Emerging Products

PACKAGED AIR CONDITIONING UNIT ENHANCED CONTROLLER

ET13SCE1030 Report

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EXECUTIVE SUMMARY

This Air Conditioning (AC) Controller field assessment measures the electrical energy and demand savings of an enhanced controller for a packaged AC unit retrofit. This controller included variable supply fan speed control, Demand Control Ventilation (DCV), and Fault Detection and Diagnostics (FDD).

Approximately 65% of California's commercial floor area is conditioned by packaged AC units[i]. The majority of existing units are constant volume systems, delivering a fixed amount of conditioned supply air to the building space. This supply air quantity is based upon the space's maximum cooling demand. Because this demand occurs for only a few hours per year, the AC units operate at part load for the majority of the year. However, the AC unit supply fan consumes the same amount of energy regardless of the space's cooling or heating demand.

Several manufacturers have developed systems that convert constant volume AC units to variable air volume operation. These systems reduce the supply air flow when there is no call for cooling or heating from the served space. The systems can be retrofitted into existing AC units. They consist of a variable frequency drive (VFD), sensors, and controls.

In this study, a retrofit controls system was added to four rooftop packaged heat pumps serving the common atrium area of an enclosed shopping mall in the metropolitan area of Loa Angeles County, California. This system reduced the AC units' supply fan flows when there was no call for cooling or heating in the conditioned spaces. The supply fan and cooling system energy consumption was measured and recorded, and compared to baseline operation (constant volume fan).

Fan energy savings were measured at each AC unit when outside air temperature was between 50 - 95°F. As outside air temperature rose, the demand for cooling increased, and fan energy savings decreased. Table 1 summarizes the supply fan energy savings for each of the tested AC units, for 5°F outside air temperature bins. These savings are compared to the same supply fans operating continuously at full speed during the AC units' hours of operation.

Table 2 estimates the annual supply fan energy savings given the site's operating hours and ASHRAE weather information.

TABLE 1. SUPPLY FAN ENERGY SAVINGS – ONE AND TWO-STAGE COOLING

	OUTSIDE AIR TEMPERATURE BIN (F)							
UNIT	55 - 60	60 - 65	65 - 70	70 - 75	75 - 80	80 - 85	85 - 90	90 - 95
AC-18 (One-Stages Cooling)	89%	89%	81%	64%	38%	27%	25%	25%
AC-19 (Two-Stages Cooling)	90%	90%	82%	74%	51%	38%	30%	25%
AC-20 (Two-Stages Cooling)	91%	91%	89%	81%	65%	52%	40%	34%

TABLE 2 ESTIMATED ANNUAL SUPPLY FAN ELECTRICAL SAVINGS

	Annual Supply Fan Energy (kWh/yr)		
UNIT	Baseline	Savings	% Savings
AC-18 (One-Stages Cooling)	7145	4316	60%
AC-19 (Two-Stages Cooling)	15468	10076	65%
AC-20 (Two-Stages Cooling)	11952	8508	71%

Based upon the results of this field assessment, retrofit control systems that adjust the AC unit's supply fan speed in response to cooling and heating demand is recommended for the Energy Efficiency Program. In addition to their fan control, these control system's FDD technology can identify and notify the building owner of such faults as economizer failure, problems with the cooling system, and fan belt slippage. The systems' DCV features can provide energy savings for spaces that have high occupant densities.

Because this assessment was applied to only one building type in one climate zone, energy simulations need to be performed to calculate energy savings for several market segments, in each climate zone. A work paper was submitted to the California Public Utilities Commission's (CPUC) Energy Division (ED) by Portland Energy Conservation Incorporated (PECI) in June 2011, on SCE's behalf. This work paper presented the results of an eQuest computer simulation of stepped fan control for RTUs. The paper's results covered all of the climate zones in SCE territory. Assembly, Education, Restaurant, Retail, and Office applications were modeled in that analysis. The energy model included DCV for market sectors where California Title 24 allows this control strategy. Once the review of that work paper is complete, its underlying eQuest model can be used to estimate the energy savings of stepped fan control and DCV, for the commercial market sectors that use packaged AC units. The model can also be altered to remove DCV from the calculations for energy savings.

Retrofit control systems have some potential for demand response, as they can temporarily lock out cooling and reduce fan power upon receiving an external signal. However, this controls retrofit is primarily an energy reduction strategy for periods of lower cooling loads.

ABBREVIATIONS

AC	Air Conditioning
AHU	Air Handling Unit
ALC	Automated Logic Corporation
CFM	Cubic Feet per Minute
CO_2	Carbon Dioxide
DCV	Demand Control Ventilation
FDD	Fault Detection and Diagnostics
HTSDA	HVAC Technologies and Systems Diagnostics Advocacy program
HVAC	Heating, Ventilation, and Air Conditioning
LEED	Leadership in Energy and Environmental Design
OSA	Outside Air
ppm	Parts per million
RA	Return Air
RTU	Roof-Top Unit
SA	Supply Air
SCE	Southern California Edison
sf	Square feet
Title 24	California Energy Code, California Code of Regulations Title 24, Part 6
VFD	Variable Frequency Drive

CONTENTS

FIGURES

TABLES

EQUATIONS

INTRODUCTION

Approximately 65% of California's commercial floor area is conditioned by packaged AC units[i]. These units consist of supply fans, Direct eXpansion (DX) cooling systems, heating, and air filters. The majority of these units are located on building roofs, referred to as Roof-Top Units (RTUs).

The majority of existing RTUs are constant volume systems, delivering a fixed amount of conditioned supply air to the building space. This supply air quantity is based upon the space's maximum cooling demand. Because this demand occurs for only a few hours per year, the RTU operates at part load for the rest of the year.

