reduction in fan speed. During this phase, researchers adjusted the fan speed to a higher level during compressor operation. The chart again shows no economizing.

Figure 25. AAON Unit One-Minute Time Series in High Loading and A-2 Control Regime



After a service call by Wytek personnel, researchers/Wytek personnel adjusted the economizer and fan controls. Figure 26 offers an example of the unit's operation during this A-3 regime. Inspection of the data shows that the economizer is now functioning, shown as a brief dip in power in the first cycle. The fan speed during ventilation is unchanged from A-2, with a minimum power of approximately 0.3 kW; however, the fan speed during compressor operation was reduced.

Figure 26. AAON Unit One-Minute Time Series in Ultra-High Loading and A-3 Control Regime

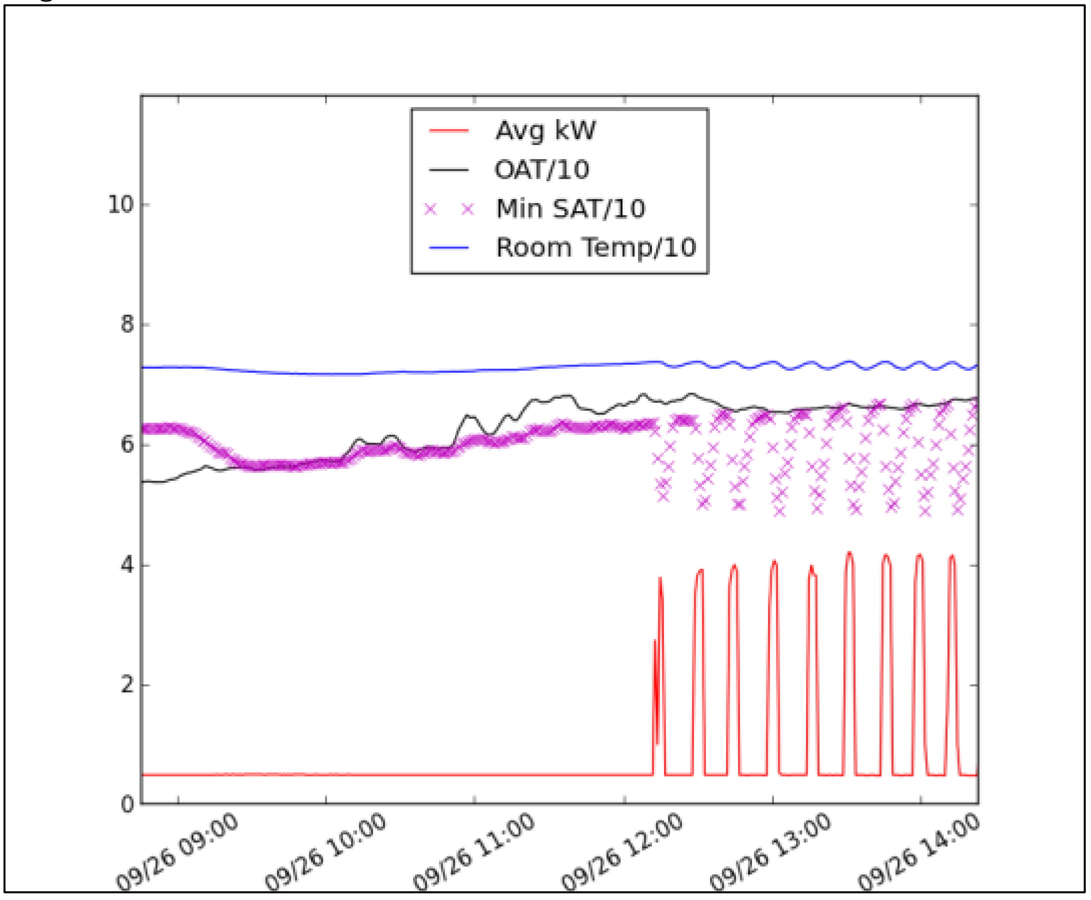


Figure 27 illustrates the mode map indicating the hourly impact of these changes, showing only the two periods of Ultra-High loading for clarity. The trend in maximum kW makes evident that the change from A2 to A3 controls significantly reduced energy use; this is due to the improved fan control that reduced fan speed (and power) during compressor operation. This effectively shows a tradeoff between refrigeration efficiency and fan power: if the fan runs faster, the delta-T between the evaporator and airstream will be greater, resulting in a more efficient refrigeration loop. However, this small efficiency increase is offset by the increased fan power required to move additional air through the ductwork.

The mode map also indicates an overall reduction in kWh across all temperatures from A2 to A3. Some economizer activity is also evident in the hours below 65 degrees F, where the maximum and average kW are both less than 1 kW.

Figure 27. AAON RQ Hourly Mode Map for Ultra-High Loading

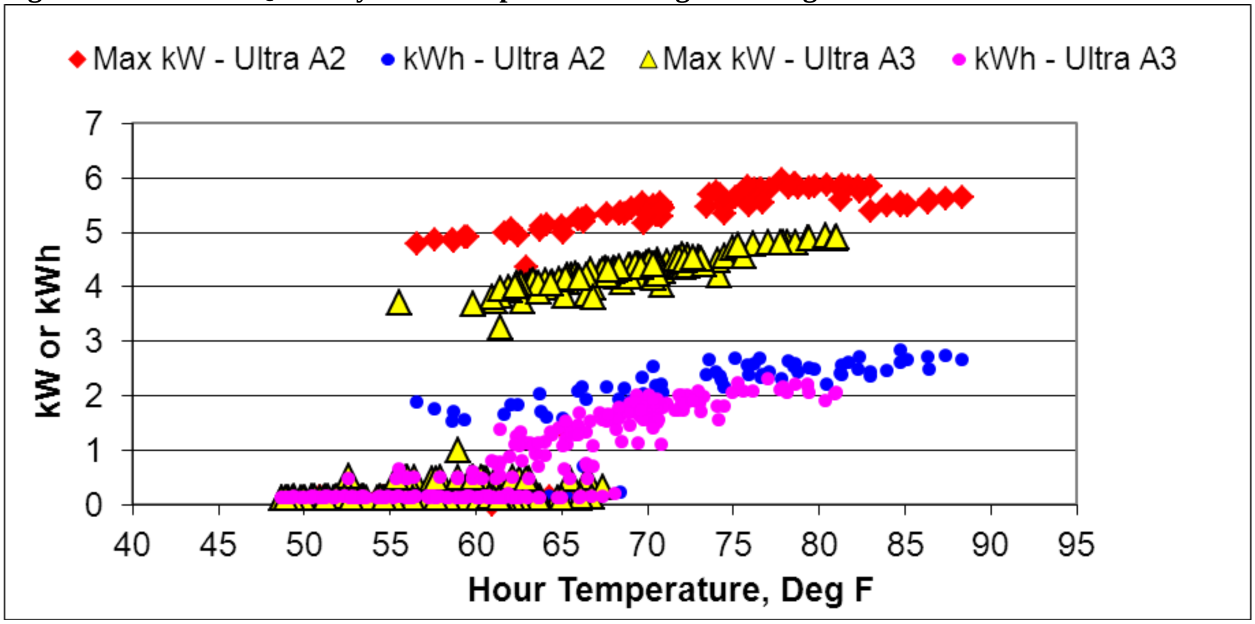


Table 21 summarizes the normalized cooling season energy use under each set of conditions for seven-day and five-day per week schedules, using the detailed energy signature parameters in Table 20.

Table 21. AAON Unit Cooling Season Energy Use Summary

Loading/Control	Seven-Day (kWh)			Five-Day (kWh)		
	Total	Base	Cooling	Total	Base	Cooling
High A1	3,668	1,862	1,806	2,619	1,326	1,294
High A2	3,174	1,012	2,162	2,261	721	1,541
Ultra-High A2	5,105	1,012	4,093	3,643	721	2,923
Ultra-High A3	3,672	1,012	2,660	2,615	721	1,894

3.2. Field Test Results

This section summarizes results from field testing at the Nampa, Idaho Fred Meyer. Researchers conducted testing on the Daikin McQuay unit for a total of 32 days between 9/15/2012 and 10/16/2012. Table 22 shows fan and operational data for each control mode. Earlier, Table 11 explains each control mode.

Table 22. Fan and Operating Data for Each Field Control Setting

	ID-1	**ID-2**
Standby power (kW)	0.14	0.12

Fan above standby (kW)	0.13	0.23
Fan hours	24	24
Outdoor Air (%)	39%	40%
Compressor-on mode airflow (CFM)	633	779
Compressor-on mode airflow (lb/min.)	43	53
Economizer	Yes	Yes

3.2.1. Daikin

Figure 28 illustrates an initial representation of this RTU's performance based on its daily energy signature. The graph shows the results for each control mode in the standard energy signature plot. The analysis notably showed that the ventilation-only fan power is higher in the "ID-2" case.

Figure 28. Daikin Unit Energy Signature - Idaho Field Testing

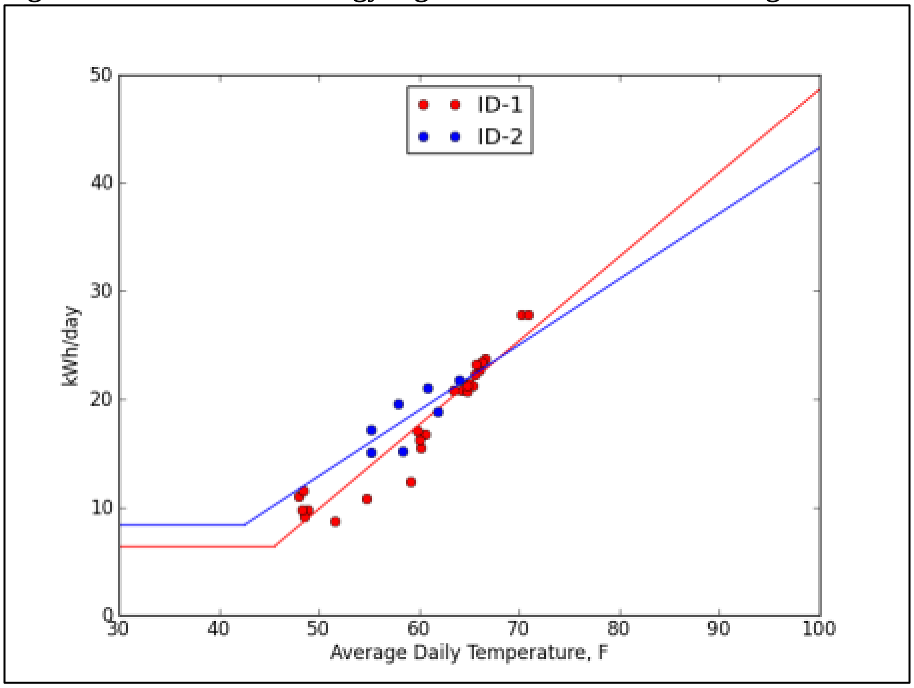


Table 23 outlines the energy signature parameters for the Daikin unit.

Table 23. Daikin Energy Signature Parameters – Idaho Field Testing

Control Settings Label	Slope [kWh/day*F]	Base load [kWh/day]	Balance Temp [°F]	R^2	Sample Size
ID-1	0.77	6.48	45.6	0.91	26
ID-2	0.61	8.4	42.5	0.59	6

Figure 29 offers an overview of each day's operation based on the hourly time series. ID-2 shows an increase in fan power based on the overnight energy use. Since this site used a twenty-four-hour fan schedule, the overnight energy use corresponds to a period of fan-only operation.

Figure 29. Daikin Hourly Time Series, Idaho Field Testing

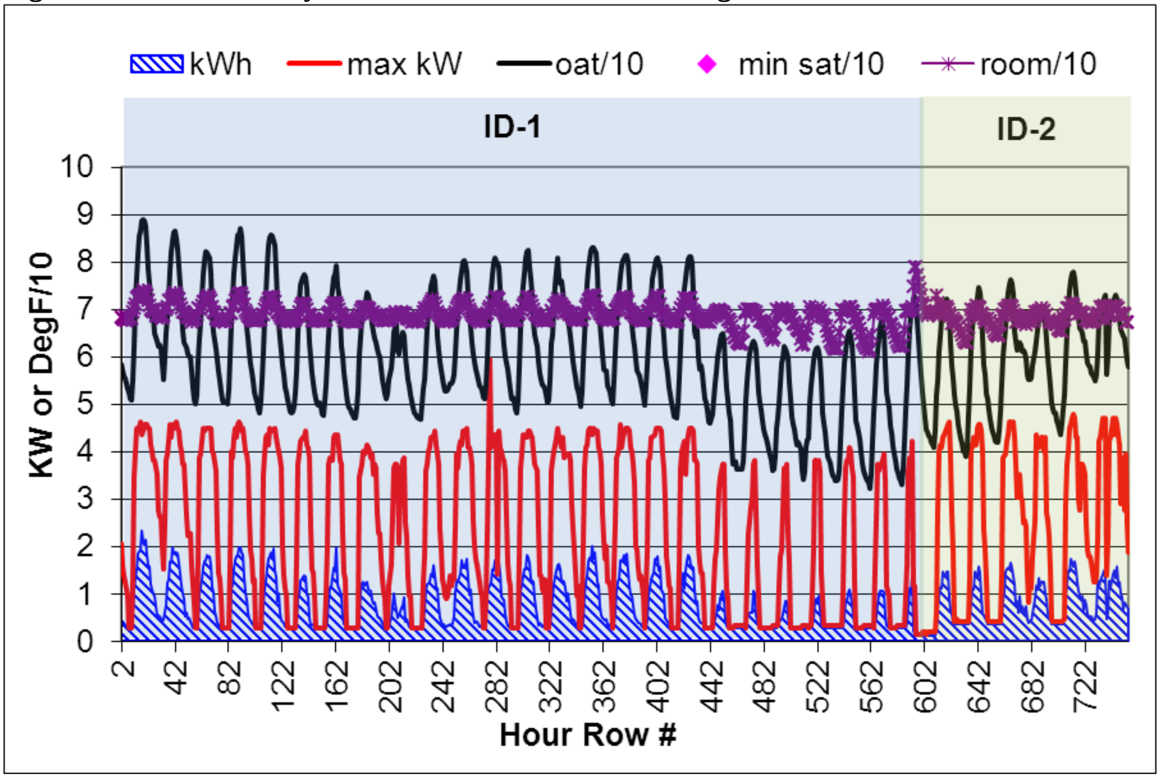


Figure 30 provides a detailed look at the RTU cycle during the ID-1 control regime with a one-minute time series. Figure 31 provides a detailed one-minute time series look at ID-2.

The ID-1 and ID-2 modes are very similar, and differ only in the matter of fan control. ID-1 uses a constant low speed fan with airflow of about 156 CFM; ID-2 uses a slightly higher base fan speed of 232 CFM, along with increases in the fan speed during economizer mode and compressor mode. Figure 31 illustrates this increase in fan speed (and therefore fan power).

The general economizer control was consistent in both modes, aside from the increasing fan speed in ID-2. For outside air temperatures above 55 degrees F, the economizer cycle would start with dampers opening for approximately five minutes of economizer-only operation. After the short economizer-only cycle, the compressor would turn on to finish the cycle in an integrated economizer mode.

Temperatures at or below 55 degrees F exhibited very few economizer-only cycles due to the rather high setting (approximately forty percent) for the minimum outside air fraction. This configuration admitted sufficient cool outside ventilation air so that additional cooling by either an economizer-specific mode or a compressor was unnecessary during the unoccupied lower gain hours. In fact, this high level of outside air served as a de facto economizer at low temperatures, and would pre-cool the space by about 5 degrees F on cold nights.

The compressor control for each of the modes appeared to be the same. The compressor would start each cycle by ramping up to a high power level, then backing off until an approximate 55 degrees F supply air temperature was reached. Then it would converge on a power level that maintained the 55 degrees F supply temperature. During the day as the cooling load increased the converged power level would also increase. Cycles ranged from five to fifteen minutes.

Each cycle ends with the compressor turning off but the fan remaining on. When the compressor turns off, the ventilation airflow of about forty percent outside air would heat up the space within a few minutes, and the cycle would start again. During warm conditions (greater than 80 degrees F), this cycle resulted in a ripple in the room temperature as it oscillated between 70 and 75 degrees F.

This type of compressor cycle contrasts sharply with the compressor operation observed for the same Daikin unit at the NBIL.

Figure 30. Daikin One-Minute Time Series, ID-1 Control Regime, Field Testing

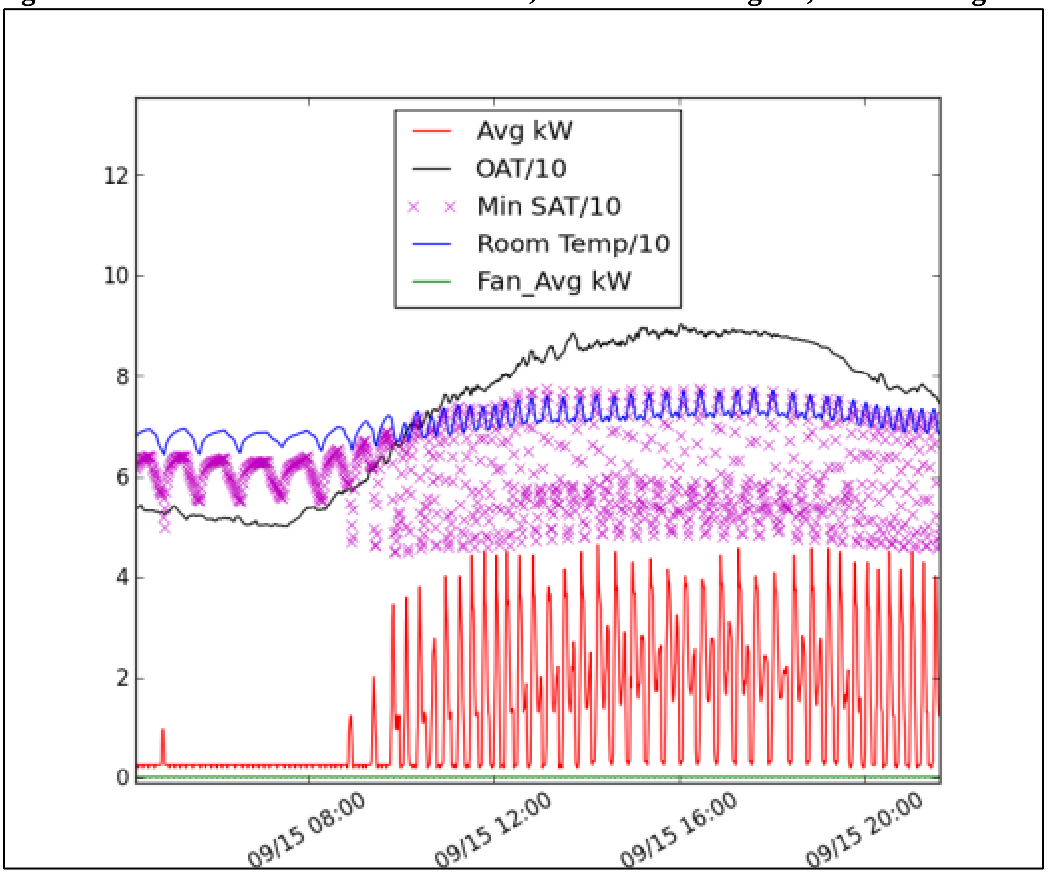


Figure 31. Daikin One-Minute Time Series, ID-2 Control Regime, Field Testing

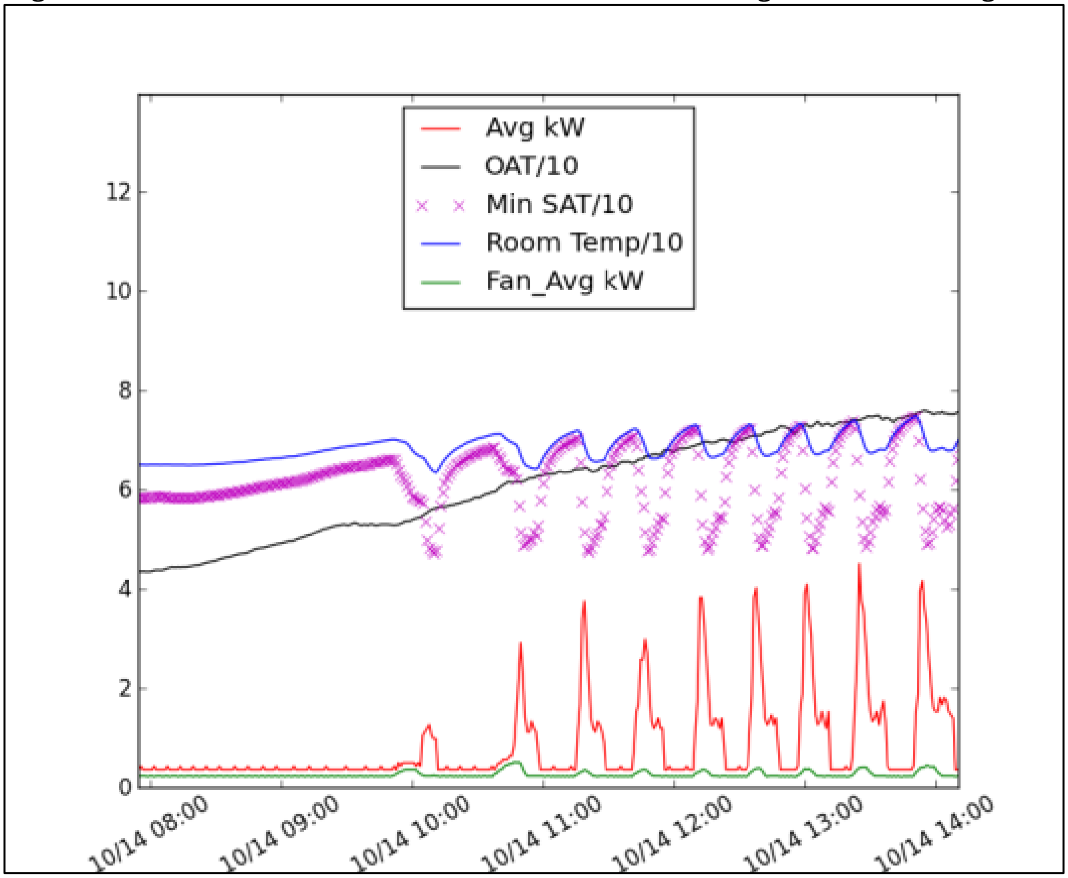


Table 24 summarizes the normalized cooling season energy use under each set of conditions for seven-day and five-day per week schedules; Table 23 presents the detailed energy signature parameters.