Larger air handling systems vary the amount of supply air delivered to conditioned spaces in response to changing cooling demands. These systems, when designed and operated properly, provide code-compliant outside air (OSA) quantities to the indoor environment. They also consume less fan energy for most hours during the year. The constant volume RTU, on the other hand, consumes the same amount of fan energy regardless of the space's cooling or heating demand.

Several manufacturers have developed systems that convert constant volume RTUs to variable volume or stepped volume operation. These systems reduce the supply air flow when there is no call for cooling or heating from the served space. The systems can be retrofitted into existing RTUs. They consist of a variable frequency drive (VFD), sensors, and controls.

These same retrofit systems can adjust the amount of OSA delivered to the conditioned space, in response to occupancy. The carbon dioxide (CO_2) level in the air is an indirect indicator of the number of occupants, and this level can be measured and used to adjust OSA flow. Reducing the OSA flow reduces the amount of cooling required to bring this air from its hot outside temperature to a comfortable inside temperature.

Some retrofit control systems have the capability to provide several added points of monitoring to the RTU, which provides greater capacity for Fault Detection and Diagnostics (FDD). For example, supply fan amperage readings give evidence of slipping or broken fan belts. FDD facilitates a switch from reactive maintenance to cost-effective, planned maintenance and AC system optimization. Added monitoring points include compressor amperage, economizer damper position, mixed air and supply air temperatures.

BACKGROUND

CONSTANT VOLUME VERSUS VARIABLE VOLUME FANS

Air Handling Systems (AHUs) with Variable Air Volume (VAV) serve large commercial facilities. These AHUs control room temperatures by varying the amount of cool air supplied to these spaces. As the room's thermostat senses an increase in temperature, it responds by increasing the flow of conditioned air. The supply fans within these AHUs speed and slow in response to varying cooling demand in the spaces. Because AHU cooling demand is less than 100% much of the time, fan energy is reduced for most operating hours of the year.

Most commercial air conditioning applications, however, are constant air volume packaged AC units. These units deliver a constant flow of conditioned supply air to the spaces, regardless of cooling or heating demand. These systems control room temperature by turning on stages of cooling and heating.

In the past, packaged AC units have not had variable speed supply fans for the following reasons:

- The cost of Variable Speed Drives (VFDs) was not economical for smaller motors.
- AC units employ DX refrigerant cooling coils. Reducing the air flow across this coil caused operational problems, such as coil freeze-up. Moisture in the air stream would freeze on the face of the coil, blocking the air path. The AC unit's control system did not adjust the capacity of the cooling system in response to reduced air flow. By contrast, AHU chilled water coils constantly adjusted capacity to meet demand, and chilled water temperatures was not low enough to cause coil freezing.

Newly packaged AC units on the market have variable speed supply fans and the necessary controls to allow the cooling system to operate properly at part loads. Effective January 1, 2012, the 2010 California Energy Code, California Code of Regulations Title 24, Part 6 (Title 24) requires that single-zone packaged AC units with mechanical cooling capacity greater than 110,000 British thermal unit per hour (Btu/hr) have supply fans controlled by VFDs or by two-speed motors. The proposed 2013 version of this code will require this level of fan control for RTUs with cooling capacities over 65,000 BTU/hr by January 1, 2016. However, there is a large inventory of existing, older AC units that have many years of useful economic life. While the published life of these units is 15 years, they are normally maintained in operation for 25 years or more. The 2013 Title 24 will require FDD for RTUs with a cooling capacity over 54,000 BTU/hr. This FDD shall, at a minimum, annunciate faulty economizer operation, excess OSA flow, and air temperature sensor faults. This code is scheduled to go into effect in July 2014.

PACKAGED AIR CONDITIONING UNIT CONTROL RETROFIT

Several manufacturers have introduced control systems targeting existing AC units. These systems do the following:

- Change the supply fan speed in response to space cooling and heating demand
- Control the OSA and return air (RA) dampers to provide demand control ventilation
- Provide Fault Detection and Diagnostics (FDD)

The design of each manufacturer's system varies, but most include a supply fan VFD, room CO_2 sensor, OSA CO_2 sensor, damper controls, and a master controller.

VARIABLE VOLUME CONTROL

The retrofit controls included a VFD that adjusts the supply fan speed. For this field test, when there was no call for cooling or heating from the room thermostat, the fan's speed was reduced to 40% of full speed. On a call for cooling, when OSA temperature was low enough to provide partial or full cooling, supply fan speed was increased to provide this cooling. If OSA was too hot or could not satisfy the full cooling load, stages of DX cooling were energized. Table 3 summarizes the supply fan speeds used in the field tests. These fan speeds were provided by the control system manufacturer.

TABLE 3 SUPPLY FAN SPEED SETTING

COOLING STAGE	OSA TEMP < 58°F	58°F < OSA TEMP <70°F	OSA > 70°F
Off	40%, Minimum OSA	40%, Minimum OSA	40%, Minimum OSA
1st	Economizer, 75%	Economizer, 90%	1st Stage Mechanical Cooling, 75%
2nd	Economizer, 90%	1st Stage Mechanical Cooling, 90%	2nd Stage Mechanical Cooling, 90%

The air-side economizer employs OSA to cool the conditioned spaces during periods of cooler outdoor temperatures. When OSA temperature is below 58°F, OSA and return air dampers are modulated to deliver 58°F mixed air to the AC system. During temperate outdoor conditions between 58°F and 70°F, the OSA damper is opened to provide 100% OSA. If additional cooling is required, the DX system is then energized. When the OSA exceeds 70°F, the control system allows 100% OSA air as long as the return air temperature is higher than the OSA. All of the above temperatures are adjustable.

This technology presents items that require additional attention during a retrofit:

1. When a packaged AC unit is first installed, its OSA damper is set at a minimum position by an air balancing contractor. This position allows a specified OSA flow, based upon code requirements or other design criteria, into the system. If the supply fan speed is reduced, the OSA flow is reduced proportionally. Insufficient OSA creates health concerns, code violations, and possibly pressurization problems within the space.

   Retrofit control systems correct this problem by simultaneously opening the OSA damper, to maintain a steady flow of OSA even when the fan speed is reduced. This can be done by pre-setting OSA and return air damper positions for each proposed fan speed, by an air balancing contractor.

2. Variable frequency drives cause high voltage spikes in the motor they serve. These spikes can ionize the air between the motor windings, causing localized arcing and damage to the insulation. Inverter duty motors are designed to withstand this damage, but existing AC unit fan motors are general-purpose type. Motor failures have not been reported when retrofitting a VFD to a general-purpose motor, but this remains a concern.

3. Harmonics: Adding VFDs to a building's electrical distribution system has the potential of introducing line-side harmonics to that system. Computers, communications, and diagnostic equipment have a low tolerance to harmonics. Care must be taken to ensure that installed VFDs contain components to minimize line-side harmonics.

ASSESSMENT OBJECTIVE

The field assessment measures the electrical energy and demand savings of supply fan speed control for packaged AC unit retrofits. The study's goal is to determine the reduction of compressor and supply fan energy usage during periods of moderate demand. This assessment also measures electrical power factor changes, if any, caused by the introduction of supply fan speed control to an RTU.

TECHNOLOGY/PRODUCT EVALUATION

In this study, a retrofit controls system was added to four rooftop packaged heat pumps serving the common atrium area of an enclosed shopping mall in metropolitan Los Angeles County, California. These AC units were selected because they serve an area with a high design occupancy, which allowed for testing of the demand controlled ventilation features of the controls.

The control system is designed to be installed in the control wiring between the RTU and the space thermostat controlling this RTU. A VFD is installed between the supply fan motor and its circuit breaker. The control system receives signals for fan operation, cooling, and heating, and in turn it controls fan, economizer damper, and compressor operation of the RTU. Figure 1 shows a simplified diagram of the controller's installation.

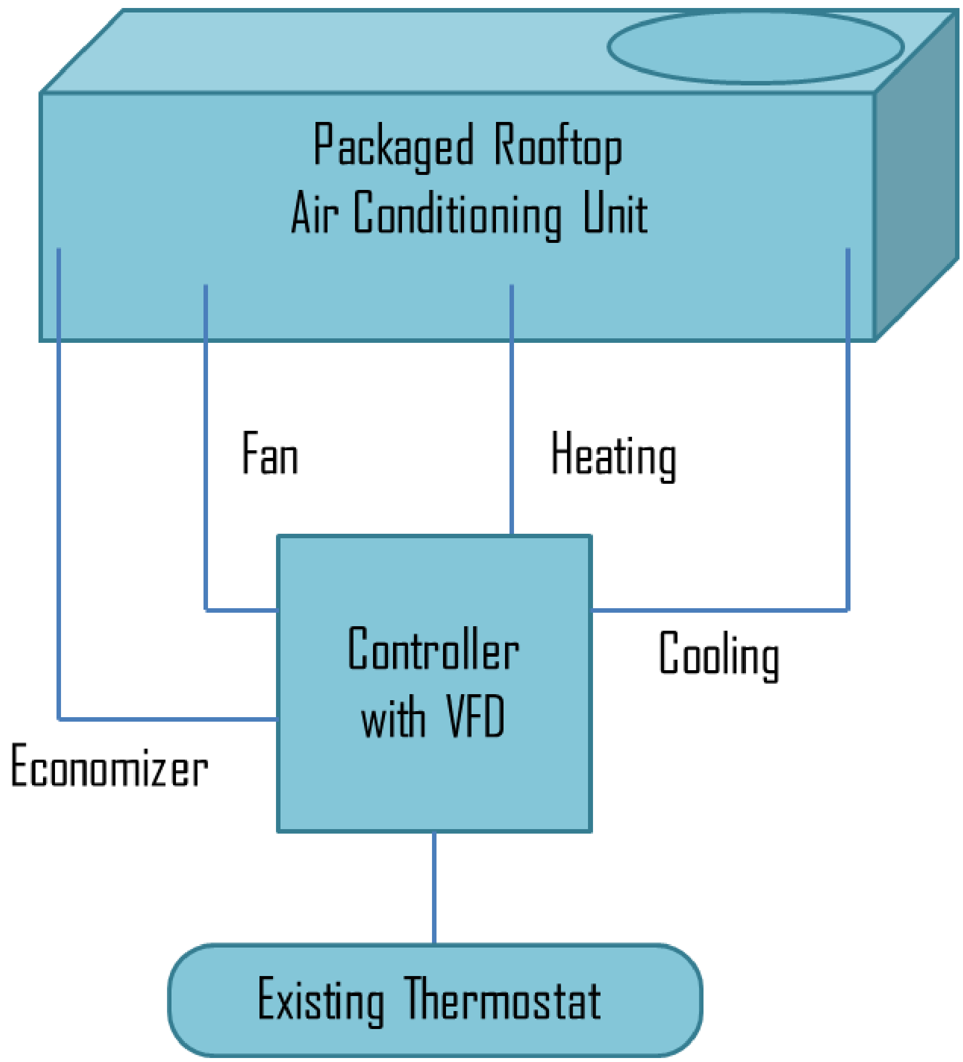

FIGURE 1 – AC UNIT RETROFIT CONTROLLER DIAGRAM

TECHNICAL APPROACH/TEST METHODOLOGY

FIELD TESTING OF TECHNOLOGY

The center arcade of the Mall is served by eight packaged rooftop air-to-air heat pumps. Each unit consists of a filter section, supply fan, DX cooling/heating coil, refrigeration system, economizer dampers, and controls. Conditioned air from each unit is delivered to the space below through separate ducted air distribution systems. Space thermostats, located within the conditioned space, control the cooling and heating stages of the heat pumps. The units operate from 10:00 AM to 6:30 PM, seven days/week.