Table 24. Daikin Cooling Season Energy Use Summary – Idaho Field Testing

Testing Regime	Seven-Day (kWh)			Five-Day (kWh)		
	Total	Base	Cooling	Total	Base	Cooling
ID-1	3,932	1,192	2,740	2,764	849	1,915
ID-2	4,025	1,546	2,479	2,842	1,100	1,742

3.3. Discussion of Results

3.3.1. Comparison of Lab-Tested RTUs

The researchers calculated normalized cooling season energy use consumption for each unit at both High and Ultra-High loading levels using an assumed seven-day and five-day schedule as described earlier. The discussion here pertains to the seven-day schedule, but the researchers saw the same trends in the five-day data with slightly lower numbers.

Given that the airflow rates and pounds per minute of outdoor air differed for each unit, the researchers do not recommend directly comparing the annual projections using the lab data. However, the testing informed the Physical Model, which is better-suited to direct comparisons.

For the Trane unit, the cooling season energy use estimates show the anticipated pattern of increasing consumption as the researchers changed the loading from Low, to High, to Ultra-High. Between Low and High loading, energy use increased by 696 kWh, or twenty-six percent. From the High to the Ultra-High loading, energy usage increased by 790 kWh, or an additional twenty-three percent. In all cases, the increase in energy usage is due to an increased cooling demand. Cooling accounts for twenty, thirty-seven, and forty-nine percent, respectively, of total energy use across the Low, High and Ultra-High loadings.

During High loading, cooling season energy use estimates range from 3,668 kWh (AAON A1) to 2,792 kWh (Daikin DM1). Ultra-High loading ranges from 5,105 kWh (AAON A2) to 3,196 kWh (Daikin DM1). This equates to energy savings between twenty-four and thirty-seven percent between the lowest and highest energy usage units.

Testing on the AAON unit provided the opportunity to evaluate the impact of different control strategies applied to the same RTU. Under High loading, a change in controls from A1 to A2 reduced the base load significantly and resulted in a cooling season savings of thirteen percent. Almost all of the savings came from reduced ventilation-only energy use. Under Ultra-High loading, a change from A2 to A3 enhanced the economizer operation and resulted in cooling season savings of twenty-eight percent. These savings came not from the base load, but rather from reduced compressor energy use as a result of the economizer operation.

Table 25. Normalized Cooling Season Energy Use Comparison Using Seven-Day Schedule

Testing Regime	Published SEER	Seven-Day (kWh)			Five-Day (kWh)		
		Total	Base	Cooling	Total	Base	Cooling
Trane Precedent							
Low T1	13	2,644	2,109	535	1,912	1,501	411
High T1		3,340	2,109	1,231	2,388	1,501	887
Ultra T1		4,130	2,109	2,022	2,955	1,501	1,454
Daikin McQuay Rebel							
High DM1	18	2,792	972	1,821	1,990	692	1,298
Ultra DM1		3,196	972	2,224	2,281	692	1,589
AAON RQ Series							

High - A1	14.8	3,668	1,862	1,806	2,619	1,326	1,294
High - A2		3,174	1,012	2,162	2,261	721	1,541
Ultra - A2		5,105	1,012	4,093	3,643	721	2,923
Ultra - A3		3,672	1,012	2,660	2,615	721	1,894

This series of testing has treated the Trane unit as representative of a commonly-installed code-baseline RTU. Both the AAON and Rebel units performed better than the baseline when their controls were properly configured. The AAON unit illustrates how the wrong control configuration (A1 or A2) can cause even a high-performance RTU to use more energy than the baseline model. Interestingly, the AAON A2 configuration showed positive savings under High loading, but negative savings under Ultra-High loading. This particular control configuration used an increased fan speed during compressor operation; with increased loading, the researchers expect the compressor duty cycle to increase, along with the number of hours the fan operates at the increased speed.

When the researchers configured the controls more favorably, the AAON savings range from five percent in High loading A2 to eleven percent in Ultra-High loading A3. The researchers didn't test for High A3, but a rough estimate predicts savings of twenty-eight percent. The Daikin unit saved sixteen percent under High loading and twenty-three percent under Ultra-High loading.

Compared to the AAON and Daikin units, the Trane uses more energy under almost all circumstances – not surprising, given it has the highest base load among the tested units. If the Trane reduced its fan power to the level observed on the AAON or Rebel, significant savings may be possible.

Figure 32. A Plot of Seven-Day Cooling Season Energy Use versus Published SEER for High and Ultra High Loadings

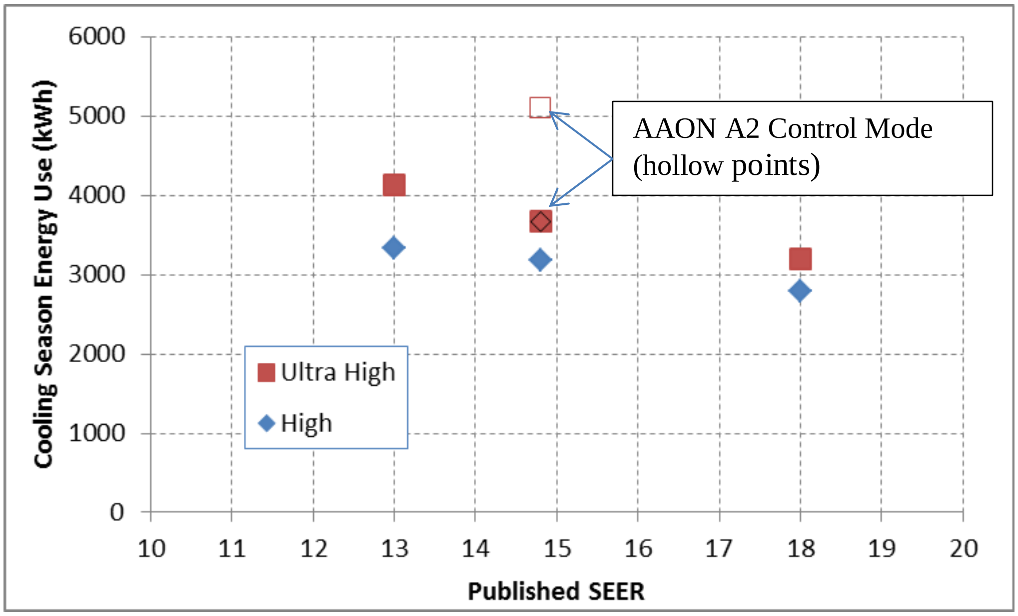


Figure 32 shows a comparison of the seven-day cooling season energy use projections for each unit versus the published SEER. Notably, increasing SEER and decreasing projected energy use appear to be correlated. However, when researchers made controls changes that resulted in very high compressor fan speeds (mode A2 shown by the hollow points), the correlation became much weaker.

While characterizing this relationship would likely provide no value, it is notable that the correlation is largely reduced by improperly-configured controls.

3.3.2. Comparison between Lab and Field Results

One of the goals of this study is to better understand how the behavior of an RTU changes in response to the conditions of a given site. Comparing results between the NBIL and the current Idaho field site illustrates the differences between these two sites. Using the Daikin as a point of comparison is a logical first step, as the researchers tested this unit at both sites.

As anticipated, the scheduling and RTU configuration at the Idaho site differed from the conditions observed at the NBIL.

Table 26 summarizes key operational parameters, as observed at each site. ID-1 and ID-2 clearly differ from one another in terms of airflow. Both modes differ significantly from the NBIL test mode in terms of fan power, fan hours, and percentage of outside air. Due to the differing conditions, the NBIL results for both the Trane code unit and the Daikin are not directly comparable to these field results without proper correction for these and other potentially significant differences.

Table 26. Comparison of Key RTU Parameters between NBIL and Field – Daikin Unit

	NBIL	Field	
	DM-1	ID-1	ID-2

Standby power (kW)	0.15	0.14	0.12
Fan power (kW)	120	130	228
Fan hours	14	24	24
Outdoor air fraction (%)	10%	39%	40%
Airflow (CFM)	840	633	779
Airflow (lb/min.)	62	43	53
Economizer	Yes	Yes	Yes

3.3.3. Applying the RTU Physical Model

The test work at the NBIL had an underlying purpose of devising a methodology for estimating the change in RTU energy use as a result of changes in RTU parameters. This allows testing of an RTU under one set of conditions to be more broadly applicable to estimating energy use under a different set of conditions. During the course of this project, NBI developed a physical RTU model for this purpose. Using the RTU Physical Model, it is possible to estimate how the code baseline Trane unit might have performed, in terms of an energy signature, at the Idaho site.

Appendix A includes details on the model.

The Physical Model also provides an alternative method of estimating normalized cooling season energy use by fitting the RTU model to the measured RTU energy use. The RTF protocol uses a simple linear regression/change-point parameters model to fit the measured energy use pattern. The Physical Model uses a more complex function that considers the interactions among the outdoor air temperature, COP, building envelope, ventilation rate, temperature setpoint, and other independent variables. The example in Figure 33 shows the calibrated model co-plotted with measured energy use data from the ID-1 and ID-2 control regimes. The dashed line represents the theoretical path for a fully functional and optimized economizer. The solid red line comes from the model and provides a base reference for a hypothetical building that has zero internal gain.

Figure 33. RTU Physical Model Energy Signature for Daikin ID-1 Data from Field Testing

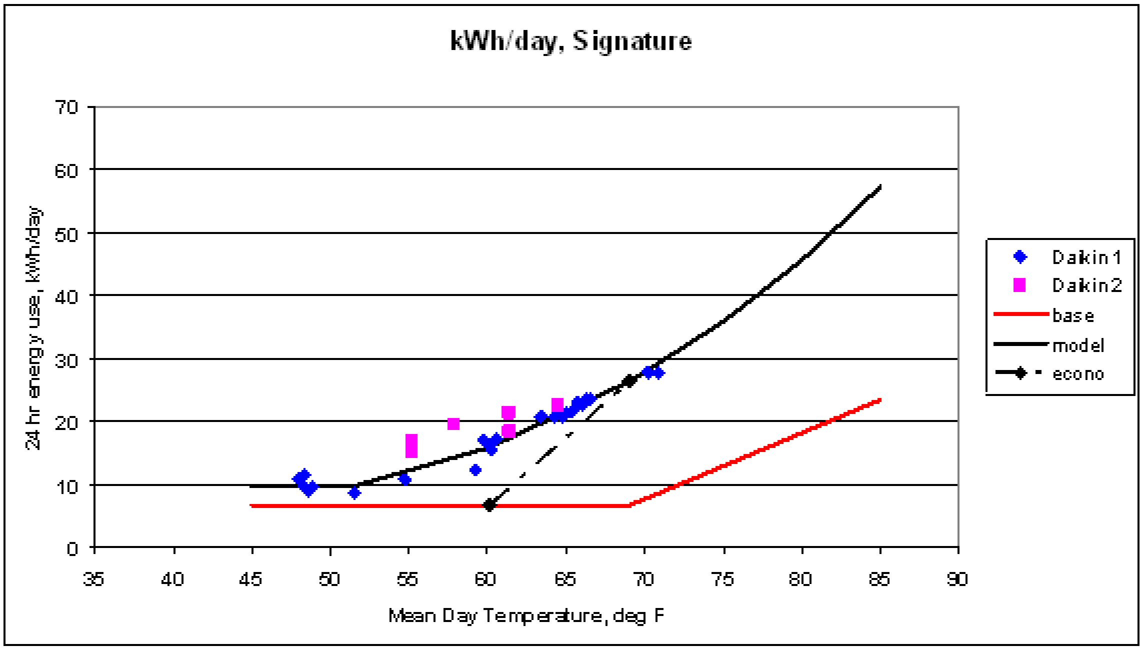
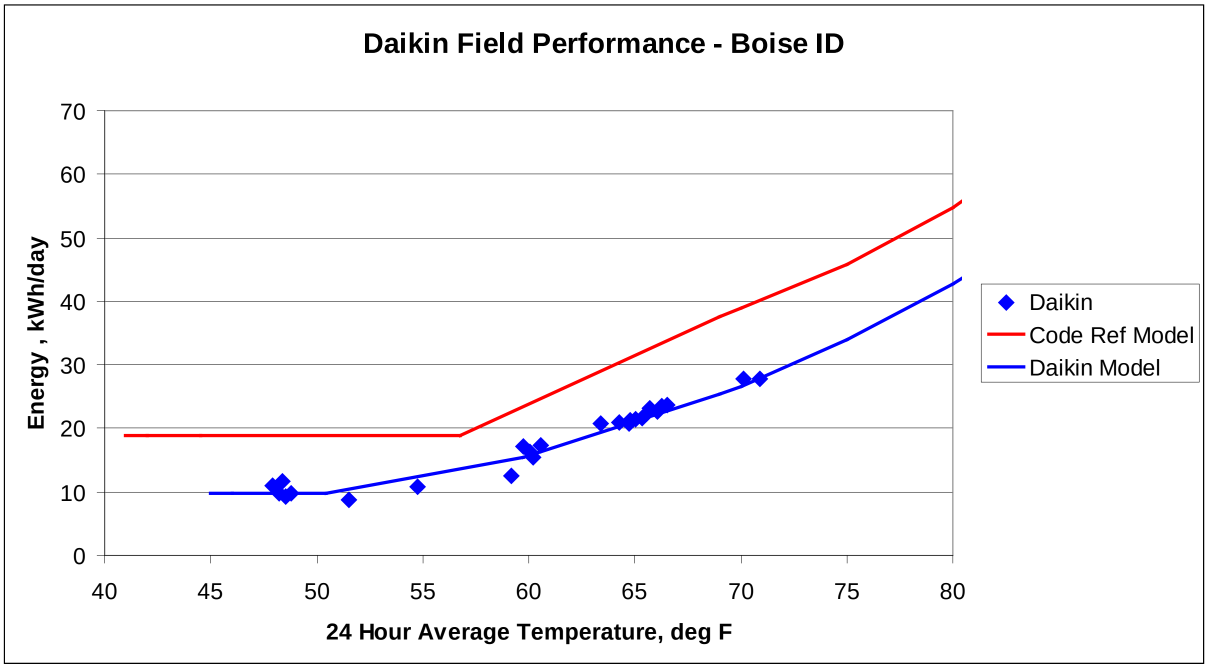


Figure 33 shows that the base load energy use never drops to the theoretical minimum base energy use expected from the fan alone. The team suspected this was due to the integrated operation of the economizer that triggers a bit of compressor energy during economizer cycles. This type of signature is not typical of an RTU with a properly-functioning economizer. In fact, this unit did have properly articulating dampers, but the compressor energy during economizing dissipated the energy savings that would have come from the fan-only fresh air cooling. The Discharge Set Point control setting regulates the use of integrated versus economizer-only operation; an increase in this setpoint would likely have permitted more economizer operation.

3.3.4. Projecting Code Unit Comparison

Applying the Physical Model necessitates setting the inputs to match the site and RTU conditions. For the Idaho installation, site measurements and inspection established all but two of the model inputs. The researchers achieved the final fit of the model to the data by altering the key site drivers for thermal effect (BTU/°F*day) and thermal gain (BTU/day) until the model fit the data. Once these two thermal parameters are determined for one unit, the model can use them to estimate the energy use of a different unit installed on the same zone. The model used this methodology to estimate the energy use of a Trane baseline unit at the Idaho site. Figure 34 shows the predicted performance of the Trane unit in Idaho, indicated by the red line labeled “Code Ref Model.” This is co-plotted with both the observed data for the Daikin unit and the fitted model of the unit’s performance.

Figure 34. Physical Model Estimated Performance - Daikin and Projected Trane Baseline Unit in Boise



During testing at the NBIL, the economizer on the baseline Trane unit was not operating as intended. Using the Physical Model makes it possible to estimate how the Trane unit will perform in Boise both with and without a functional economizer; Table 27 presents this data. In the case of the Daikin unit, researchers estimated the "No economizer" data by fitting the Physical Model to the observed data (as shown in Figure 34). Inspection of the data revealed that the Daikin was using inefficient integrated economizer logic that the researchers believe could likely be improved. In the right-side "Improved Economizer" columns, Table 27 presents the performance with a hypothetical economizer.

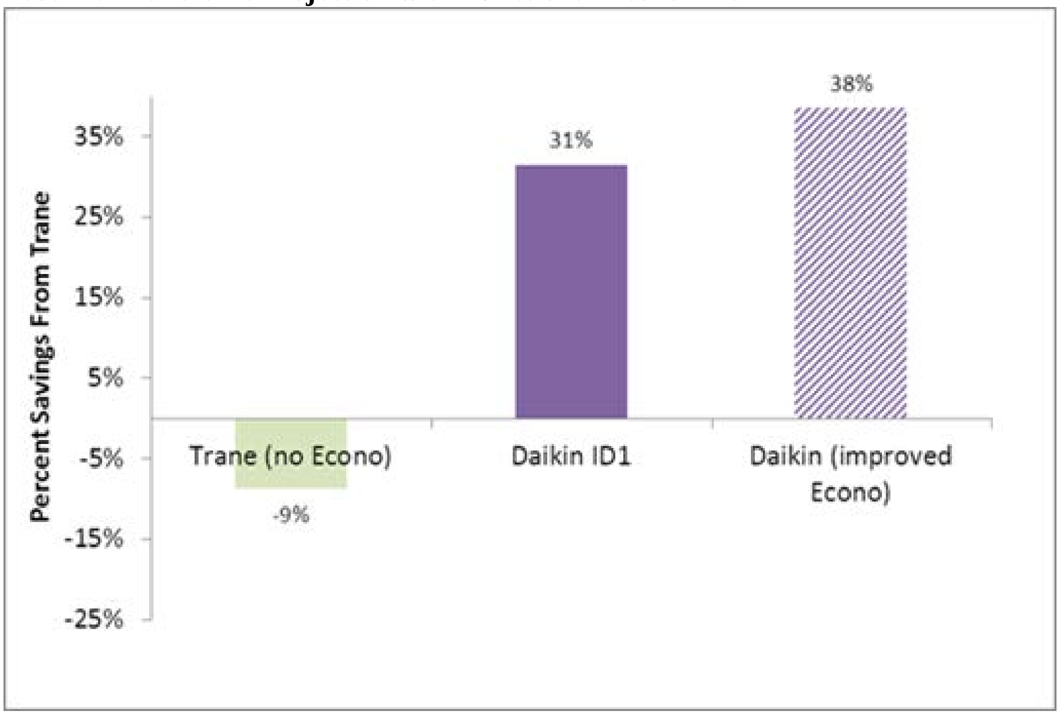
Note that the researchers estimated cooling season energy use based on the conditions of the field site, which used a twenty-four-hour fan schedule. Most of the difference in energy use between the code baseline unit and the Daikin unit is attributable to reduced fan energy (750 watts for the baseline fan vs. 130-230 watts for the Daikin fan).