Three of these eight AC units, AC-18, AC-19, and AC-20, were used for the field test. The following are the nameplate data of these units:

AC-18: Carrier Model 50TJ-016AC-620QA

Serial No. 2298F46656

Vintage: 1998

Nominal Capacity – 15 tons

Supply fan – 3.7 hp

EER – 8.9

AC-19: Carrier Model 50TJ-024AC-670QA

Serial No. 3899F51801

Vintage: 1999

Nominal Capacity – 20 tons

Supply fan – 5 hp

EER – 9.2

AC-20: Carrier Model 50TM-016-611QA
Serial No. 4405007580
Vintage: 2005
Nominal Capacity – 15 tons
Supply fan – 7-1/2 hp
EER – 9.7

Figure 2 shows the locations of the AC units used in the field evaluation.

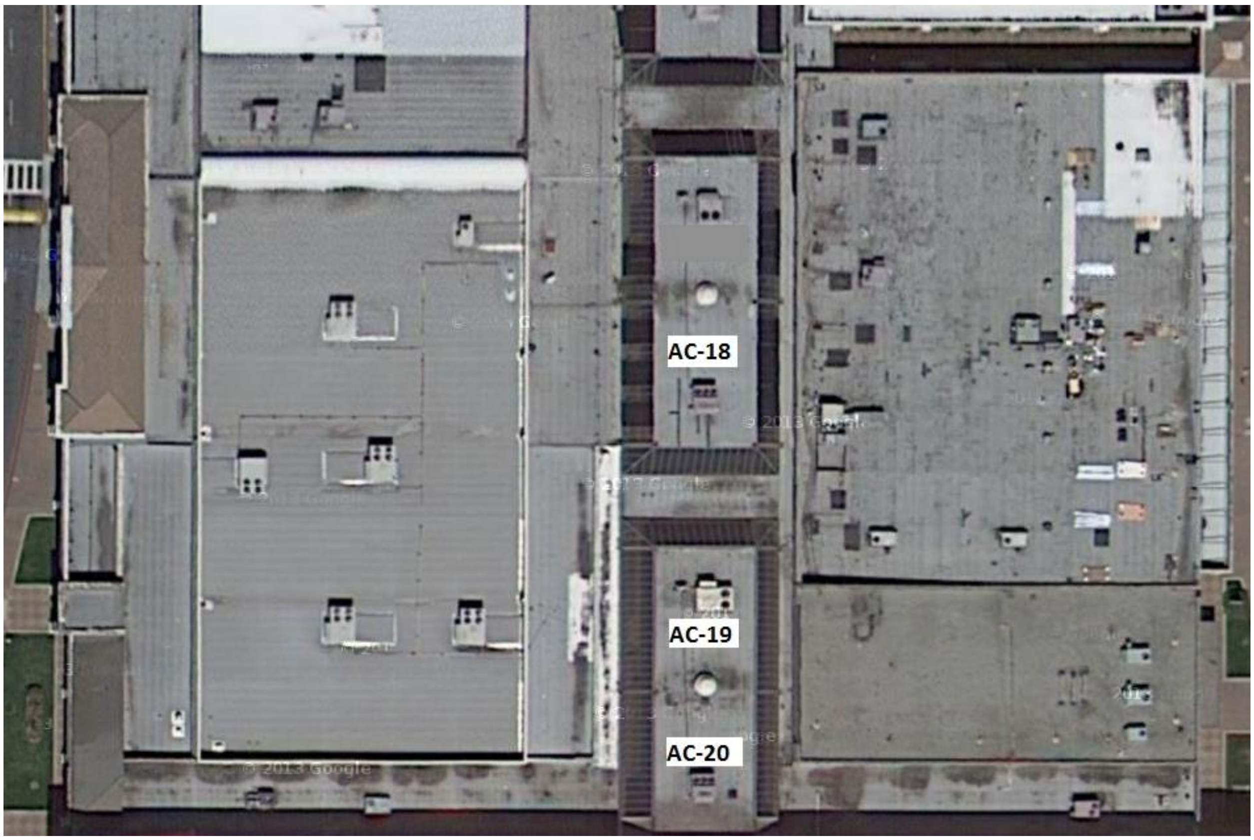

FIGURE 2 – AIR CONDITIONING UNIT LOCATIONS

The test site was selected because the cooling load of the conditioned spaces is highly weather-dependent. The internal cooling loads (lights, equipment) are small in proportion to the external (skylights, OSA) loads. Wide cooling load swings would provide the opportunity to observe the performance of the baseline and measure at part load conditions.

Because the test site was the common area of a shopping mall, it was a good candidate for DCV testing. Its design occupancy is 40 persons/1000 square feet (sf), but its actual occupant density is much less. The 2010 Title 24 requires DCV control in new systems serving spaces with densities greater than 25 people per 1,000 sf.

The baseline AC operation is as follows:

- A facility time-clock energizes the AC unit during normal operating hours
- The supply fans run continuously, at 100% speed, during operating hours
- Under normal conditions, the OSA dampers are at a minimum position.
- When the AC unit's space thermostat senses room temperature above set point:
    - If the OSA temperature is below set point, the economizer dampers open, to provide 100% OSA. If additional cooling is required, the DX system is energized.
    - If the OSA temperature is above set point, the DX system is energized. The economizer dampers close to minimum OSA position.
- When the AC unit's space thermostat senses room temperature below set point, the DX system is energized and the reversing valve puts the AC system in heating mode.

The controls system was installed in each of the tested AC units. This control system has a mode that allows existing baseline operation to continue even if the control system is in place. Control signals from the room thermostat pass through the controller unchanged.

The performance of the AC systems was recorded under the following conditions:

- Baseline operation
- Supply fan speed control

For each test, the following measurements were recorded:

- OSA temperature, dry bulb and wet bulb
- System voltage
- Supply fan amps
- Compressor amps
- Return air temperature
- Mixed air temperature
- Supply air temperature
- Return air CO_2 level
- OSA CO_2 level

Figure 3 indicates the general location of points monitored for each AC unit. OSA temperatures and CO_2 levels were measured at one point, for all air conditioning units.

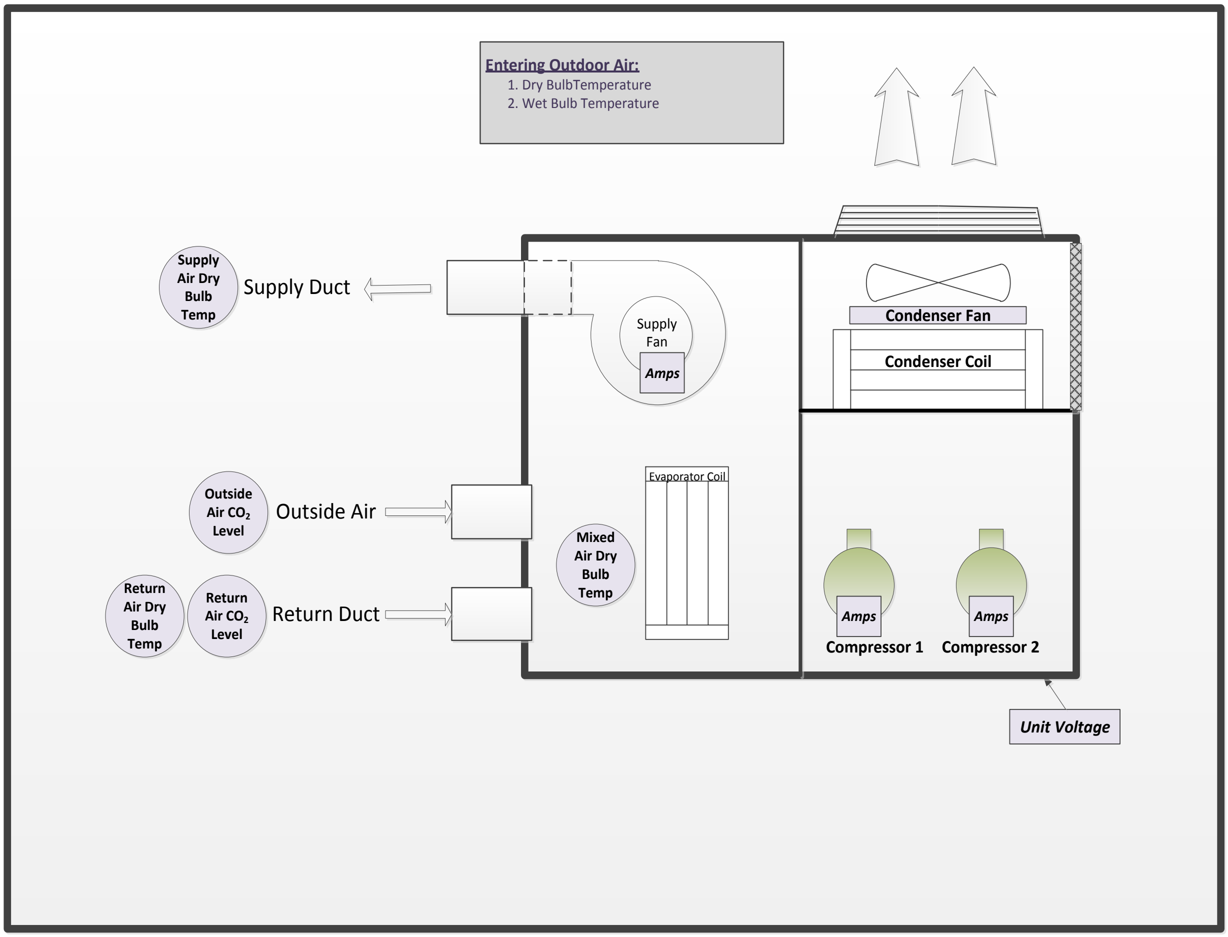

FIGURE 3. AIR CONDITIONING UNIT MONITORED POINTS

The above measurements were used to calculate fan and cooling system energy usage for each test mode.

A separate measurement of AC-18 power factor at each baseline and test condition was taken.

TEST PLAN

Before conducting any tests, the existing AC units were serviced to bring them to proper operating conditions. This service included:

- Air filter replacement
- Proper refrigerant charge
- Supply fan belt adjustment
- Economizer damper service

At this time, it was determined that the damper actuators for three of the four test units were inoperative. These damper motors were replaced. The evaporator coils were observed to be clean. The condenser coils were observed to be dirty but relatively unclogged.

Once the AC units were returned to proper operating conditions, a monitoring system was installed on-site. This system collected and stored measured data in an on-site computer. Through a wireless card, the monitoring computer was accessed off-site and collected data was downloaded.