Table 27. Normalized Cooling Season Energy Use, Estimated Using RTU Physical Model for Boise Field Site

Code Unit Economizer State	Code Baseline ("Trane") (kWh)	ID-1 (kWh)	Savings ID-1 (kWh)	Savings (%)	ID-1 Improved Economizer (kWh)	Savings ID-1 Improved Economizer (kWh)	Savings (%)
No Economizer	6,618	4,176	2442	37%	3,749	2869	43%
With Economizer	6,091		1915	31%		2342	38%

Figure 35 shows a comparison between the Daikin unit and the baseline Trane unit (with economizer). This comparison is specific to the field site and its operation pattern (the internal loading conditions, outside air rate, and use of a twenty-four-hour fan schedule). Comparing the savings observed at the Nampa field site to the twenty-three percent savings observed between the Daikin and Trane units (Ultra-High loading) at the NBIL is important. The thirty-one to thirty-eight percent savings observed in Nampa exceed the savings observed in the NBIL, a distinction largely driven by differences in fan schedule (twenty-four hours in Nampa; fourteen hours at the NBIL) and the fact that Nampa has a harsher cooling season than Portland (more cooling degree days).

Figure 35. Comparison of Cooling Season Energy Use for the Daikin ID-1 Relative to Baseline Trane Unit Projection with Functional Economizer



3.3.5. Integrated Economizing

In theory, integrated economizers reduce the overall power for an equivalent cooling rate. However, this argument does not include the option of pre-cooling. In the field case, pre-cooling could start about two hours prior to formal occupied start-up, which would erase the setback with the cool morning air and allow the space to be comfortable for the first few hours without the need for compressor operation. Such an approach would probably lower the three-hour morning cooling energy to less than fifty percent of that observed by the researchers.

4. Conclusions and Recommendations

4.1. Conclusions on Controls and Operational Settings

During this study, NBI and IDL tested three RTUs at the NBI Laboratory in Vancouver, Washington, and one RTU at a box retail store in Nampa, Idaho. The projection of cooling season energy use for each unit at different sensible internal loading and control configurations provided results that led to several conclusions:

- When properly configured, both the AAON RQ and the Daikin McQuay Rebel outperformed the Trane Precedent unit. Under High loading, the AAON saved five percent in one control configuration, and the researchers estimate it would save at least twenty-five percent in a more optimum control configuration; the Daikin saved sixteen percent. With Ultra-High loading, the AAON saved eleven percent and the Daikin unit saved twenty-three percent over the Trane baseline.

- Applying an RTU Physical Model enables estimation of the performance of a Trane baseline unit subject to the same environmental and loading conditions that the Daikin unit experienced during testing in Idaho. This estimate suggests that the Daikin is capable of thirty-one to thirty-eight percent savings at the Idaho site.

- At both sites, the advanced units achieved large proportions of their savings by reducing the supply fan speed. The units exhibited reduced fan speeds during periods of fan-only operation (ventilation mode) and slightly increased fan speeds during compressor and/or economizer operation. If the Trane baseline unit were retrofit with advanced fan controls, estimated savings of nineteen percent under High loading and eight percent under Ultra-High loading would accrue.

- The research team was unable to determine whether compressor modulation saved energy. It would be logical to conclude that compressor modulation would result in a lower EER, but compressor modulation does prevent condenser coil issues when drastically reducing fan speeds; thus compressor modulation is necessary to reduce fan speeds in some situations.

- Installing an advanced RTU does not guarantee energy savings. Although the advanced units enable a variety of energy-saving opportunities, they require proper configuration and control settings. This was especially evident when testing the AAON, which initially performed worse than the Trane baseline unit. After researchers adjusted the initial control settings, a modified control regime resulted in five percent energy savings at High loading. However, when researchers tested this same control regime under Ultra-High loading, its performance was twenty-four percent worse than the Trane baseline (-24% savings). A final control regime achieved eleven percent savings under Ultra-High loading.

- Although the Trane does not modulate its compressor or fan, the fast cycling apparently does not result in an EER penalty for the Trane. All of the “coolth” is harvested over time by the fan during each cycle. ClimaCheck personnel noted that the fast one hundred percent cycling may have deleterious effects on the compressor that would shorten its life relative to a modulated compressor.

- All of the RTUs tested showed some degree of short-circuiting between the supply and the return air, such that the return air temperature would decrease slightly during periods of heavy cooling. This short- circuiting event had no effect on the AAON or Trane, which were controlled from the wall thermostat. However, for the Daikin, the artificially lower return air temperature used as the control temperature informed the Daikin that it had already met the setpoint, which it had not. Interestingly, the relatively low airflow of the Daikin (about half that of the other two units) was able to set in motion a short circuit flow, which demonstrates that it may be easier to set up a short circuit airflow in a low-flow situation rather than in a high-flow turbulent situation.

- The Daikin used an integrated, compressor-augmented, economizer approach. The argument in favor of integrated economizers is that the overall power is lower for an equivalent cooling rate. However, this argument does not include the option of pre-cooling. In this case, pre-cooling could start about two hours prior to formal occupied start-up, which would erase the setback with the cool morning air and allow the space to be comfortable for the first few hours without the need for compressor operation. Such an approach would probably lower the three-hour morning cooling energy to less than fifty percent of what the researchers observed.

- The Trane and AAON units maintained very steady room temperatures at the designated thermostat location. However, the Daikin's room temperature tended to rise during warm periods; this drifting room temperature observed for the Daikin was due to the wider deadband of the control temperature under the Discharge Set Point control mode. During moderate outdoor temperatures in the range of 65 to 75 degrees F, the return air temperature and the room air temperature differed by very little. However, during periods of high outside air temperatures, the room air temperature would rise by about 1 to 1.5 degrees F above the return air temperature.

4.2. Recommendations

Continue field testing of advanced RTUs.
An enhanced dataset of RTU field data will continue to reinforce current estimates of energy savings for advanced RTUs. Testing should focus on using a consistent data collection methodology and controlling for independent variables such as building type and occupancy schedule. Researchers should allocate sufficient time to ensure each phase of testing occurs over a wide range of average daily temperatures and a large number of days (ideally twenty-eight days including weekdays and weekends/holidays). Researchers should also develop and adhere to detailed specifications for advanced RTU settings and configurations.

The general objectives initiated in this study continue to be relevant:

- How does the energy signature change in response to changes in thermal loading (kWh/day of internal building loads) for each of the three units, and how do the units compare?
- Which control nuances typify high performance units available today, and what are the performance implications of setup control choices?

- Can the researchers derive a functional RTU model, what is the minimum dataset needed, and how does it clarify the expected savings of the higher efficiency units in the lab and in the field?

Develop specification guidance.

The standard practice of simply specifying a unit with a higher EER or SEER is not sufficient to capture the savings potential of an advanced RTU. Not only must the proper RTU features be specified, but the installation, configuration, and control settings must also be appropriate for the space served. Sufficient research would allow development of a comprehensive decision matrix to ensure proper specification and control of advanced RTUs on small to medium commercial buildings.

In addition, a new regime of remote connectivity in which devices can use Wi-Fi, cellular, or hard-wired connections to the Internet brings into question the level of “Monitoring-Based Commissioning” possible using routines developed for Fault Detection and Diagnostics to ensure that installers set up the units properly. This concept invites further study by working with the manufacturers to determine the minimum period of time and data points needed to establish a best practice for temporary verification of proper installation and controls setup.

Continue research on control retrofit options.

This round of testing identified the supply air fan as the major energy user in a traditional RTU, a widely-supported conclusion. Observation of the two advanced RTUs demonstrated that fan energy savings are achievable and translate to overall RTU savings on the order of thirty percent. In theory, the proper fan control retrofit may achieve similar savings at a lower cost, or may provide a cost-effective retrofit option for existing RTUs. Researchers should test existing RTU retrofit options with an emphasis on technology options that focus on supply fan savings. NBI also recommended this action in an earlier NBI report on rooftop units prepared for NEEA. This investigation may begin to explore demand control ventilation options and their overlap with traditional thermostatically-controlled systems.

The application of the Physical Model has been limited thus far to the VRTUT test units and data. To extend Physical Model development and verification, NBI proposes to test the model using the following:

- Field data being collected by PNNL in collaboration with BPA. This dataset is also being accessed by Bill Koran of Northwrite, Inc. to support the build-out of the Energy Charting and Metrics (ECAM) model for NEEA and BPA.
- Additional datasets to be identified in discussions with several evaluation consulting firms involved in RTU measurement-related projects in the Northwest and California.

With the Physical Model, NBI researchers expect to be able to identify in any given unit the following conditions:

- Proper sizing for the space served, oversized or undersized
- Refrigerant circuit performance within expected range based on expected energy usage
- Supply fan operating performance within expected range

- Ventilation sufficient, not sufficient, excessive in the unit and for the space
- Duct system efficiency
- Other imbalances, including air leakage and overpressurization (zone level)

NBI staff expects data acquisition costs for inputs to the model from one-time site measurements on the order of $1,500, including a two-week monitoring period.

In addition, NBI staff recommends applying the Physical Model to variable refrigerant flow (VRF) systems products to determine its applicability and to identify additional parameters that would enable the model to apply to VRF systems.

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Appendix A – RTU Physical Model

A primary objective of this work is to relate the performance of the high efficiency RTU to the performance of a code level unit serving the same load. However, in the field test, the researchers only installed the high efficiency unit; the code level unit will be modeled. This section describes the initial working model and its parameters. A subsequent report will provide additional details and analysis.

It is important to recognize that the conditions that ultimately drive the RTU operation at the field site will in general be unique to that site. Therefore, the model of the code unit will need to be responsive to these new conditions. The testing of the three units at the NBIL under a structured sequence of known loading conditions is intended to lead to a model of RTU energy use that is responsive to the most significant of the site conditions, in essence a Physical Model of an RTU. The researchers derived the resulting Physical Model by fitting the average day energy signature metered at the NBIL to an analytical model.

In principle, the analytical model is a simple energy balance, but in practice, a minimum set of parameters is required, as shown in Table 28. The situation is more complicated, but in reality, the following input parameters were sufficient to achieve a good fit between the data and the model.

Table 28. RTU Physical Model Input Parameters

Parameter	Description	Units
Nominal Size (tons)	Size of RTU	Tons of cooling
Neutral Temperature	Point of interception with the “fan only” portion of energy signature	degF
COP at neutral temp	Calculated COP at neutral temperature	BTU/BTU
Airflow	Airflow rate at full fan power	lb/min
Fixed fan run time	Ventilation schedule in hours	hours
Fixed fan power	Fan power in ventilation mode	Watts
Standby power	Power when unit is off	Watts
Fan Power during compressor	Fan power during call for cooling	Watts
Economizer effect, 0 to 1	Aggressiveness of the economizer settings	Dim.
Minimum OSA fraction, percent of full flow	Outside air fraction	%
Thermal effect at space	Solved parameter	BTU/degF-day
Space gain	Solved parameter	BTU/day

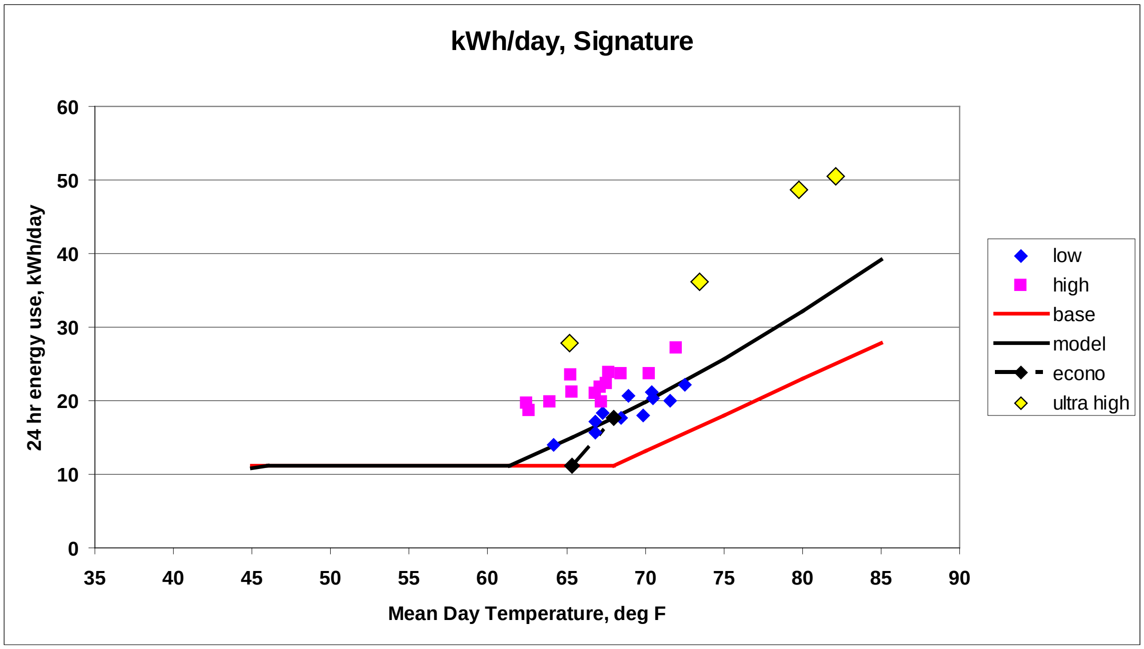
Researchers can derive most of the parameters in Table 28 by inspection or by simple one-time measurements, except for the last two. These last two parameters, Thermal Effect and Space Gain, describe the internal load to which the RTU is responding; as such, they are the most significant determinants of the energy use of the RTU. These two parameters are difficult, if not impossible, to measure independently because they include interactions of the conditioned space with adjacent spaces, un-quantified air leakage effects, and solar gain and other radiant effects.

In spite of the potential complexity of the situation, the empirical work at the NBIL has shown that these parameters are identifiable in aggregate form in the energy signature.

Recent Lab Simulations

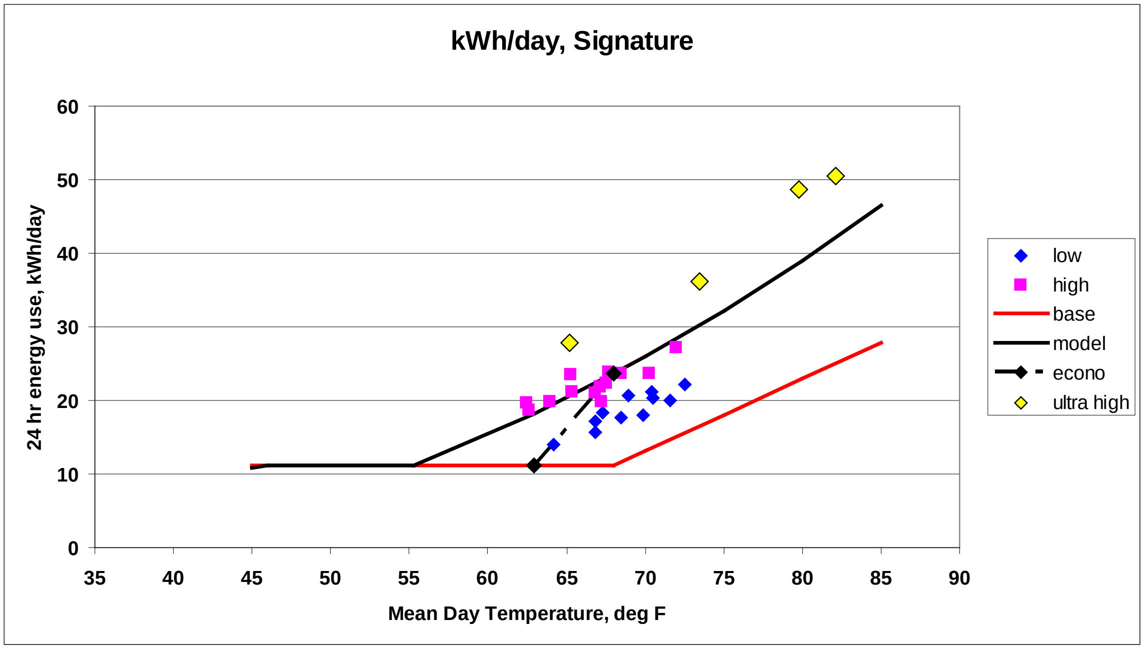
Recent work at the NBIL used a conditioned space and a metered RTU to simulate different levels of internal gain. This yielded a set of results that reveals the underlying structure of a physical RTU model. Figure 36, Figure 37, and Figure 38 show the energy signature results for three levels of simulated internal gain: Low, High, and Ultra-High. In these figures, the black line is the RTU model without an economizer, and the dotted black line shows the model with a properly functioning economizer. The line labeled "Base" is the estimated performance of the RTU in the absence of any internal gain; in other words, it is just counteracting the external gains of solar, conductive heating, and introduced ventilation air.

Figure 36. Energy Signature of the Trane with Model Fit to Low Loading Points



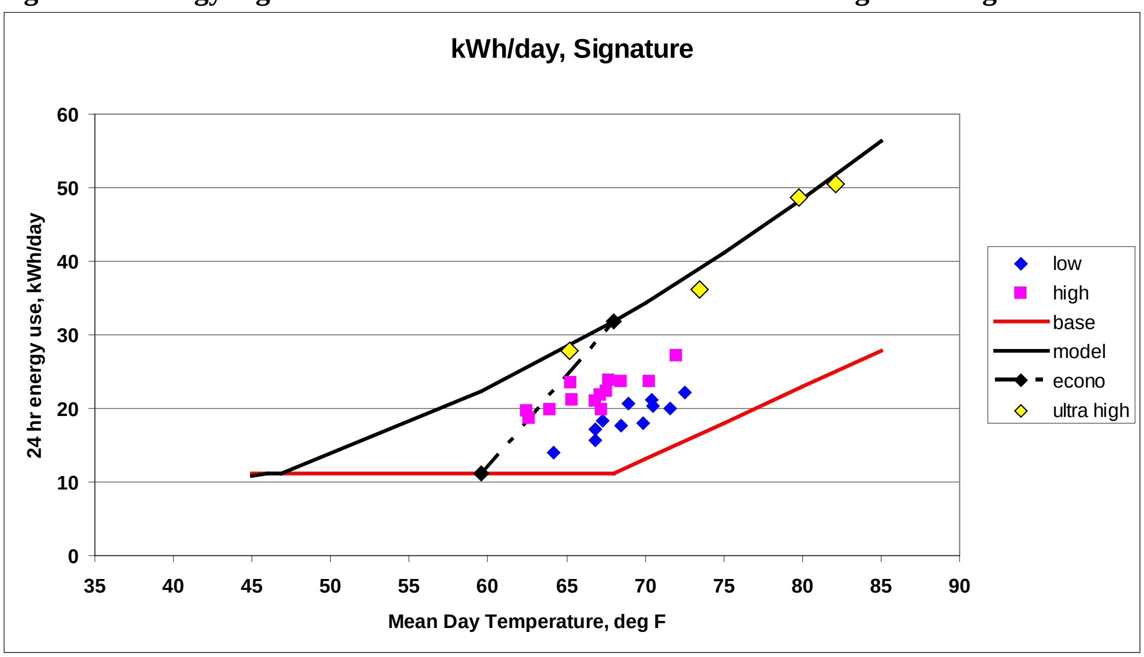
This Low loading corresponds to that of a very efficient office; it also provides little benefit to the economizer.

Figure 37. Energy Signature of the Trane with Model Fit to High Loading Points



This High loading corresponds to a typical office environment. At this higher level of loading, the benefit of the economizer is significantly increased.