Two test conditions were monitored and compared:

Test 1. Baseline Operation

- Supply fan operating at 100% speed, during occupied hours
- OSA damper at minimum position, unless economizer operation is called for
- Economizer operation, on a thermostat call for cooling and OSA temperature below set point
- DX cooling and heating operational

Test 2. Supply Fan Speed Control

- During occupied hours, fan operates at 40% speed on no thermostat call for cooling or heating. The OSA damper opens to a position that maintains baseline OSA flow.
- During occupied hours, the fan operates at:
  - Ninety percent speed on thermostat call for cooling or heating, for AC-18 (one stage cooling/heating)
  - Seventy-five percent speed on thermostat call for first stage cooling or heating. Ninety percent speed on thermostat call for second stage cooling/heating. This applies to AC-19 and AC-20, which each have two stages of cooling and heating.
- DX cooling and heating and economizer operational

The AC unit system energy usage for each test was compared for the following OSA dry bulb temperature bins:

- 55º - 60ºF
- 60º – 65ºF
- 65º – 70ºF
- 75º – 80ºF
- 80º – 85ºF
- 85º – 90ºF
- 90º – 95ºF

Data were recorded every 10 minutes. This time interval allowed determination of the AC units' run condition (cooling, heating, fan only). Once energized, a compressor will operate for a minimum of 3 minutes. Once de-energized, a compressor has a start time delay of a minimum of 3 minutes.

Data was downloaded remotely to Excel worksheets every week. At times, communications with the monitoring system was lost, but data was nonetheless being collected by the on-site computer. In these cases, data was collected after communications were re-established.

One time measurements were taken of power factor for unit AC-18. These measurements were taken under the following operating conditions:

<u>Baseline</u>

- 100% fan speed, no cooling
- 100% fan speed, cooling energized

<u>Measure</u>

- 40% fan speed, no cooling
- 75% fan speed, no cooling
- 90% fan speed, cooling energized

INSTRUMENTATION PLAN

Appendix A provides the list of monitored points, sensors used, and sensor accuracy.

An Automated Logic Corporation (ALC) model SE6166 control module was installed in each RTU to read the monitored points for that unit. These modules communicated with an ALC model #LGR-25 wireless router, which stored collected data. These data were uploaded via Internet to SCE's project engineer.

A Fluke® model #435 portable Power Quality Analyzer was used to record power quality of AC-18.

RESULTS

For each of the test modes, data was collected from the AC units for a period of two – three weeks. The time period for each test depended upon the range of OSA temperatures experienced during that period. If a wide range of OSA temperatures was measured, the monitored period was shortened.

Baseline data from March 1 -24, 2012 were analyzed, and the following observations were made:

- The measured OSA temperature rose above expected range in the mid-afternoon.
- The AC units didn't call for cooling, even when the return air temperatures rose several degrees above set point.

The following conditions were found and corrected:

- The OSA sensor had been mounted on the west side of an AC unit. In the early afternoon, this sensor was exposed to direct sunlight, causing higher OSA temperature readings to spike. This sensor was subsequently relocated to the shade on the north side of the AC unit.
- The controller was incorrectly set, causing it to block calls for cooling from the space thermostats. These settings were corrected, and proper cooling operation was observed thereafter.

After the above corrections were made, Test 2, Supply Fan Speed Control, was conducted on AC-18 thru 20. This test ran from March 28 to April 13, 2012.

Test 1, Baseline, was conducted from April 14 to April 30, 2012.

After analysis of the test data, it was determined that the current transducers had been installed in the incorrect location. They had been installed downstream of the VFDs rather than upstream. Analysis of baseline and measure fan amperage showed that these measured values remained stable for each operating condition. As a result, current transducers were relocated upstream of the VFDs, and amperage measurements were recorded between January 15 – 19, 2013. These measured values were used for analysis.

DATA ANALYSIS

Table 4 summarizes the measured supply fan amps for AC-18 thru AC-20, during Baseline Test 1 and Supply Fan Speed Control Test 2.

TABLE 4. FAN AMPERAGE, BASELINE AND FAN SPEED CONTROL

UNIT #	BASELINE CURRENT (A), TEST 1	CURRENT (A), TEST 2, NO COOLING	CURRENT (A), TEST 2, STAGE 1 COOLING	CURRENT (A), TEST 2, STAGE 2 COOLING
AC-18	1.9	.2	1.4	NA
AC-19	4.2	.4	1.9	3.1
AC-20	3.2	.3	1.4	2.4

Equation 1 provides the baseline fan energy for single stage cooling unit AC-18.

EQUATION 1. FAN ENERGY CONSUMPTION, BASELINE AC-18

$$FanEnergy_{Baseline} = Volts * Amps_{Baseline} * 3 / PF$$
$$= 275V * 1.9A * 3 / 0.98$$
$$= 930W$$

Where PF is the unit power factor. Refer to Table 8 for power factors measured for baseline and measure operation.

Equation 2 calculates the fan power savings, in watts, from the measured currents and voltages. The calculation of power savings for single-stage cooling unit AC-18 is shown, as an example. Equation 3 calculates the fan power savings, compared to baseline. Again, AC-18 is used as an example calculation.

EQUATION 2. SUPPLY FAN POWER SAVINGS, AC-18 (SINGLE STAGE COOLING)

$$FanEnergySavings = Volts * \left[\left(Amps_{Baseline} / PF_{Baseline}\right) - \left(Amps_{Measure} / PF_{Measure}\right)\right] * 3$$
$$= 275V * \left[(1.9 / 0.69) - (1.4 / .67)\right] * 3$$
$$= 563W$$

EQUATION 3. FAN ENERGY SAVINGS, AC-18, COOLING STAGE

$$FanSavings = \frac{FanEnergySavings}{FanEnergy_{Baseline}} * 100$$
$$\frac{563W}{2287W} * 100$$
$$= 25\%$$

Table 5 shows the calculated supply fan power savings for each AC unit, at each cooling stage, compared to baseline power.