Figure 38. Energy Signature of the Trane with Model Fit to Ultra-High Loading Points



This Ultra-High loading corresponds to a heavily-occupied office operating for long hours. In this case, the economizer benefit is quite pronounced.

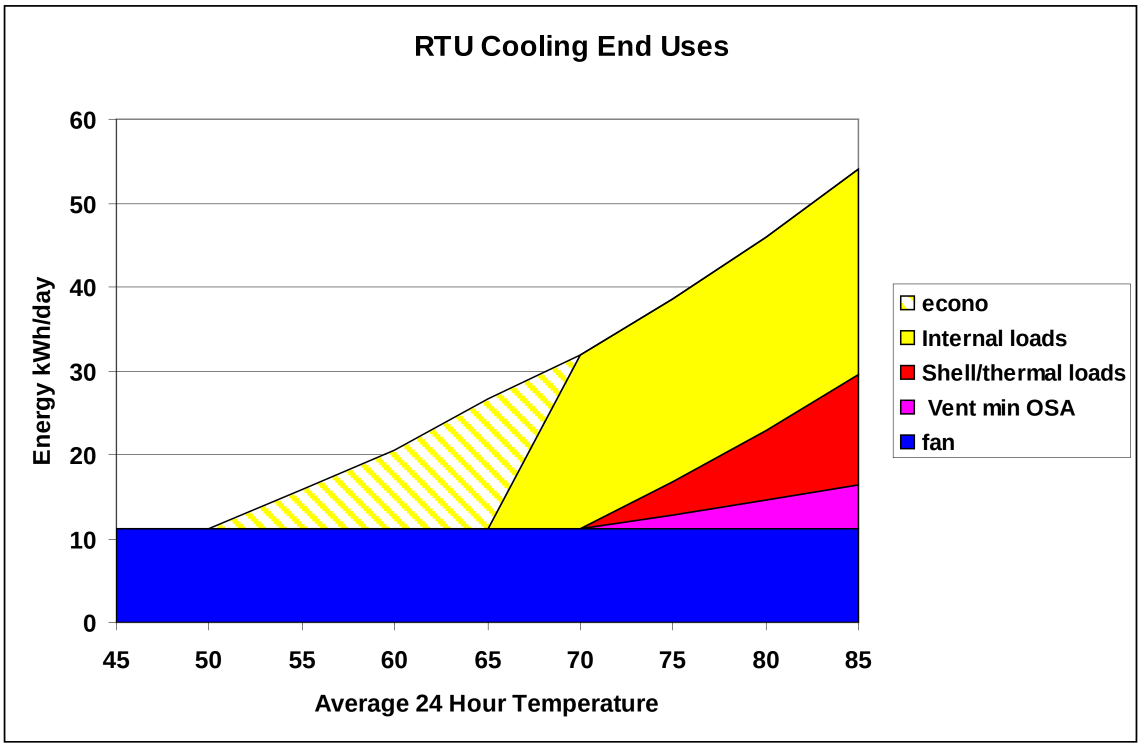
In general, the economizer benefit is proportional to the loading. These figures show that the significantly different gain levels do not alter the slope of the signature, but rather they manifest as a uniform increase in the whole signature. They also show that the conditioned space is behaving as a localized energy balance such that increased internal gain leads to a lowered breakpoint temperature, as is typical in an energy balance.

Building the Physical Model

Using data such as those shown in the previous figures and table yields a Physical Model of the RTU energy use. A key feature of this Physical Model is that it produces an energy signature, which can then be used with a histogram of daily average temperatures in a normal year to produce an estimate of the normalized annual energy use in the typical manner (as used in the RTU protocol).

Figure 39 shows the basic elements of the Physical Model.

Figure 39. RTU Physical Model Energy Components vs. Outdoor Air Temperature

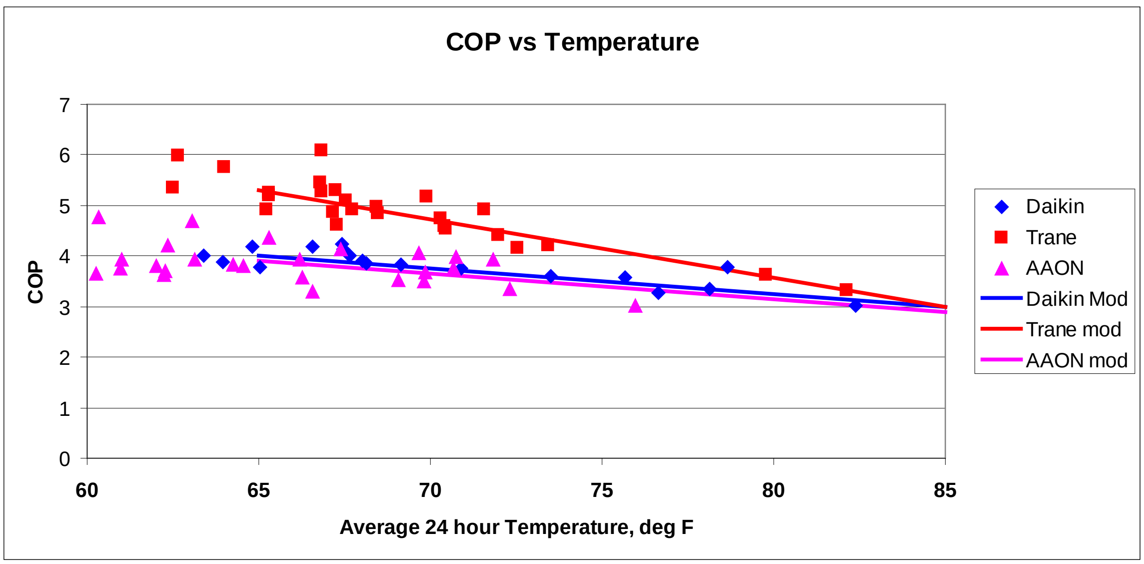


The model readily infers the fan portion from fan power measurement and thermostat settings. The vent from induced outside air, the purple portion, results from cooling introduced ventilation air. This contribution grows as outside air temperature increases and can also be derived from site measurements. The red portion corresponds to temperature-dependent loads from conduction, and the yellow portions correspond to the cooling energy attributable to internal loads. In practice, the parameters for the red and yellow portions are a unique combination. In other words, only one combination of these parameters will fit both the slope and the magnitude of the metered data. In practice, the key item of information necessary to make this model work is knowledge of the RTU COP.

The physical fact for all RTUs is that the COP is principally a function of outdoor temperature, independent of fan power. In the experimental process of the NBIL, the observed COP is also a function of the supply fan power. Therefore, a key part of this model for each different RTU is the function of the RTU COP corrected for the effects of the supply fan. The need to derive a COP corrected for the effects of supply fan power became evident in metering results that showed wide variations in the apparent COP at times when the supply fan power was high. As subsequent empirical experience showed, the ability to estimate changes in COP due to changes in supply fan power is important.

This corrected COP is an unusual presentation since it is an average daily sensible COP for the compressor/condenser activity only. This unusual daily calculation of COP is necessary to interface with the rest of the model (which uses daily averages), and it is necessary to capture the full amount of cooling delivered during the portions of the cycles when the compressor is not active. The model adjusts the thermal output to include the Joule heating from the fan that is inherent in the NBIL thermal output measurements, which are made downstream of the supply fan. Figure 40 shows the resulting RTU COP functions.

Figure 40. COP versus Temperature for Each RTU Based on Lab Testing



Using the Physical Model

The model uses the field site characteristics (such as fan power, schedules, and percentage outdoor air) as well as the COP curve (which the model considers general to all sites) for both the models of the efficient unit and of the code unit. The physical model adjusts the efficient unit model to fit the metered data by changing the parameters for thermal effect and internal gain until the model fits the metered data. When so adjusted, the model of the efficient unit will show an energy signature that the physical model can use to annualize into an estimate of the normalized annual energy use for the efficient unit at the field site.

The physical model then puts the parameters for thermal effect and internal gain at the field site into the code model, which will produce the energy signature for the code RTU operated at the conditions of the field site. The physical model then annualizes this signature into an estimate of the normalized annual energy use of the code unit under the conditions of the field site.

The improvement ratio associated with the efficient RTU relative to the code RTU for this location and these operating conditions is the simple ratio of the annualized energy for the code unit divided by the annual energy for the efficient unit.

Appendix B – ClimaCheck RTU Report

NBI researchers used the ClimaCheck system to analyze performance; the subsequent report provided by ClimaCheck personnel is included here. Note that NEEA is preserving the contributor's original formatting, and the report is presented in its entirety.

Performance Analyses

NBI test three RTUs

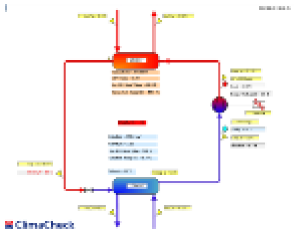
Analyses performed by

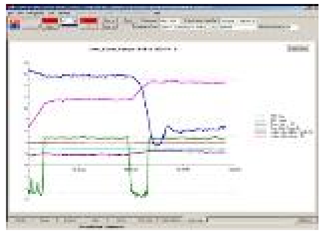
Emerging Energy Solutions

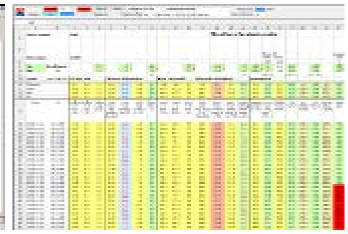
with

ClimaCheck Performance Analyser

Analyzes and visualization of the process is done with ClimaCheck software







SUMMARY

NBI tested three different brands of Roof Top Units (RTUs). Below each one is referred to as RTU1, RTU2 and RTU3 in the order the tests were executed.

ClimaCheck portable Performance Analyzer tested the refrigerant and air side of the RTUs in parallel to NBI's testing process. The tests were done to demonstrate the functionality of ClimaCheck Performance Analyzer and add understanding of the refrigeration cycle performance to the air side performance.

EQUIPMENT

RTU1 Trane Precedent
Nominal Capacity: 5 Tons
Nominal EER: 1
Compressor Type: Scroll
Refigerant Metering Device: TXV
Indoor Blower Motor: Direct Drive
Condenser Fan: Direct drive on during cool

RTU2 Daikin McQuay Rebel
Nominal Capacity: 5 Tons
Nominal EER: 12.7
Compressor Type: Scroll / Inverter
Refrigerant Metering Device: Electronic TXV
Indoor Blower Motor: ECM / VFD
Condenser Fan: ECM /VFD

RTU 3 AAON RQ Series
Nominal Capacity: 5 Tons
Nominal EER: 12.7
Compressor Type: Digital Scroll
Refrigerant Metering Device: TXV
Indoor Blower Motor: ECM / VFD
Condenser Fan: Direct drive on during cool

BACKGROUND

The ClimaCheck measurements were executed by Emerging Energy Solutions ClimaCheck's distributor.

The RTU's have been installed on a "building" with a load that is varied to test RTU operation at different load conditions. The "manual" variation of load will create a load pattern/energy profile that is different from that of most buildings as it is decoupled from ambient climate which in most buildings together with occupancy hours will define the energy consumption profile.

In principle the systems are based on a straight forward standard refrigeration cycle. There are enhanced features installed in several of the systems to increase the energy efficiency of the system.

With current design and control strategy RTU1 and RTU3 were not capable of creating stable conditions for the refrigeration process during the test. The capacity and COP/EER never reach stability due to short cycling but the operation can be evaluated based on the conditions created.

RTU2 has a significantly more suitable design and control for the conditions tested. The system created very stable and good operation over most of the test. Only at some conditions the system became unstable.

It should be noted that the power meters used were clamp-on current transformers with a range of 100Amp which is not optimal for the size of equipment. Fixed transformers of suitable size of for currents below 10Amp direct through would increase accuracy.

The aim with the measurement was to demonstrate the method.

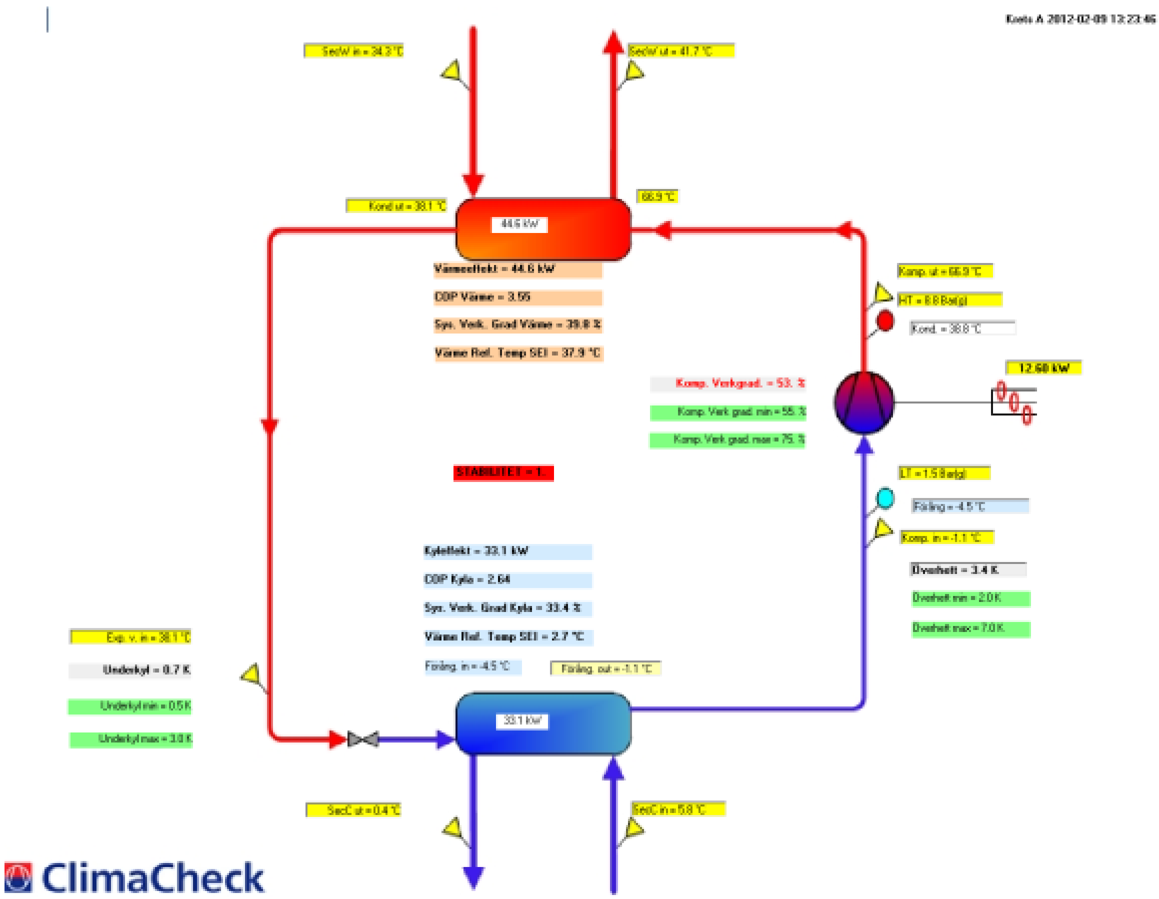


Figure 1, Flow chart of tested system

ClimaCheck Analysis of Refrigeration Process

System and Controls

RTU1 satisfied the load by on-off operation of the compressor. When the thermostat calls for cooling the unit would start and run at full capacity until the space temperature setpoint is reached then shuts off. Capacity of this unit exceeded the load of the space at all times during the test leading to short run times and frequent compressor cycling. The condenser fan would run at full speed every time the compressor came on. The evaporator fan had a fixed speed with no controls. Refrigerant metering was controlled by mechanical Thermal Expansion Valve. RTU1 was only tested for 5 days with the ClimaCheck.

RTU2 had control capability that allowed good control during the tests. The unit would produce as much capacity as needed to condition the space. A variable speed inverter scroll compressor controlled the capacity well from 40% to 100%. This compressor would slowly come to speed while starting. Varying the condenser fan speed resulted in high utilization of the condenser coil. Although the indoor blower motors speed could be varied it was never recorded doing so during testing. An electronic thermal expansion valve kept superheat low and stable resulting in high utilization of the evaporator coil.

A control issue arose when the unit would encounter very low load situations with low ambient conditions. The economizer would open bringing in outside air 65°F or less. A setpoint of 57°F leaving air would cause the compressor to start to cool the air a couple of degrees. This setpoint would be reached almost instantly and the compressor would shut off. When the supply air warmed this cycle would take place over and over. Due to this issue rapid compressor cycling, as much as 15 cycles per hour could take place at certain conditions. This issue could easily be solved with some simple setpoint changes in the controller but were missed when the unit was started up.

RTU3 was special ordered by the end user to be controlled by their existing building controls. For this to happen the manufacturer’s controls were removed and an input output board installed to control the unit. RTU3 has a digital scroll compressor. This compressor is advertised to vary capacity from 10% to 100%. During the test period the compressor capacity was only varied at start up and shut down and only for a few seconds. Capacity of this unit exceeded the load of the space at all times during the test leading to short run times and frequent compressor cycling. The end users controls inhibited the compressor from matching the capacity to the load, essentially running the compressor with an on-off cycle to match the load instead of varying capacity to match the load. The indoor blower motor of this unit is controlled by a VFD but the VFD was set up to run the fan at one speed and can only be changed manually with these controls. The condenser fan would run at full speed every time the compressor came on. Refrigerant metering was controlled by mechanical Thermal Expansion Valve.

The sizing of the systems versus the load makes it difficult to make a good comparison of the three systems. As the systems are significantly over sized for the load RTU1 and RTU3 are cycling on-off with short intervals which normally will decrease performance, comfort and reliability. The ambient temperature also varied significantly over the test periods of the different systems.

The results of these tests would be affected if the load was closer to the capacity of the RTUs.

Due to the low load versus capacity RTU1 and RTU3 operate at very unstable conditions. Even at high load the cycling time is such that the system never stabilizes enough for the internal control to achieve stable operation.

RTU2's ability to adjust capacity according to demand also results in a supply temperature that is kept almost constant regardless of ambient conditions. This should have benefits from a comfort point of view as well as avoiding unnecessary dehumidification. RTU2 controls work well over most of the envelope but at certain loads there are erratic behaviours. It should be possible to avoid these behaviours with adjustments of control.

On-off operation off RTU1 and RTU3 can be expected to create temperature fluctuation decreasing comfort. Due to the low evaporation temperature when the compressor runs full load performance will decrease and dehumidification will increase. This will cause extra energy consumption beyond achieving desired indoor temperature.

During the test of RTU3 the load change in the test period is clearly shown as periods above and below the Energy profile that is correlated only to ambient. The energy profile is shown as a green line following the trend of the kilowatt usage, shown as orange vertical bars. The orange horizontal line at the top of the graph shows the average ambient temperature.

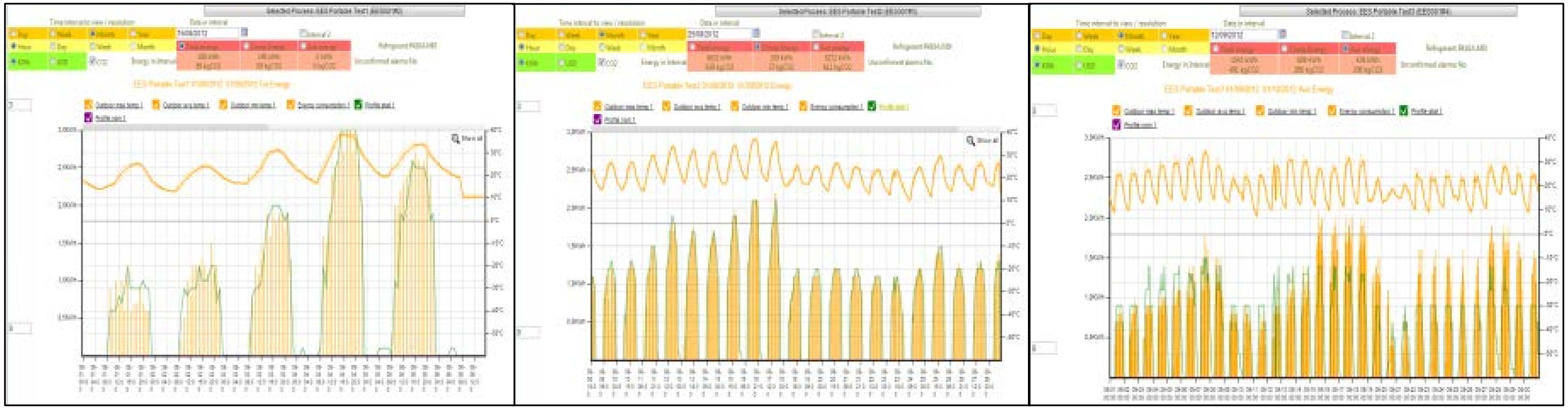


Figure 2, RTU1-3 energy consumption in kWh/h over test period (°C)

Power Profile is in this case not a good comparison of the different objects as the load is manually manipulated. But in normal cases this load profile is relevant to compare the performance of buildings/systems over time and also to benchmark different buildings.

Figure 3, Energy Consumption kWh/h over test period (°F)

Energy Profile as function of ambient

For RTU1 only compressor power was measured. Varying ambient conditions were recorded during the tests (up to approximately 35° C) for all three RTUs but different loads were applied over time which makes comparison over time challenging.

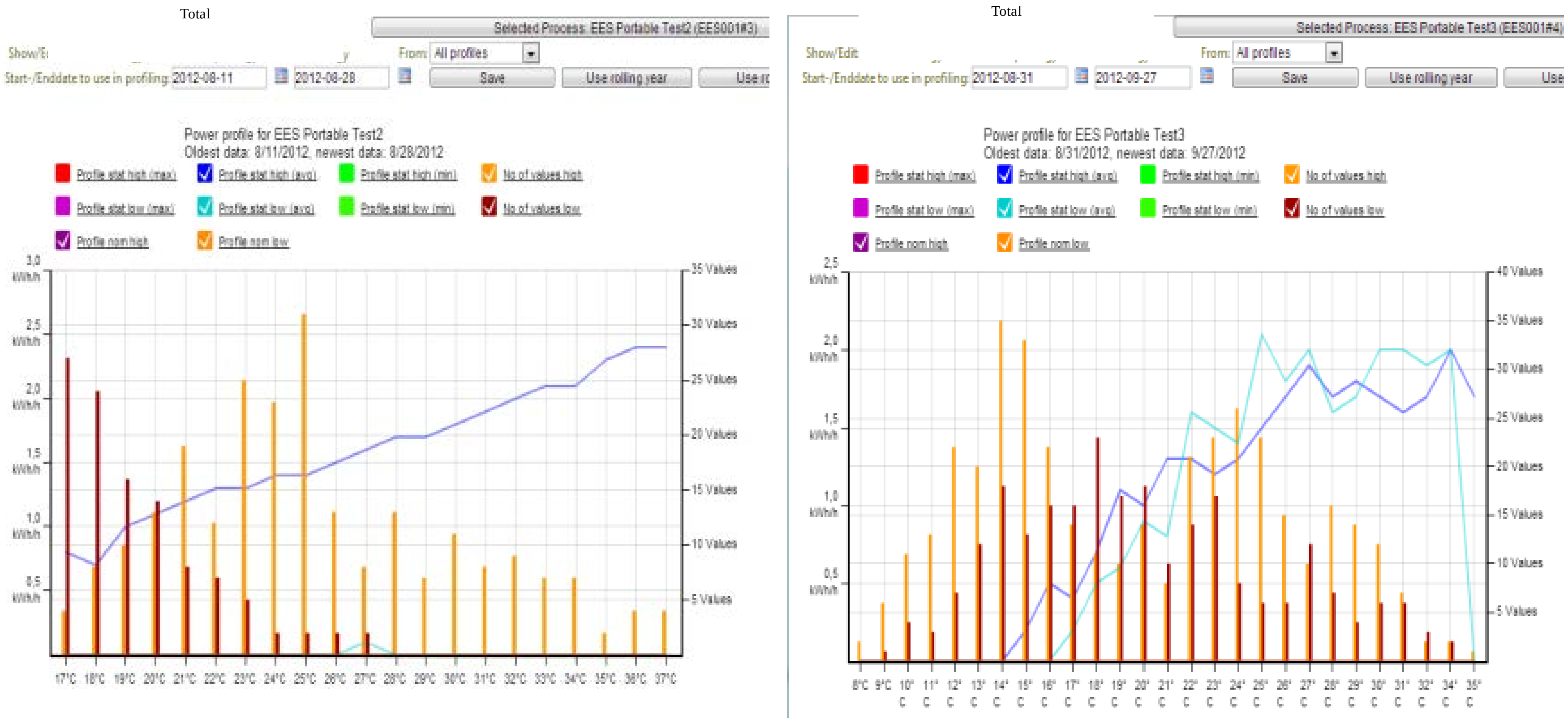


Figure 4, Energy Profile for RTU2 and RTU3

Digital Scroll Compressor

Digital Scroll Compressor Operation

The digital scroll is capable of seamlessly modulating its capacity from 10% to 100%. A normally closed (de-energized) solenoid valve is a key component for achieving modulation. When the solenoid valve is in its normally closed position, the compressor operates at full capacity, or loaded state. When the solenoid valve is energized, the two scroll elements move apart or into the unloaded state. During the unloaded state the compressor motor continues running but since the scrolls are separated there is no compression. During the loaded state the compressor delivers 100% capacity and during the unloaded state the compressor delivers 0% capacity. A cycle consists of one loaded state and one unloaded state. By varying the time of the loaded state and the unloaded state an average capacity is obtained. The lowest achievable capacity is 10% which equates to 1.5 seconds of pumping during one 15-second cycle.

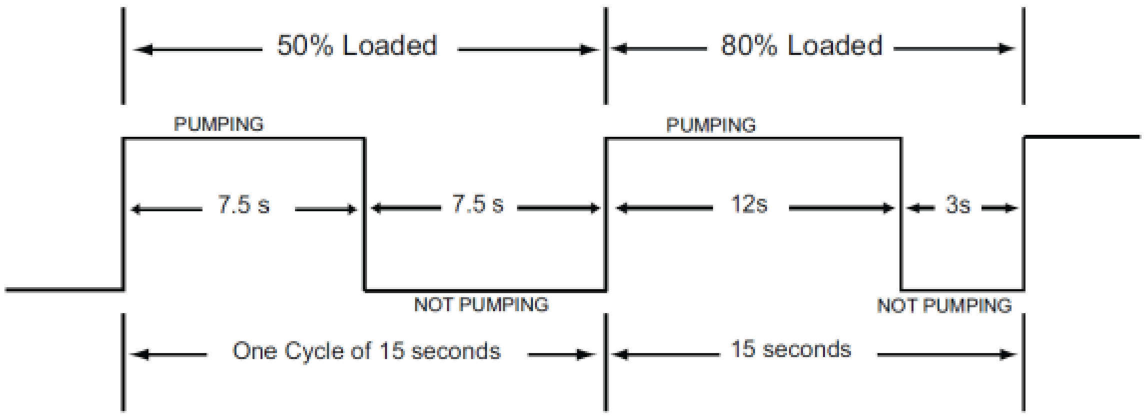


Figure 5, Digital Cycle

An example for the 15-second controller cycle: In any 15-second cycle, if the loaded tin ıe is 5 seconds the average capacity is 66% or if the loaded time is 5 seconds and the unl

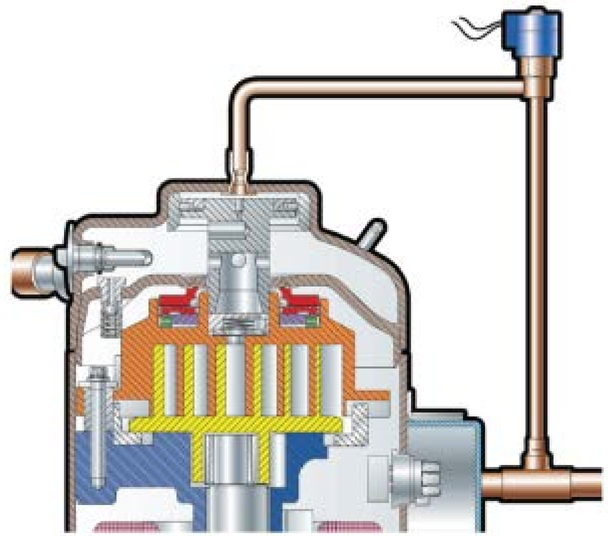


Figure 6, Inner Components of a Digital Scroll Compressor

during that 15 second period is 33%. See Figure 5 for a graphical representation of the digital cycle. For a cut-out profile of the inner components of a digital scroll compressor see Figure 6.

RTU3 Digital Scroll Compressor Cycling

This data was taken at 1-second intervals showing an entire cycle of the digital scroll compressor in RTU3. For the first 15 seconds of start up the controls vary the compressors capacity then loads to 100%. At the end of the cycle the controls attempt to vary the capacity and then shut the compressor down.

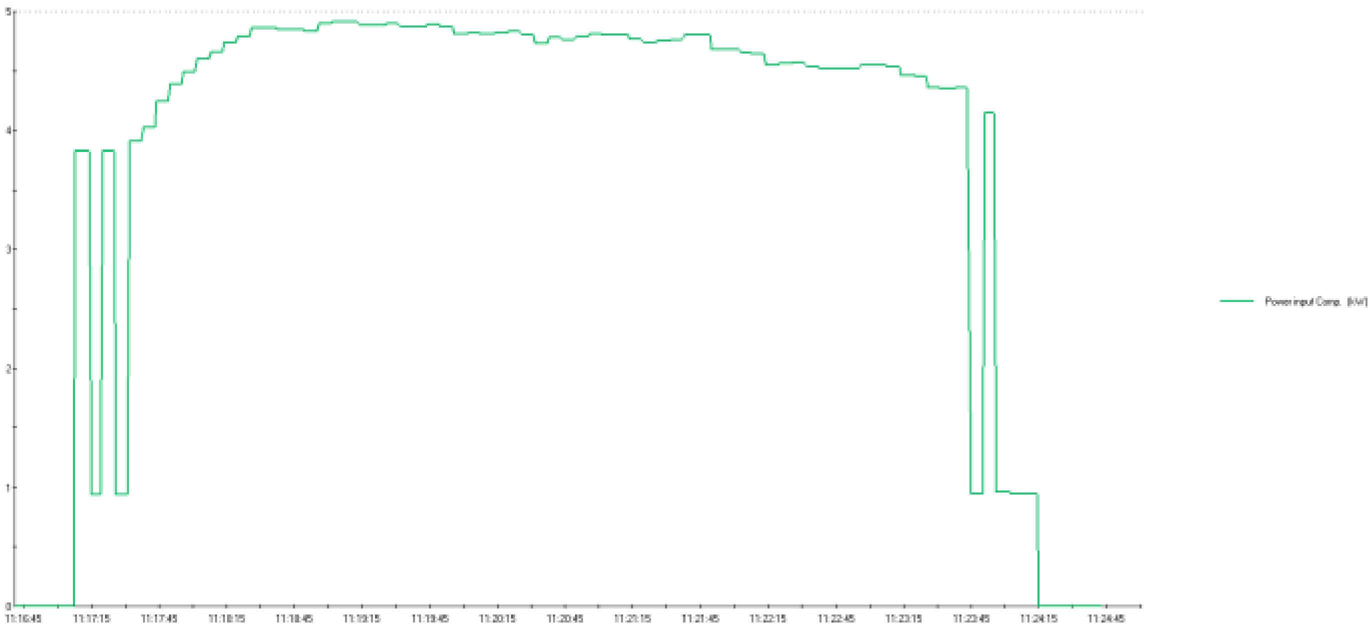


Figure 7, RTU3 One Compressor Cycle

System Evaluation

As only RTU2 is controlled in a way that stable operation is possible at low load conditions there is an uncertainty in data that is dependent on stability for measurement. Such as compressor efficiency, refrigeration circuit COP and capacity. Still these systems and process can be evaluated.

Using an inverter results in a few percent of loss in the inverter but offers a possibility for better utilization of heat exchangers. To vary the capacity also allows for a more free control of fans which offers a saving potential on auxiliary load in a system (fans/pumps).

Evaporator performances are similar for the three units although the different operating modes; on-off versus variable capacity will have an impact due to the difference in capacity (For RTU3 we could not match a high ambient during the high load time frame) so an intermediate temp was selected for the comparison.

Condenser performance is similar for all units.

Table 1. Condenser Performance

Test	Comment	Air on Evap F°	Air off Evap F°	dT air F°	Evap F°	dT Evap air on F°	dT Evap air off F°	Comp in F°	Suction PSIG	Super heat F°
2012-08-04 15:50	RTU1 High ambient	80	57.6	22.4	42.8	37.2	14.8	68.9	125.9	26.1
2012-08-16 14:39	RTU2 High Ambient before	80.6	58.8	21.8	45.8	34.8	13	51.6	133.4	5.8
2012-08-16 14:59	RTU2 High Ambient after	80.8	59.7	21.1	45.8	35	13.9	54.8	133.4	9
2012-09-18 16:04	RTU3 Intermediate Ambient	73	47.8	25.2	36.3	36.7	11.5	39.7	111.2	3.4

Test	Comment	Air off Cond F°	dT air F°	Cond F°	dT Cond air on F°	dT Cond air off F°	Liquid line F°	Discharge PSIG	Sub Cool F°	Comp out F°	Power input kW
2012-08-04 15:50	RTU1 High ambient	118.9	18.9	120.4	20.4	1.5	104	424.4	16.4	182.8	4.50
2012-08-16 14:39	RTU2 High Ambient before	114.6	16	113.2	14.6	-1.4	102.9	385.3	10.3	161	2.33
2012-08-16 14:59	RTU2 High Ambient after	105.8	8.1	105.1	7.4	-0.7	98.6	344.5	6.5	150.6	2.08
2012-09-18 16:04	RTU3 Intermediate Ambient	100.2	13.3	102.2	15.3	2	94.3	330.7	7.9	142	3.80

Refrigerant Charge

RTU1 and RTU3 are not stable resulting in fluctuations in superheat and sub cool making accurate evaluation of charge uncertain but indication is that charge is sufficient for good operation for periods studied.

RTU2 is operating with sufficient with sub cool expected for good functionality.

Expansions Device

RTU1 and RTU3 work with on-off cycles to satisfy the load. During the test under these conditions the cycles are shorter than required for an expansion valve to adjust and stabilize. No operation outside expected has been observed.

RTU2 is working with stable and low superheat resulting in high utilization of evaporator.

Condenser Performance and Flow Rates

RTU1 is operating with the largest dT 20.4°F (lowest flow relative capacity) on air over condenser resulting in the highest condensing temperatures at a given ambient temperature.

RTU2 is operating with a lower dT on air over condenser and this decreases further at change of operation around 2:40 pm August 16. When this happens the power of auxiliary load remains the same so it does not seem to be an increase of fan speed. The air temperature changes from 16°F to 8.1°F resulting in lower condensing. This change improves performance of refrigeration cycle significantly (15%) as compressor power decreases with approximately 0.2 KW.

The cause of change in air temperatures and condensing is unknown to undersigned and the explanation below should be validated by information from test. One logical explanation is an increase of airflow. There is said to be a refrigerant by pass that could open to reduce capacity by bypassing refrigerant passed condenser. This would reduce condensing and dT on air over condenser but it would be expected to have a large impact on evaporator that cannot be seen here. The condensing temperature is controlled by the unit controller as function of outdoor temperature. This saves the maximum amount of energy for the process.

Evaporator Performance and Flow Rate

All systems are operating with similar temperature decrease of air temperature with 21.1°F-25.2°F.
Temperature difference between incoming air and evaporation is around 35°F for all three units.

Compressor

Compressor isentropic efficiency of RTU1 and RTU3 never reaches stability due to the short cycling. Thus they cannot be fully evaluated unless systems are provoked to run for a longer time. This would ensure that oil has reached its working temperature. When the compressor oil reaches this temperature dissolved refrigerant has evaporated to its balance point and discharge temperature has stabilized.

For RTU2 the compressor efficiency is what is expected of a good compressor during the whole test sequence.

Measurements performed by: ***Joel Klobas* - Emerging Energy Solutions**
Report done by: ***Klas Berglöf* - MSc and CEO ClimaCheck Sweden AB**

Appendices

Measured Points Standard System

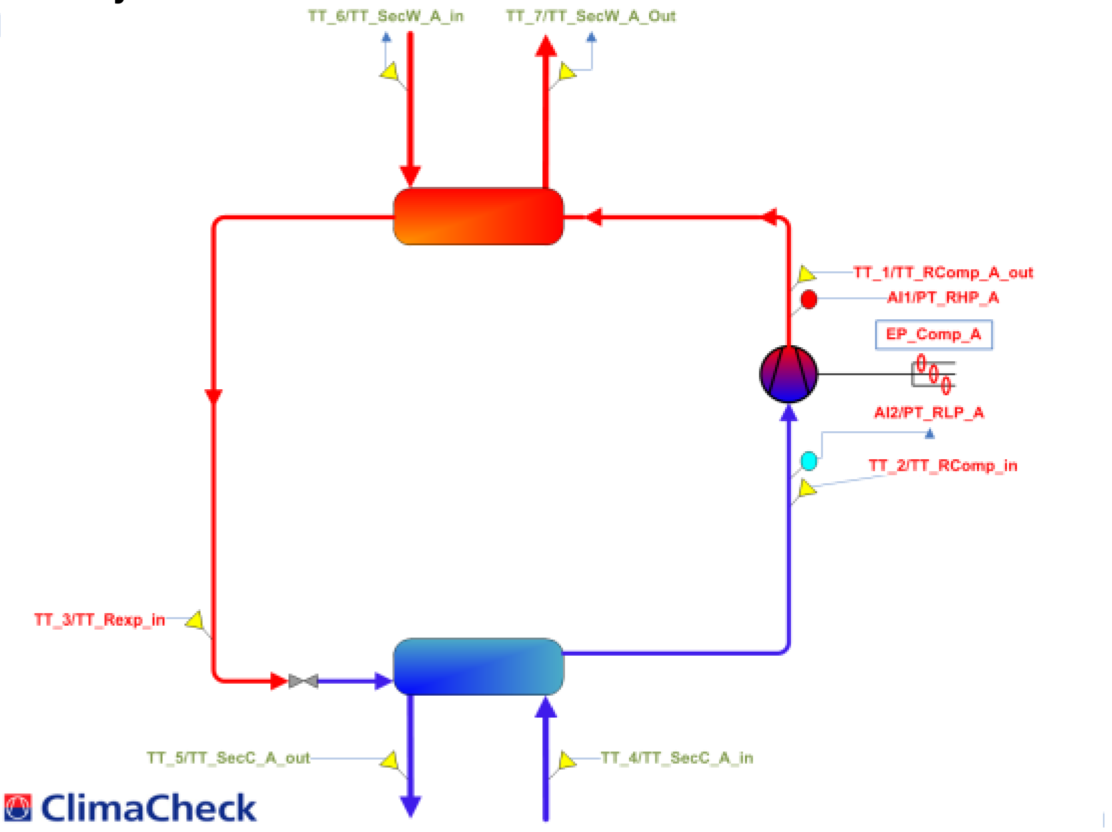


Figure 8, Standard sensor mounting

Measurements
RTU1

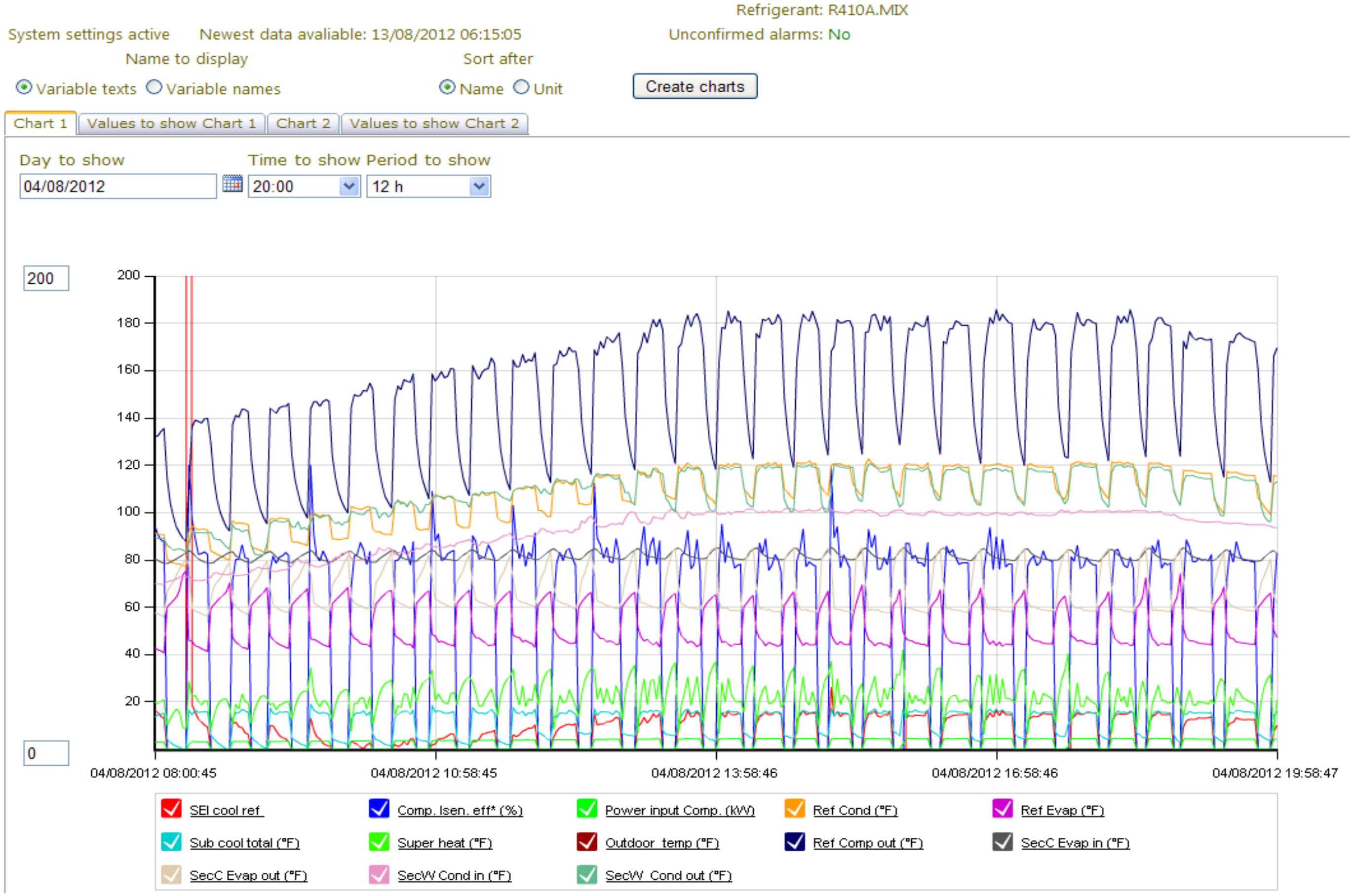


Figure 9, RTU1 operates with on-off and does not reach stable operation even on a 100° day (°F)

RTU1 is working with low load versus maximum capacity and is continuously cycling on off even at high ambient temperatures. The system never reaches stable operation.