TABLE 5. SUPPLY FAN POWER SAVINGS

UNIT		Cooling Stage	Volts, phase-phase	Fan Amps	Fan Watts	Savings, watts	Savings, %
AC-18	Baseline	Off	275	1.9	893	-	-
		Stage 1	275	1.9	629	-	-
	Measure	Off	275	0.2	92	801	89
		Stage 1	275	1.4	447	182	25
AC-19	Baseline	Off	275	4.1	3486	-	-
		Stage 1	275	4.1	4952	-	-
		Stage 2	275	4.1	4952	-	-
	Measure	Off	275	0.4	340	3146	90
		Stage 1	275	1.9	2396	2556	52
		Stage 2	275	3.1	3837	1114	23
AC-20	Baseline	Off	275	3.2	2694	-	-
		Stage 1	275	3.2	3826	-	-
		Stage 2	275	3.2	3826	-	-
	Measure	Off	275	0.3	255	2439	91
		Stage 1	275	1.4	1724	2102	55
		Stage 2	275	2.4	2948	878	23

Table 6 summarizes the percentage of time each stage of mechanical cooling was energized for each AC unit. Results are presented in 5°F air temperature bins, from 55 to 95 degrees F. These values, combined with the measurements in Table 4, provided a calculated energy savings provided by varying supply fan speeds.

TABLE 6. AIR CONDITIONING UNIT COOLING OPERATION

		OUTSIDE AIR TEMPERATURE BIN (°F)							
UNIT	Cooling Stage	55 - 60	60 - 65	65 - 70	70 - 75	75 - 80	80 - 85	85 - 90	90 - 95
AC-18	1	0%	1%	14%	39%	80%	96%	99%	100%
AC-19	1	0%	0%	15%	31%	67%	85%	95%	99%
	2	0%	0%	9%	13%	46%	65%	82%	93%
AC-20	1	0%	0%	4%	22%	52%	70%	85%	92%
	2	0%	0%	0%	5%	23%	42%	63%	75%

Energy savings for one-stage cooling units, like AC-18, was calculated for each temperature bin by multiplying the fan energy difference between the measure and the baseline by the percentage of time cooling was not energized. For example, for temperature bin 65 – 70°F, cooling was energized only 14% of the time. Equation 2 shows the electrical energy savings calculation. Equation 5 uses this equation for AC-18 at outside air bin 65 - 70°F.

EQUATION 4. FAN ENERGY SAVINGS, ONE-STAGE COOLING AC UNIT

$$\%FanSavings = 100 * \left\{ \begin{array}{l} \left[FanEnergySavings_{1STAGE} * \%Operation_{1STAGE}\right] \\ + \left[FanEnergySavings_{NO-COOL} * (\%Operation_{NO-COOL} - \%Operation_{1STAGE})\right] \end{array} \right\} / FanEnergy_{Baseline}$$

EQUATION 5. ENERGY SAVINGS, AC-18, OUTSIDE AIR BIN 65 – 70°F

$$\%FanSavings = 100 * \left\{ \begin{array}{l} \left[244W * (14\%)\right] \\ + \left[816W * (100\% - 14\%)\right] \end{array} \right\} / 911W = 81\%$$

Equation 6 provides the fan energy savings for a two-stage cooling unit for a specific outside air temperature bin. Equation 7 uses this formula to calculate fan energy savings for AC-19 for outside air bin 65 - 70°F.

EQUATION 6. ENERGY SAVINGS, TWO-STAGE COOLING AC UNIT

$$\%FanSavings = 100 * \left\{ \begin{array}{l} \left[FanEnergySavings_{2STAGE} * \%Operation_{2STAGE}\right] \\ / FanEnergy_{Baseline2STAGE} + \\ \left[FanEnergySavings_{1STAGE} * (\%Operation_{1STAGE} - \%Operation_{2STAGE})\right] \\ / FanEnergy_{Baseline1STAGE} \\ + \left[FanEnergySavings_{NO-COOL} * (\%Operation_{NO-COOL} - \%Operation_{1STAGE})\right] \\ / FanEnergy_{BaselineNO-COOL} \end{array} \right\}$$

EQUATION 7. ENERGY SAVINGS, AC-19 OUTSIDE AIR BIN 65 – 70°F

$$\%FanSavings = 100 * \begin{Bmatrix} [1114W * 9\%] / 4952 + \\ [2556W * (15\% - 9\%)] / 4952 \\ + [3146 * (100\% - 15\%)] / 3486 \end{Bmatrix} = 82\%$$

Table 7 summarizes the fan energy savings for each AC unit, taking into account the hours each unit was measured operating with one stage and with both stages of cooling.

TABLE 7. WEIGHTED FAN ENERGY SAVINGS – ONE AND TWO-STAGE COOLING

	Outside Air Temperature Bin (°F)							
Unit	55 - 60	60 - 65	65 - 70	70 - 75	75 - 80	80 - 85	85 - 90	90 - 95
AC-18 (One-Stage)	89%	89%	81%	64%	38%	27%	25%	25%
AC-19 (Two-Stage)	90%	90%	82%	74%	51%	38%	30%	25%
AC-20 (Two-Stage)	91%	91%	89%	81%	65%	52%	40%	34%

Table 8 summarizes the measured power factor for AC-18, measured on May 7, 2013, for both baseline and measure. On a call for cooling, the power factor only decreases minimally, from 0.69 to 0.67. In fan-only mode, power factor decreases significantly when its speed is reduced to 40%, from 0.98 to 0.79.