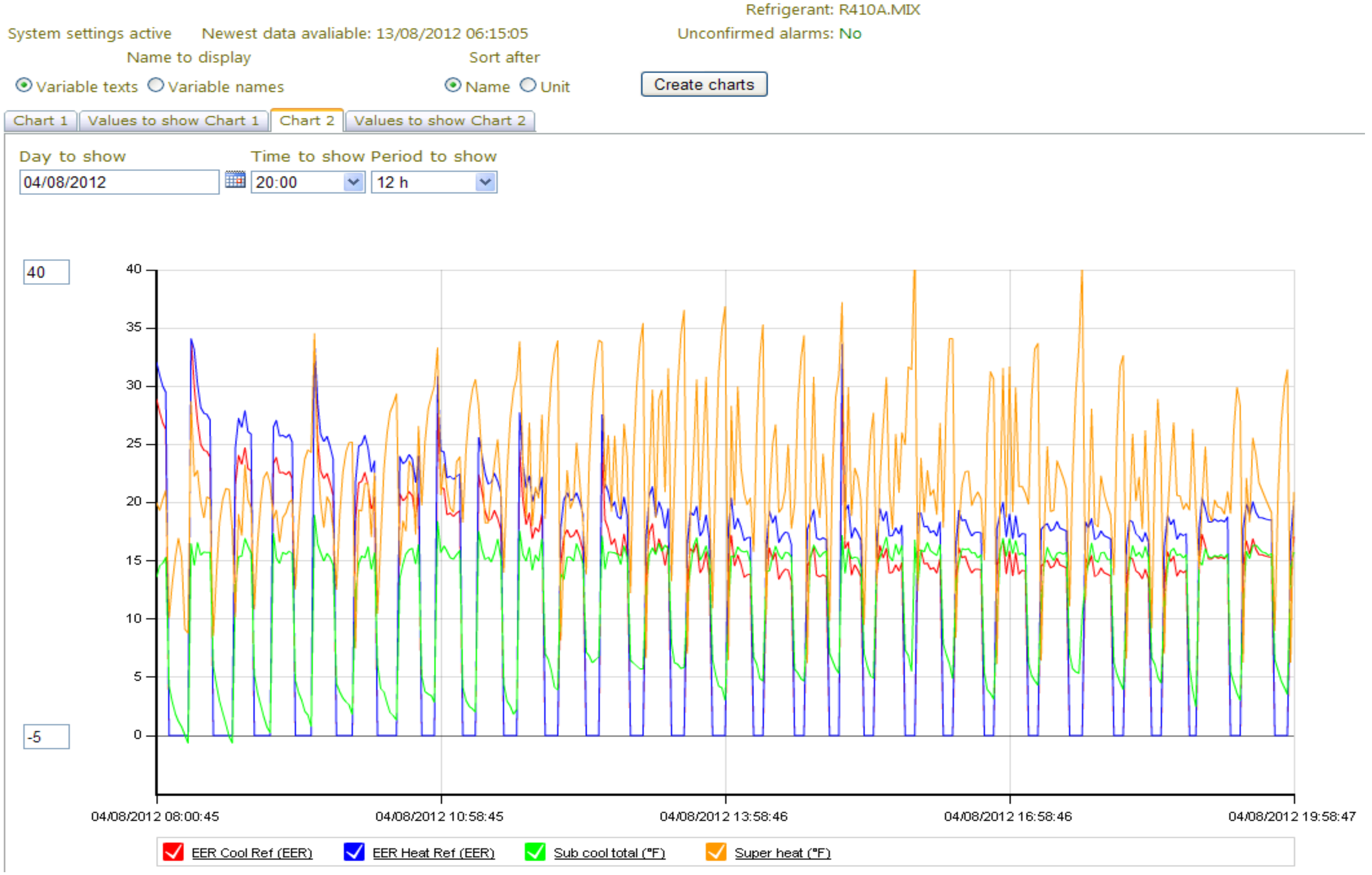


Figure 10, RTU1 Super heat and sub cool during On-Off operation (°F)

Selected Process: EES Portable Test1 (EES001#2)

This site uses the

Refrigerant: R410A.MIX Unconfirmed alarms: No

Date to show | No of values per page | Max-Min-Avg calculated on the last | Time Search | Search | Export functions

User settings active 04/12/2012 | 40 | 60 min | 2012-08-04 16:00 | Page to XL | All to XL | Page to PDF

older data> oldest data>>

Time	SecC Evap in (°F)	SecC Evap out (°F)	Ref Low press. (psi)	Ref Evap (°F)	Ref Comp in (°F)	Super heat (°F)	SecW Cond out (°F)	Ref High press. (psi)	Ref Cond (°F)	Ref Exp. Valve in (°F)	Sub cool total (°F)	Ref Comp out (°F)	Comp. Isen. eff* (%)	Power input Comp. (kW)	EER Cool Ref (EER)	Cap. Cool (btu/h)	EER Heat Ref (EER)	Cap. Heat (btu/h)	Energy Comp (kWh) ( )	Outdoor temp (°F)	SEI cool ref	SEI heat Ref	Ref Temp Cool (°F)
2012-08-04 16:09:46	80.6	58.3	129.51	44.7	64.6	19.8	118.4	417.69	119.8	104.4	15.3	179.6	77.9	4.4	14.02	61,893.7	17.20	75,917.0	20,692	99.5	14.4	0.0	80.6
2012-08-04 16:08:51	80.8	58.6	129.80	44.8	65.5	20.6	118.6	417.83	119.8	103.8	15.9	179.8	78.7	4.4	14.26	62,610.2	17.44	76,531.2	20,692	99.3	14.3	0.0	80.8
2012-08-04 16:07:46	81.0	58.6	130.09	44.9	66.0	21.0	118.4	419.57	120.1	103.8	16.2	180.3	79.2	4.4	14.33	63,565.6	17.50	77,657.1	20,692	99.3	14.3	0.0	81.0
2012-08-04 16:06:46	81.1	58.8	130.24	45.0	67.8	22.8	118.2	420.59	120.3	103.6	16.6	180.5	82.0	4.4	14.81	65,817.5	17.98	79,874.9	20,692	99.1	14.5	0.0	81.1
2012-08-04 16:05:46	81.3	58.8	129.80	44.8	69.4	24.6	117.5	414.93	119.3	103.1	16.1	178.5	86.0	4.4	15.73	69,093.0	18.90	83,014.0	20,692	98.6	14.7	0.0	81.3
2012-08-04 16:04:46	81.9	59.4	131.98	45.7	64.9	19.2	118.4	419.14	120.1	104.2	15.8	176.9	80.5	4.4	14.74	64,759.8	17.91	78,714.8	20,692	99.3	13.9	0.0	81.9
2012-08-04 16:03:46	82.2	59.5	132.99	46.1	66.0	19.9	118.8	421.17	120.4	104.7	15.6	178.0	80.4	4.5	14.71	65,646.9	17.88	79,806.7	20,692	99.5	13.8	0.0	82.2
2012-08-04 16:02:46	82.9	59.5	132.99	46.1	70.0	23.8	118.4	421.02	120.4	104.4	15.9	178.2	86.9	4.5	15.93	71,208.4	19.11	85,402.4	20,692	99.5	14.3	0.0	82.9
2012-08-04 16:01:46	83.8	59.4	134.73	46.8	55.2	8.3	118.4	422.91	120.7	106.2	14.5	170.2	73.7	4.4	13.55	59,983.0	16.72	74,040.4	20,692	99.5	11.4	0.0	83.8
2012-08-04 16:00:46	84.7	60.4	134.88	46.9	59.4	12.4	118.9	421.60	120.5	107.6	12.8	166.3	87.1	4.4	15.90	70,423.7	19.07	84,481.1	20,691	99.7	12.8	0.0	84.7
2012-08-04 15:59:46	85.1	66.2	145.18	50.9	57.2	6.2	117.5	424.07	120.9	112.1	8.7	163.0	78.1	4.4	14.81	65,476.3	17.98	79,533.7	20,691	99.5	11.5	0.0	85.1
2012-08-04 15:58:46	85.3	85.1	143.14	50.1	92.3	42.1	103.8	407.10	117.9	100.9	16.8	145.0	-20.0	2.4	0.00	0.0	0.00	0.0	20,691	100.8	-670.2	0.0	85.3
2012-08-04 15:57:46	84.4	85.3	203.48	70.7	99.9	29.1	101.7	345.75	105.9	100.6	5.2	124.9	0.0	0.0	0.00	0.0	0.00	0.0	20,691	100.9	0.0	0.0	84.4
2012-08-04 15:56:46	83.7	84.2	193.76	67.6	99.1	31.4	102.4	348.80	106.5	100.9	5.5	128.5	0.0	0.0	0.00	0.0	0.00	0.0	20,691	101.3	0.0	0.0	83.7
2012-08-04 15:55:46	82.6	83.3	182.59	64.1	97.5	33.3	104.0	355.61	107.9	101.5	6.3	132.8	0.0	0.0	0.00	0.0	0.00	0.0	20,691	101.3	0.0	0.0	82.6
2012-08-04 15:54:46	81.9	82.2	178.39	62.7	94.5	31.7	106.5	360.11	108.8	101.8	6.9	138.0	0.0	0.0	0.00	0.0	0.00	0.0	20,691	100.9	0.0	0.0	81.9
2012-08-04 15:53:46	80.8	80.6	175.63	61.8	90.7	28.8	109.2	363.59	109.5	102.4	7.1	144.5	0.0	0.0	0.00	0.0	0.00	0.0	20,691	100.8	0.0	0.0	80.8
2012-08-04 15:52:46	80.1	76.6	173.17	61.0	86.0	24.9	112.1	366.93	110.2	102.7	7.4	153.3	0.0	0.0	0.00	0.0	0.00	0.0	20,691	100.6	0.0	0.0	80.1
2012-08-04 15:51:46	79.9	68.0	170.70	60.1	80.1	19.9	115.3	371.71	111.2	103.1	8.0	166.6	0.0	0.0	0.00	0.0	0.00	0.0	20,691	100.6	0.0	0.0	79.9
2012-08-04 15:50:46	80.1	57.6	125.02	42.8	68.9	26.0	118.9	421.31	120.5	104.0	16.3	182.8	85.8	4.5	14.88	66,772.8	18.05	81,035.0	20,691	100.0	16.1	0.0	80.1
2012-08-04 15:49:46	80.1	57.9	129.22	44.6	63.7	19.0	118.4	428.56	121.7	105.3	16.4	180.9	79.0	4.5	13.82	62,610.2	16.99	76,974.7	20,691	100.2	15.1	0.0	80.1
2012-08-04 15:48:47	80.1	57.7	127.92	44.0	64.9	20.9	117.5	412.90	118.9	103.3	15.6	179.8	78.0	4.4	14.13	61,416.0	17.30	75,234.6	20,691	98.8	14.4	0.0	80.1

Figure 11, RTU1 Table with data unstable operation result in uncertainty of performance (°F)

RTU2

RTU2 is equipped with variable speed drive for the compressor and operate well over almost the whole test only at some low load condition the control is not able to avoid erratic operation.

Figure 12, RTU2 Control following change in ambient condition showing also change of condenser behaviour 14:40 (°F)

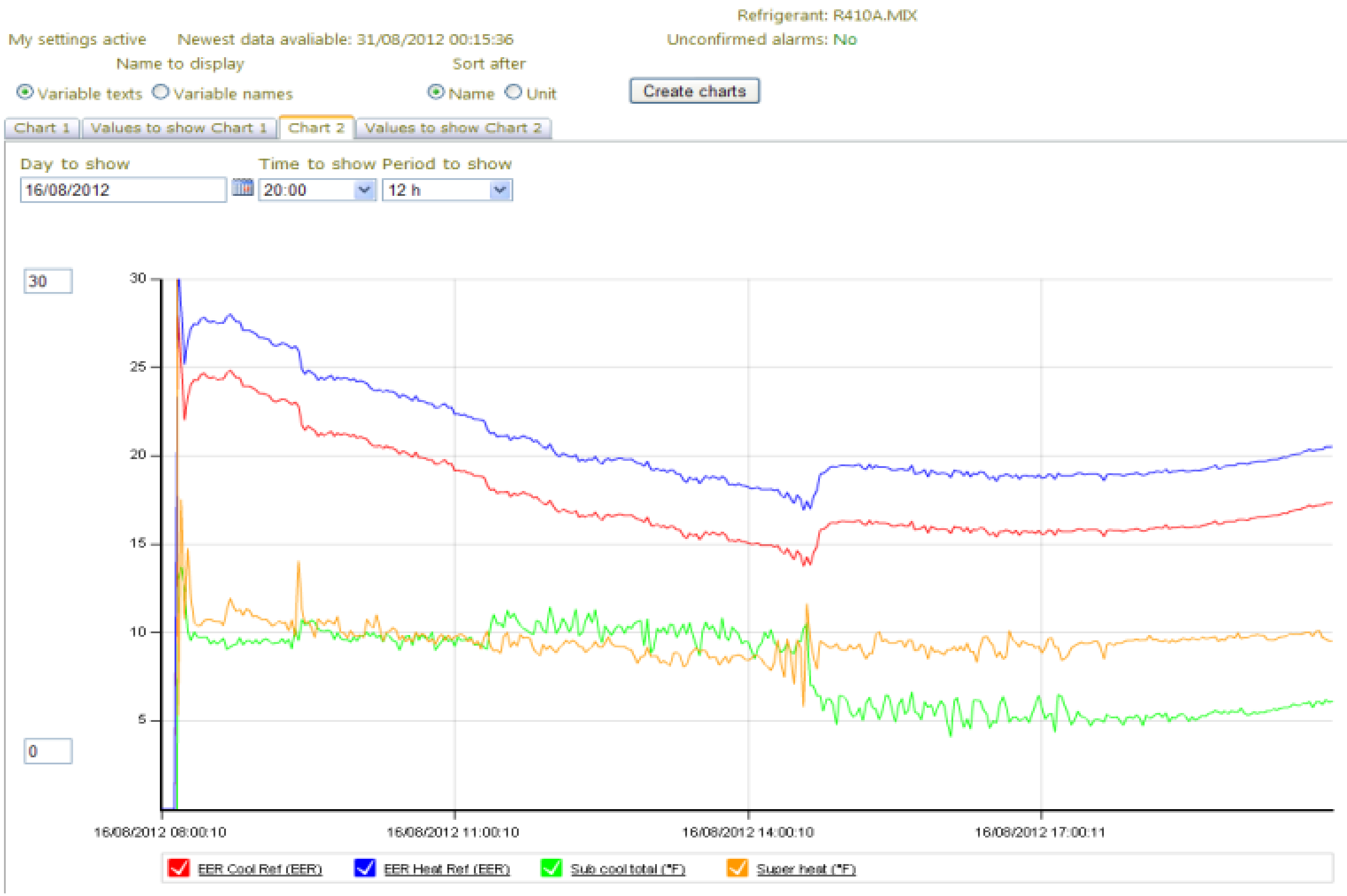


Figure 13, RTU2 superheat and sub cool (°F)

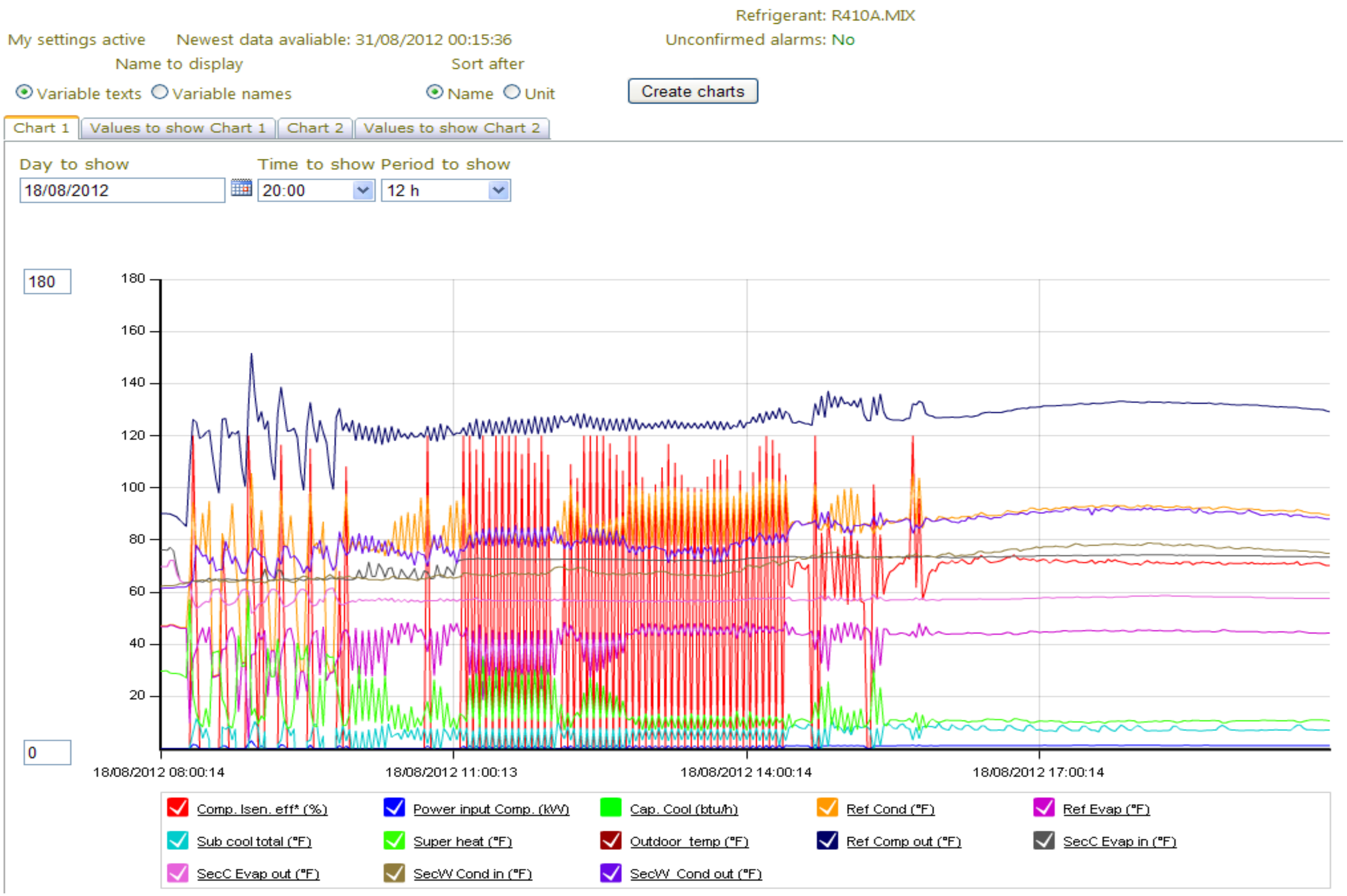


Figure 14, RTU2 excessive short cycling at low load and low ambient (control issue resolved)

Refrigerant: R410A.MIX Unconfirmed alarms: No

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User settings active 04/12/2012 | 40 | 60 min | 2012-08-16 16:00 | Page to XL | All to XL | Page to PDF

older data> oldest data>>

Time	SecC Evap in (°F)	SecC Evap out (°F)	Ref Low press. (psi)	Ref Evap (°F)	Ref Comp in (°F)	Super heat (°F)	SecW Cond out (°F)	Ref High press. (psi)	Ref Cond (°F)	Ref Exp. Valve in (°F)	Sub cool total (°F)	Ref Comp out (°F)	Comp. Isen. eff* (%)	Power input Comp. (kW)	EER Cool Ref (EER)	Cap. Cool (btu/h)	EER Heat Ref (EER)	Cap. Heat (btu/h)	Energy Comp (kWh) ( )	Outdoor temp (°F)	SEI cool ref	SEI heat Ref	Ref Temp Cool (°F)	Ref Temp Heat (°F)
2012-08-16 16:09:11	80.6	59.7	131.11	45.3	54.5	9.1	106.3	343.72	105.5	99.5	5.9	151.7	69.9	2.1	15.97	33,198.8	19.14	39,818.0	20,891	97.9	24.5	0.0	70.2	97.9
2012-08-16 16:08:11	81.0	59.7	130.96	45.3	54.3	8.9	106.3	343.14	105.4	99.0	6.3	151.7	69.4	2.1	15.93	33,608.2	19.11	40,295.7	20,891	97.9	24.3	0.0	70.3	97.9
2012-08-16 16:07:11	80.8	59.9	131.25	45.4	54.5	9.0	106.7	343.58	105.5	99.0	6.4	151.9	69.4	2.1	15.93	33,608.2	19.11	40,295.7	20,891	98.4	24.7	0.0	70.3	98.4
2012-08-16 16:06:11	80.8	59.7	131.25	45.4	54.7	9.2	106.7	345.75	105.9	99.7	6.1	152.1	70.4	2.1	16.00	34,222.4	19.18	41,012.2	20,891	98.4	24.9	0.0	70.3	98.4
2012-08-16 16:05:11	81.0	59.9	130.82	45.2	54.7	9.4	106.3	342.85	105.3	99.1	6.1	151.9	69.8	2.1	16.00	33,744.7	19.18	40,432.2	20,891	98.4	24.7	0.0	70.4	98.4
2012-08-16 16:04:11	81.1	59.9	131.69	45.6	54.7	9.0	107.4	345.32	105.8	101.7	4.1	152.4	69.2	2.1	15.59	32,925.8	18.77	39,613.3	20,891	98.6	24.2	0.0	70.5	98.6
2012-08-16 16:03:11	80.8	59.9	131.40	45.4	54.7	9.2	106.9	346.48	106.1	100.4	5.5	152.1	70.7	2.1	15.97	33,642.3	19.14	40,364.0	20,891	98.4	24.8	0.0	70.3	98.4
2012-08-16 16:02:11	81.0	59.9	131.69	45.6	54.7	9.0	107.2	346.48	106.1	101.1	4.8	152.2	70.0	2.2	15.80	34,085.9	18.97	40,944.0	20,891	98.6	24.6	0.0	70.4	98.6
2012-08-16 16:01:11	81.0	59.9	131.54	45.5	54.5	8.9	107.1	347.35	106.3	100.8	5.4	152.1	70.5	2.1	15.87	33,983.5	19.04	40,773.4	20,891	98.6	24.7	0.0	70.4	98.6
2012-08-16 16:00:11	81.0	59.7	131.54	45.5	54.3	8.7	106.9	346.19	106.0	100.0	5.9	152.1	69.8	2.1	15.83	33,847.0	19.00	40,636.9	20,891	98.8	24.9	0.0	70.3	98.8
2012-08-16 15:59:11	80.8	59.7	131.69	45.6	54.5	8.9	107.1	346.91	106.1	100.0	6.0	152.2	70.0	2.2	15.87	34,393.0	19.04	41,285.2	20,891	98.8	25.0	0.0	70.3	98.8
2012-08-16 15:58:11	80.8	59.9	131.69	45.6	54.5	8.8	107.1	345.75	105.9	99.7	6.1	152.2	69.4	2.1	15.83	33,369.4	19.00	40,056.9	20,891	98.8	24.9	0.0	70.3	98.8
2012-08-16 15:57:11	80.8	59.9	131.69	45.6	54.5	8.9	107.1	348.36	106.5	99.9	6.5	152.4	70.4	2.1	15.90	33,847.0	19.07	40,602.8	20,891	98.8	25.0	0.0	70.3	98.8
2012-08-16 15:56:11	81.0	59.9	131.83	45.6	54.7	9.0	107.4	346.62	106.1	100.2	5.8	152.4	69.6	2.1	15.80	33,812.9	18.97	40,602.8	20,891	98.8	24.8	0.0	70.4	98.8
2012-08-16 15:55:10	81.1	60.1	131.40	45.4	54.7	9.2	107.1	347.06	106.2	99.7	6.4	152.2	70.7	2.1	16.00	34,085.9	19.18	40,841.6	20,891	98.8	24.9	0.0	70.6	98.8
2012-08-16 15:54:11	81.0	59.9	131.98	45.7	54.7	8.9	107.6	352.71	107.4	102.0	5.2	152.6	72.1	2.2	15.90	34,665.9	19.07	41,558.2	20,891	98.8	24.9	0.0	70.4	98.8
2012-08-16 15:53:11	81.0	59.9	131.54	45.5	54.9	9.3	107.2	348.07	106.4	101.1	5.2	152.2	71.2	2.1	15.93	34,120.0	19.11	40,909.9	20,891	98.6	24.8	0.0	70.4	98.6
2012-08-16 15:52:11	80.8	59.9	131.25	45.4	54.7	9.2	106.7	345.46	105.9	99.9	5.9	151.9	70.5	2.1	16.00	33,778.8	19.18	40,500.4	20,891	98.4	24.9	0.0	70.3	98.4
2012-08-16 15:51:10	81.0	59.9	131.54	45.5	54.3	8.7	106.9	344.45	105.7	99.7	5.9	152.1	69.0	2.1	15.76	33,437.6	18.94	40,159.2	20,891	98.8	24.7	0.0	70.4	98.8
2012-08-16 15:50:11	81.1	60.1	132.12	45.8	54.3	8.5	107.6	346.62	106.1	100.2	5.8	152.4	68.7	2.1	15.63	33,301.1	18.80	40,056.9	20,891	99.1	24.7	0.0	70.6	99.1
2012-08-16 15:49:11	81.3	60.1	132.70	46.0	54.5	8.5	108.1	354.31	107.7	101.5	6.1	153.0	71.2	2.2	15.76	34,563.6	18.94	41,489.9	20,891	99.1	24.8	0.0	70.7	99.1
2012-08-16 15:48:11	81.0	59.9	131.83	45.6	54.9	9.2	107.2	349.81	106.8	100.6	6.1	152.4	71.6	2.1	16.04	34,290.6	19.21	41,080.5	20,891	98.8	25.1	0.0	70.4	98.8

Figure 15, RTU2 Test part load (°F)

Refrigerant: R410A.MIX   Unconfirmed alarms: No

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User settings active   04/12/2012   40   60 min   2012-08-16 14:50   Page to XL   All to XL   Page to PDF

older data> oldest data>>

Time	SecC Evap in (°F)	SecC Evap out (°F)	Ref Low press. (psi)	Ref Evap (°F)	Ref Comp in (°F)	Super heat (°F)	SecW Cond out (°F)	Ref High press. (psi)	Ref Cond (°F)	Ref Exp. Valve in (°F)	Sub cool total (°F)	Ref Comp out (°F)	Comp. Isen. eff* (%)	Power input Comp. (kW)	EER Cool Ref (EER)	Cap. Cool (btu/h)	EER Heat Ref (EER)	Cap. Heat (btu/h)	Energy Comp (kWh) ( )	Outdoor temp (°F)	SEI cool ref	SEI heat Ref	Ref Temp Cool (°F)	Ref Temp Heat (°F)
2012-08-16 14:59:11	80.8	59.7	132.27	45.8	54.9	8.9	105.8	341.55	105.0	98.6	6.3	150.6	69.7	2.1	16.28	33,881.2	19.45	40,466.3	20,889	97.7	24.7	0.0	70.3	97.7
2012-08-16 14:58:10	81.0	59.5	132.12	45.8	54.9	9.0	105.8	341.84	105.1	98.6	6.4	150.6	70.0	2.1	16.31	34,597.7	19.48	41,319.3	20,889	97.7	24.8	0.0	70.3	97.7
2012-08-16 14:57:10	81.0	59.7	131.98	45.7	54.9	9.1	105.6	339.66	104.6	98.1	6.5	150.6	69.1	2.1	16.28	33,983.5	19.45	40,602.8	20,889	97.7	24.6	0.0	70.3	97.7
2012-08-16 14:56:11	80.8	59.7	132.41	45.9	55.0	9.1	106.2	342.85	105.3	98.8	6.4	151.0	70.0	2.1	16.28	34,188.2	19.45	40,841.6	20,889	97.7	24.7	0.0	70.3	97.7
2012-08-16 14:55:11	80.8	59.7	132.12	45.8	55.0	9.2	106.0	343.43	105.4	99.5	5.8	150.8	70.9	2.1	16.34	34,973.0	19.52	41,762.9	20,889	97.7	24.8	0.0	70.3	97.7
2012-08-16 14:54:10	80.6	59.7	131.83	45.6	55.0	9.4	105.4	339.23	104.6	99.7	4.8	150.6	69.5	2.1	16.17	33,642.3	19.35	40,261.6	20,889	97.5	24.5	0.0	70.2	97.5
2012-08-16 14:53:12	80.6	59.5	131.54	45.5	54.9	9.3	104.9	336.76	104.0	98.4	5.5	150.3	68.8	2.1	16.24	33,608.2	19.41	40,193.4	20,889	97.5	24.7	0.0	70.1	97.5
2012-08-16 14:52:11	80.4	59.5	131.83	45.6	54.9	9.2	105.1	337.63	104.2	98.2	5.9	150.4	68.7	2.1	16.21	33,574.1	19.38	40,159.2	20,889	97.3	24.6	0.0	70.0	97.3
2012-08-16 14:51:10	80.6	59.5	131.83	45.6	54.9	9.2	105.3	339.66	104.6	98.2	6.3	150.8	69.1	2.1	16.21	33,881.2	19.38	40,534.6	20,889	97.3	24.5	0.0	70.1	97.3
2012-08-16 14:50:10	80.8	59.5	131.98	45.7	54.9	9.1	105.3	340.24	104.8	98.4	6.2	151.0	69.0	2.1	16.17	33,778.8	19.35	40,398.1	20,888	97.3	24.3	0.0	70.2	97.3
2012-08-16 14:49:11	80.8	59.5	131.83	45.6	54.9	9.1	105.1	336.03	103.9	97.5	6.3	151.0	67.1	2.1	16.00	32,823.4	19.18	39,340.4	20,888	97.2	23.9	0.0	70.2	97.2
2012-08-16 14:48:11	80.6	59.7	131.98	45.7	55.0	9.3	105.6	338.50	104.4	98.1	6.2	151.5	67.6	2.1	15.97	33,369.4	19.14	39,988.6	20,888	97.3	24.0	0.0	70.2	97.3
2012-08-16 14:47:10	80.6	59.5	131.25	45.4	55.0	9.6	104.9	339.66	104.6	97.9	6.7	151.5	68.9	2.1	16.14	33,369.4	19.31	39,954.5	20,888	97.2	24.2	0.0	70.1	97.2
2012-08-16 14:46:10	80.6	59.5	131.69	45.6	55.0	9.4	105.3	339.37	104.6	99.0	5.5	151.9	67.9	2.1	15.83	33,267.0	19.00	39,954.5	20,888	97.2	23.8	0.0	70.1	97.2
2012-08-16 14:45:11	80.6	59.4	131.40	45.4	55.0	9.5	105.1	339.08	104.5	98.8	5.6	151.9	68.0	2.1	15.87	33,028.2	19.04	39,613.3	20,888	97.2	23.9	0.0	70.0	97.2
2012-08-16 14:44:10	80.6	59.4	130.96	45.2	54.9	9.5	104.5	335.02	103.7	97.2	6.4	151.9	66.4	2.0	15.80	32,209.3	18.97	38,692.1	20,888	97.0	23.6	0.0	70.0	97.0
2012-08-16 14:43:10	80.6	59.0	131.54	45.5	54.5	8.9	104.9	336.32	103.9	97.5	6.3	152.4	65.1	2.1	15.46	32,345.8	18.63	38,965.0	20,888	97.2	23.4	0.0	69.8	97.2
2012-08-16 14:42:10	80.6	58.1	132.27	45.8	53.8	7.9	105.3	335.02	103.7	97.2	6.4	153.3	61.7	2.1	14.88	30,639.8	18.05	37,156.7	20,888	97.3	23.1	0.0	69.4	97.3
2012-08-16 14:41:11	80.6	58.1	130.53	45.1	53.8	8.6	105.6	339.52	104.6	97.0	7.5	154.8	63.5	2.3	14.84	34,017.6	18.02	41,285.2	20,888	97.5	23.2	0.0	69.4	97.5
2012-08-16 14:40:10	80.8	58.1	130.96	45.2	54.0	8.6	106.5	341.98	105.1	98.1	7.0	156.2	62.7	2.3	14.50	33,062.3	17.67	40,295.7	20,888	97.9	22.8	0.0	69.4	97.9
2012-08-16 14:39:10	80.6	58.5	129.37	44.6	54.1	9.4	106.2	344.01	105.6	97.5	7.9	157.6	63.4	2.3	14.43	33,369.4	17.61	40,705.2	20,888	98.1	22.8	0.0	69.5	98.1
2012-08-16 14:38:11	81.0	58.8	129.08	44.5	54.1	9.6	108.9	342.27	105.2	98.1	7.0	159.8	60.7	2.3	13.82	31,629.2	16.99	38,896.8	20,888	98.6	21.9	0.0	69.9	98.6
2012-08-16 14:37:11	81.0	59.2	130.67	45.2	54.9	9.6	113.5	377.08	112.2	102.0	10.1	163.6	68.7	2.3	13.99	32,209.3	17.16	39,511.0	20,888	98.6	22.1	0.0	70.1	98.6
2012-08-16 14:36:10	81.0	59.5	126.76	43.5	55.2	11.6	112.6	372.73	111.3	100.8	10.5	163.9	71.0	2.3	14.26	32,106.9	17.44	39,238.0	20,888	98.6	22.4	0.0	70.3	98.6
2012-08-16 14:35:10	81.0	59.5	124.87	42.7	54.7	11.9	113.5	374.03	111.6	101.1	10.4	163.2	73.7	2.3	14.50	33,369.4	17.67	40,671.0	20,888	98.6	22.8	0.0	70.3	98.6

Figure 16, RTU2 Test part load (°F)

RTU3

RTU3 is operating with short cycles similar to that of RTU1.

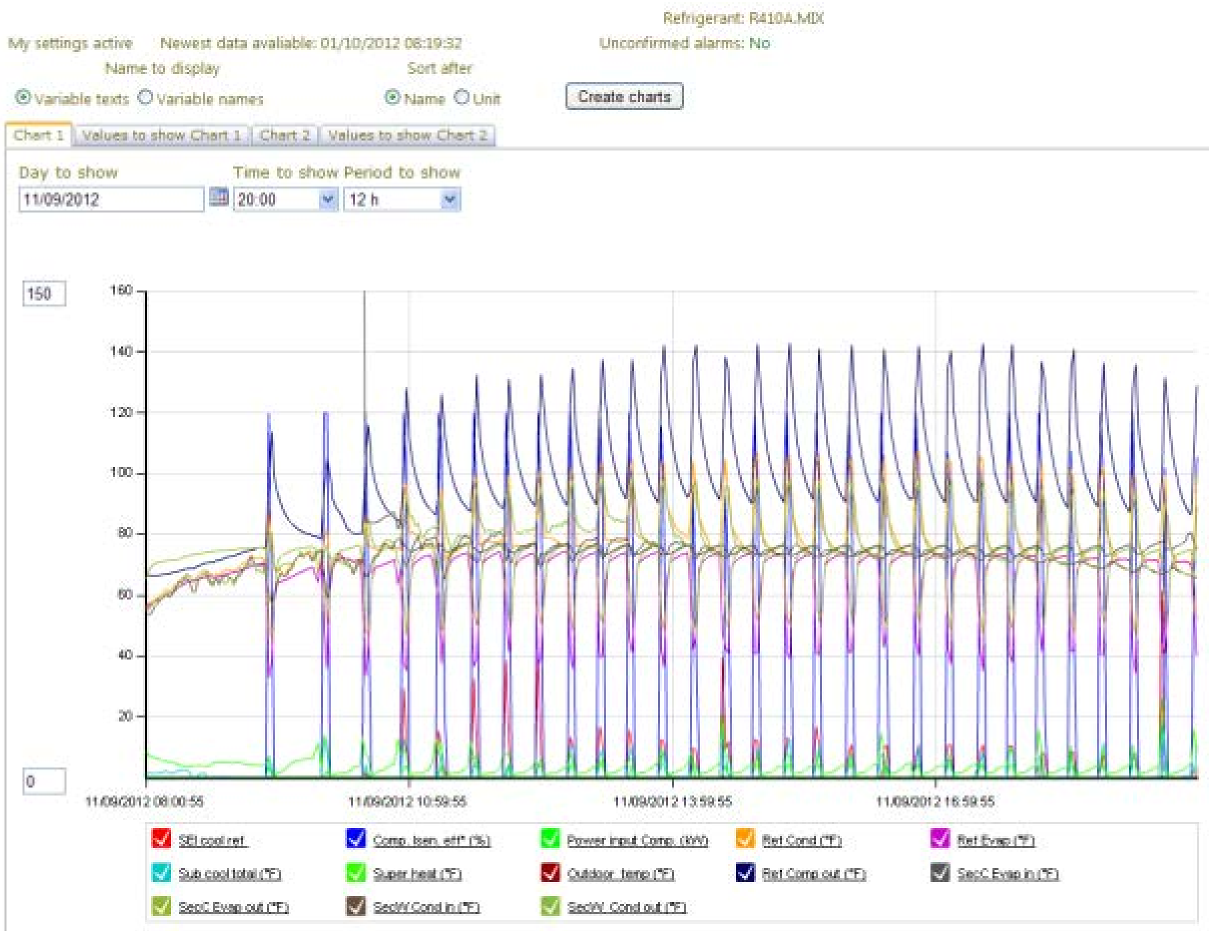


Figure 17, RTU3 on-off operation (°F)

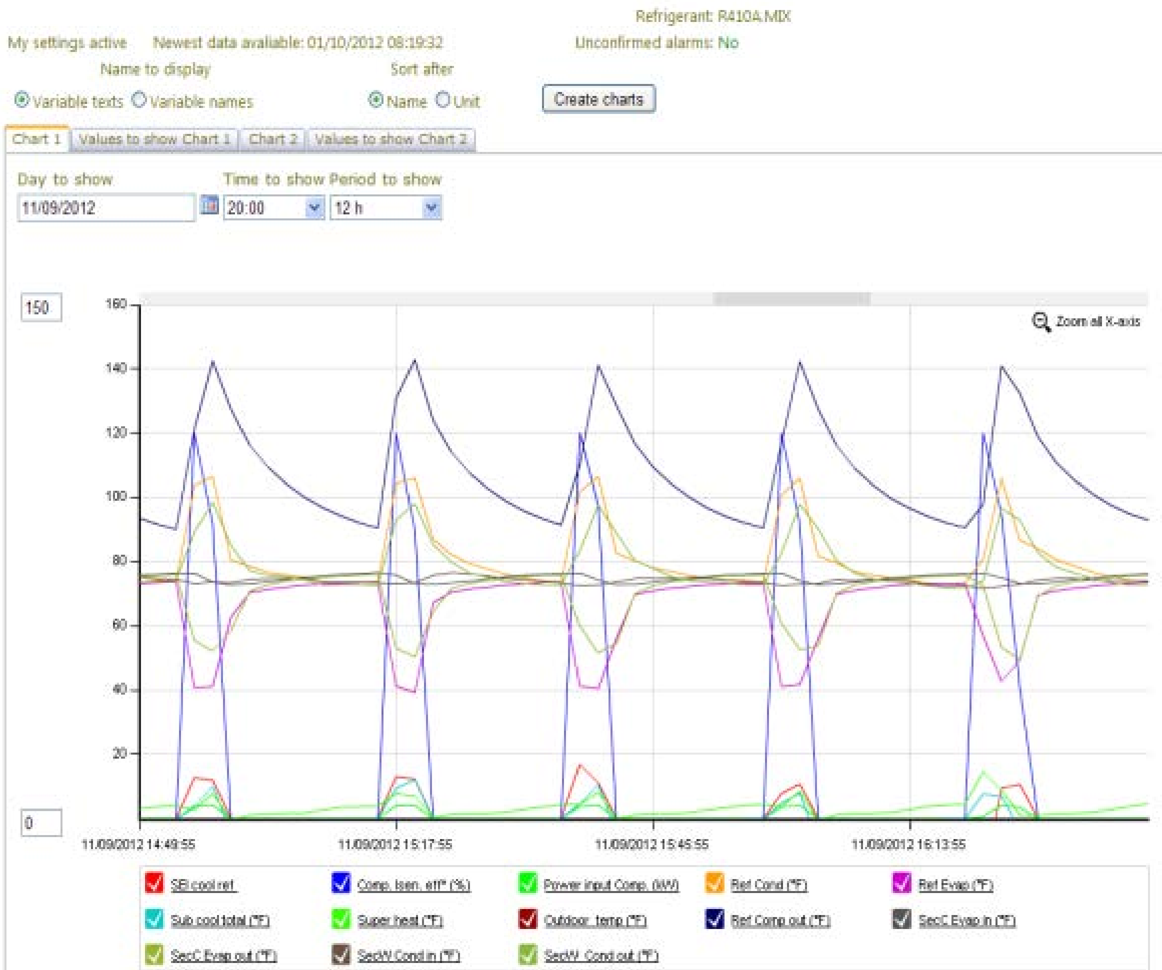


Figure 18, RTU3 sequence of short cycles (°F)

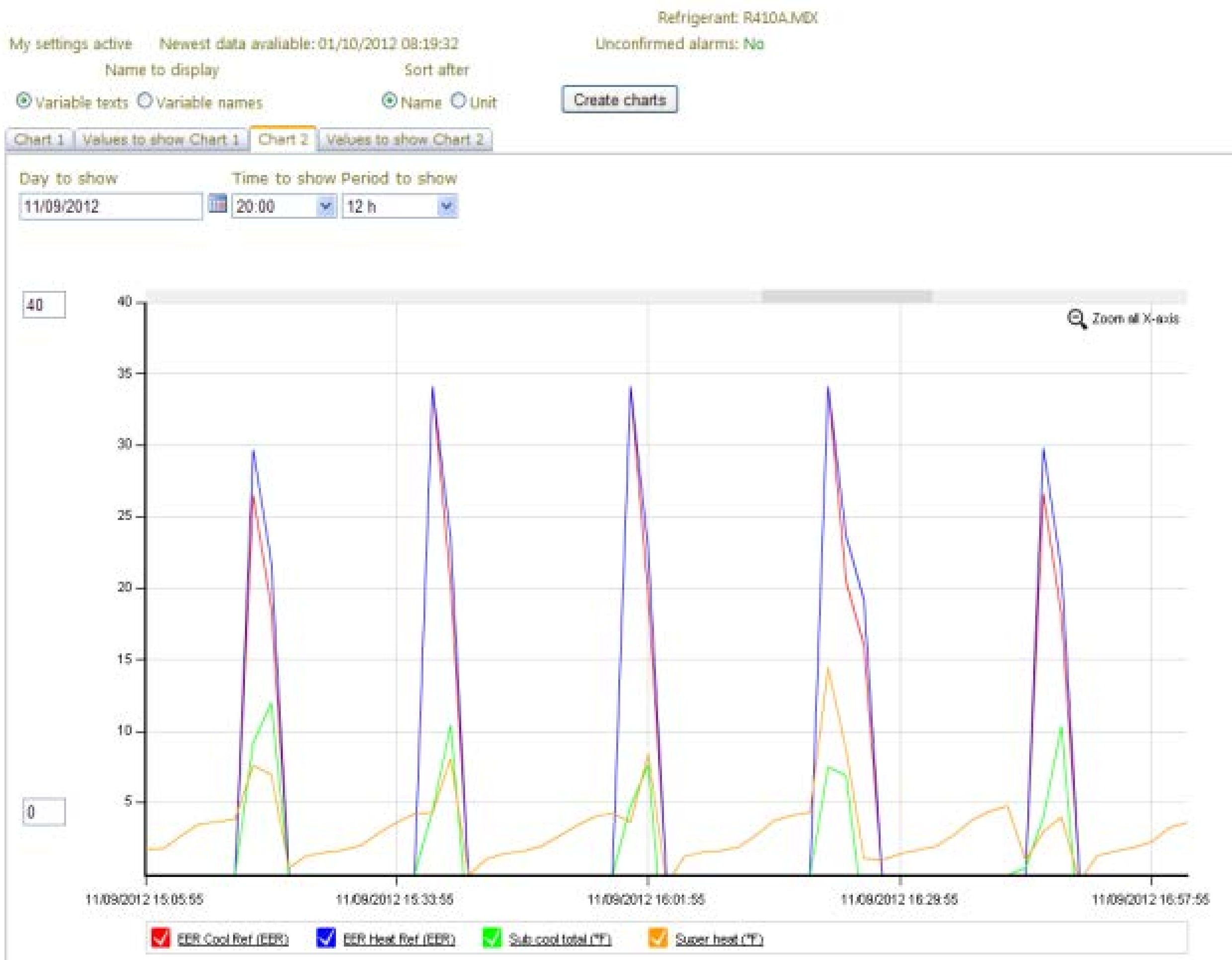


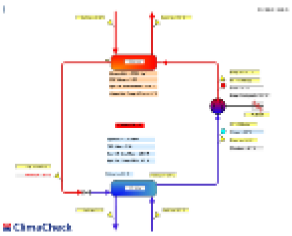
Figure 19, RTU3 Superheat and sub cool (°F)

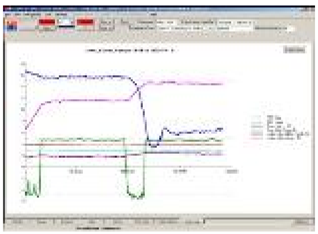
The Internal Method for Performance Analysis, Field Measurement Method for Refrigeration and Air-Conditioning Systems

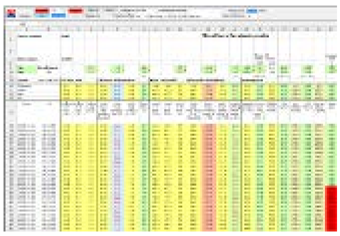
The performance analyser based on the “Internal Method” is an innovative technology that has the potential to revolutionise the industry’s approach to commissioning, trouble-shooting, service and energy optimisation.

Figure 20, ClimaCheck Performance Analyser in portable and fixed versions

It enables engineers in the field to cost-effectively in real time determine the performance of the refrigeration process, its actual COP, capacity, and other vital performance parameters without hours of tedious calculations of a highly skilled engineer. This vital data is presented dynamically in charts and tables, enabling the engineer and/or end user to gain an immediate picture of the actual performance of the system. Suggested optimisation measures can be validated.







The performance is documented in an un-biased way without inputs of manufacturers of system or components. **The method is based purely on fundamental thermodynamic properties and the first law of thermodynamics e.g. energy cannot be destroyed only transformed.**

Accuracy of Results
It accurately determines a working system’s performance:

- Coefficient of Performance (± 5%)
- Cooling and heating capacity (± 7%)
- Power input (± 2 %)
- Compressor isentropic efficiency (± 3%)

The accuracies stated above are based on ClimaCheck PA Pro data acquisition system, standard ClimaCheck sensor accuracy mounted in accordance with

ClimaCheck's manuals and good measuring practice on a standard refrigeration process with a semi-hermetic or hermetic compressor as shown below.

Accuracy of Sensors
Error calculations are based on the below stated accuracy of input sensors.

- Pressure sensor, ± 1% FS
- Temperature sensors PT1000 Class A
- Electrical Energy power meter ClimaCheck EP Pro, Class B
- Current transformers (20-120% of rating), ± 2%

Innovative Approach – How it Works
The system uses ten easy to apply sensors that are attached at strategic points around the system. This is 7 temperatures, 2 pressures and active power as shown.

An engineer can hook up the equipment in 20 minutes. From the information gathered the key operating parameters that pinpoint the system's actual performance can be determined independent of any supplier data. Required measuring points for a standard system as shown, (left).

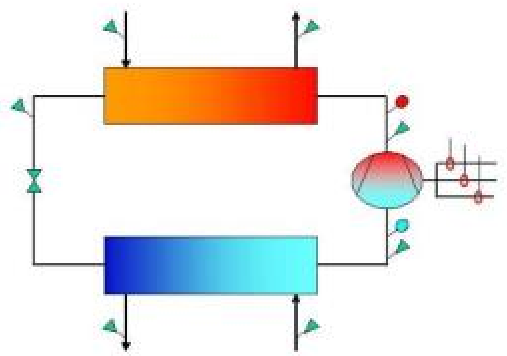


Figure 21, Sensors required and their location to establish performance of a standard refrigeration system

- Temperature and pressure at entrance of compressor.
- Temperature and pressure at compressor exit.
- Liquid refrigerant before expansion device.
- Active electrical power.

For reference of operating condition and heat exchanger evaluation the temperature of air/liquid entering and exiting condenser and heat exchanger are measured. IN total 10 measurements that are easy to apply to almost all systems in the field.

At the heart of the performance analyser is the energy balance over the compressor and a series of algorithms, based on the thermodynamic properties and operating characteristics of the refrigerant in use.

The heat losses are low relative the total input power limiting the impact of variation as documented by (Asercom, 2003) and (Naumburg, 1987). So equation (1) will give a good accuracy of mass flow of refrigerant.

The losses varied in documentation and tests between three and ten percent in hermetic and semi-hermetic compressors without external cooling representing the vast majority of compressors on the market. For open drive and compressors with cooling the same methodology can be used by adding a model of the amount of energy not introduced in the refrigerant flow. When the net energy to the refrigerant flow calculated as the measured electrical power – heat losses are known the mass flow is also known through equation (1).

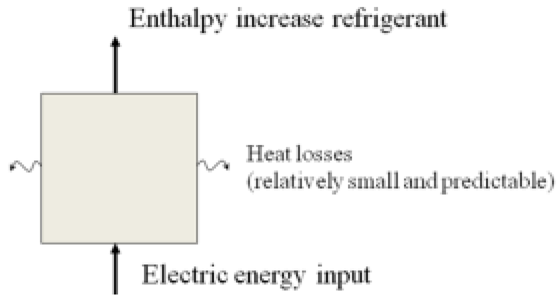


Figure 22, The energy balance with consideration of heat losses over the compressor allows calculation of mass flow

$$\text{Mass flow} = \frac{\text{(electrical input-heat losses)}}{\text{Enthalpy difference}} \quad (1)$$

From the above described energy balance and these enthalpies all data required can be derived including COP, Capacities, and the compressors total isentropic efficiency. Method described in more detail by i.e. (Berglof, Methods and Potential for Performance Validation of Air Conditioning, Refrigeration and Heat Pump Systems, 2004), (Berglof, Methods and Potential for on-site Performance Validation of Air Conditioning, 2005), and (Fahlén, Methods for commisioning and performance checking heat pumps and refrigeration equipment., 2004).

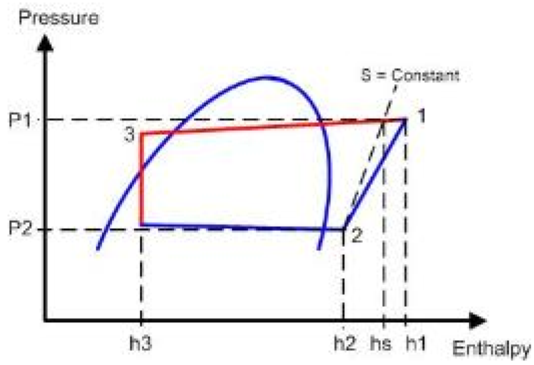


Figure 23, Pressure – enthalpy graph of "standard" refrigeration process

$$\text{Cooling Capacity} = \text{Mass flow} * (h2 - h3) \quad (2)$$

$$\text{Heating Capacity} = \text{Mass flow} * (h1 - h3) \quad (3)$$

$$\text{Isentropic Effic} = \frac{(hs - h2) * (1 - \text{rel. heat loss})}{(h1 - h2)} \quad (4)$$

Well-Proven Method

The method and technology was first developed in Sweden 1986 and validated by SP the national Swedish testing institute (Fahlén, Capacity measurements on heat pumps - A simplified measuring method, 1989). More than 40 manufacturers and 300 contractors in 20 countries have introduced the "Internal Method" as a tool to improve their development, production and aftermarket activities. Examples of world leading companies in the industry that has validated and use the Internal Method to document the performance of their products and optimise the systems are Carrier, Trane, Johnson Control, Copeland, Bitzer, Gea, Danfoss Heat pumps and DuPont.

Practical Benefits

All data required for a full evaluation of the system are available as soon as sensors are connected - most of the time without requirement to stop the system. With the information provided, engineers can identify plant performance problems, including among many others:

- refrigerant shortage or over-charge
- incorrect superheat setting
- compressor damage or wear

- fouling of heat exchangers
- oil logging in the condenser/evaporator
- fan/pumps underperformance, Flow problems on secondary medias (air/water/brine)
- control problems

The system identifies irregularities in compressor, component performance that could result in future impairment of performance – or even plant breakdown, enabling pre-emptive maintenance and energy optimisation.
Armed with this vital information, engineers can address the issues identified, optimising system performance. The result is huge potential savings in power consumption and carbon emissions over a plant's lifetime.

Without an effective method and an efficient tool, these problems normally go unrealized, with the plant continuing to perform inefficiently – or eventually breaking down with potentially catastrophic consequences for refrigerant loss and stock damage.

Whenever required a modem can be connected to the data collection unit and information in real time transferred to an Internet server where calculations are done and made available to any expert in the world who is given access through user name and password for validation and advice on best actions to take.

Appendix C – Definitions of Key Terms

Long Name	Abbreviation	Description
Outdoor Air Temperature	OAT	Dry bulb temperature measured with a rooftop sensor located adjacent to the RTU in a radiation shielded enclosure.
Supply Air Temperature	SAT	Dry bulb temperature measured in the supply air duct near diffuser.
Mixed Air Temperature	MAT	The average dry-bulb temperature before the cooling coil. An average of four dry-bulb temperature sensors are suspended from the air filter.
Return Air Temperature	RAT	Dry bulb temperature measured with a sensor located in the return air duct.
Rooftop Unit	RTU	Also known as a unitary, or packaged HVAC system. This is a single enclosure that contains compressor, condenser and supply fan designed to provide cooling and heating to a building, or to a portion of a building.
Coefficient of Performance	COP	The ratio of the rate of heat transfer out of the supply air and the power input to the compressor in BTU/BTU.
Energy Efficiency Rating	EER	The ratio of the rate of heat transfer out of the supply air and the power input to the compressor in BTU/W.
Seasonal Energy Efficiency Ratio	SEER	A lab-tested value in which the projected EER at several conditions is combined.
Integrated Energy Efficiency Ratio	IEER	A lab-tested value in which the EER is tested at several partial loading conditions and an integrated value is found.
Cooling Season Energy Use	-	Estimated energy use of supply fan, compressor and condenser fan under typical meteorological year conditions from May 1 to October 31.
Dry bulb temperature	T_{db}	Dry bulb temperature
Dew point temperature	T_{dp}	Dew point temperature
Relative humidity	RH	Relative humidity
Cubic feet per minute of air	CFM	Actual cubic feet per minute of air

Appendix D – Flow Calibration Curves

Figure 41. Calibration Curve for the NBIL Flow Measurement

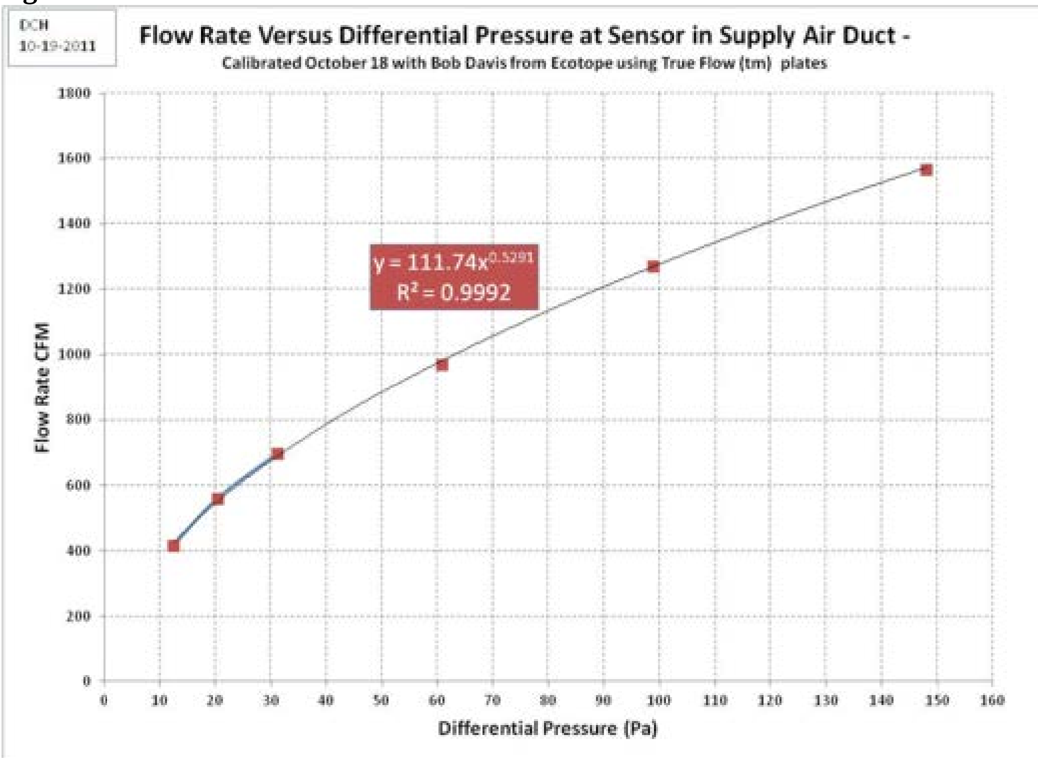


Figure 42. Calibration Curve for the Daikin Field Flow Measurement