TABLE 8. AC-18 POWER FACTOR MEASUREMENT

Fan Speed	Cooling (Y/N)	Power Factor
100% (Baseline)	N	.98
90% (Measure)	N	.97
75% (Measure)	N	.91
40% (Measure)	N	.79
100% (Baseline)	Y	.69
90% (Measure)	Y	.67

ERROR ANALYSIS

Fan amperage readings were recorded every 10 minutes, 24 hours per day. Each RTU operated between 8 and 10 hours per day, resulting in a sample size of 350 – 450 points per week. For each test of each air conditioning unit, these readings did not vary.

A sample size of 1700 voltage readings produced a mean voltage of 274.8, with 95% confidence that voltage would be between 274.6 and 274.9.

DISCUSSION

ENERGY SAVINGS

The stepped fan speed control provided supply fan energy savings at part load conditions. At OSA temperatures below 70°F, the AC units studied at this site operated in the cooling mode for only 0 – 15% of their total operating hours, which provided many hours of reduced energy operation.

As expected, observed operating hours in first and second stage cooling modes increased as OSA temperature increased, reducing the potential fan savings. At OSA above 90°F, 25 – 35% fan savings were observed, depending upon the AC unit.

Actual supply fan energy savings depends upon the number of hours that a packaged AC unit is operating in cooling or heating mode, versus in ventilation-only mode. These hours, in turn, are dependent upon several components of the cooling load, including:

- OSA temperature
- Solar loads
- People loads
- Lighting loads
- Miscellaneous equipment loads

In this test, results were highly dependent upon OSA temperatures. The internal cooling loads, like lighting, people, and miscellaneous equipment, were a small proportion of the overall cooling demand in the mall atrium. On the other hand, solar loads were a large proportion of the cooling load, due to the presence of large skylights in the conditioned spaces. OSA sensible cooling loads were also a large proportion of the overall load.

In other applications, internal cooling loads predominate, and energy savings fluctuate with those loads. Buildings with a small proportion of glass or a low overall envelope U-value, will have savings that are less dependent upon OSA temperatures.

Using Local climatic data from ASHRAE[3], annual electrical energy savings were estimated for the three studied air conditioning units. These estimates were based upon mall operating hours as follows:

Monday – Friday:	10:00 AM – 9:00 PM
Saturday:	10:00 AM – 10:00 PM
Sunday:	11:00 AM – 9:00 PM

Table 9 provides the hours of operation at various OSA temperatures, for the nearby Los Angeles International Airport. Table 10 adjust these hours to coincide with the mall atrium's hours of operation.

TABLE 9 ASHRAE BIN WEATHER – LOS ANGELES INTERNATIONAL AIRPORT

TEMPERATURE	DAY HOURS/YEAR (0900-1600)	NIGHT HOURS/YEAR (1600-2400)
55	353	415
60	319	355
65	286	300
70	279	254
75	267	196
80	245	144
85	202	98
90	167	75
95	99	45
100	40	19
105	5	2

TABLE 10 ANNUAL WEATHER, ADJUSTED FOR SITE HOURS OF OPERATION

Temperature	Adjusted Day Hours/Year	Adjusted Night Hours/Year	Total Hours/Year
55	265	311	576
60	239	266	506
65	215	225	440
70	209	191	400
75	200	147	347
80	184	108	292
85	152	74	225
90	125	56	182
95	74	34	108
100	30	14	44
105	4	2	5

Applying the weighted fan energy savings tabulated in Table 7 to these annual operating hours, Table 11, Table 12, and Table 13 estimate the annual supply fan energy savings for each air conditioning unit.

Table 11 Annual Fan Energy Savings – AC-18

Temperature	Baseline Supply Fan Energy (kW)	Measure Supply Fan Savings	Total Hours/Year	Savings (kWh/yr)
55	2.3	89%	576	1,178
60	2.3	89%	506	1,026
65	2.3	81%	440	811
70	2.3	64%	400	586
75	2.3	38%	347	300
80	2.3	27%	292	183
85	2.3	25%	225	130
90	2.3	25%	182	102
95	2.3	0%	108	-
100	2.3	0%	44	-
105	2.3	0%	5	-
			Savings	4,316
			Baseline	7,145
			% Savings	60%

Table 12 Annual Fan Energy Savings – AC-19

Temperature	Baseline Supply Fan Energy (kW)	Measure Supply Fan Savings	Total Hours/Year	Savings (kWh/yr)
55	5.0	90%	576	2,574
60	5.0	90%	506	2,259
65	5.0	82%	440	1,781
70	5.0	74%	400	1,472
75	5.0	51%	347	879
80	5.0	38%	292	555
85	5.0	30%	225	332
90	5.0	25%	182	225
95	5.0	0%	108	-
100	5.0	0%	44	-
105	5.0	0%	5	-
			Savings	10,076
			Baseline	15,468
			% Savings	65%

TABLE 13 ANNUAL FAN ENERGY SAVINGS – AC-20

Temperature	Baseline Supply Fan Energy (kW)	Measure Supply Fan Savings	Total Hours/Year	Savings (kWh/yr)
55	3.8	91%	576	1995
60	3.8	91%	506	1751
65	3.8	89%	440	1497
70	3.8	81%	400	1240
75	3.8	65%	347	862
80	3.8	52%	292	581
85	3.8	40%	225	348
90	3.8	34%	182	234
95	3.8	0%	108	0
100	3.8	0%	44	0
105	3.8	0%	5	0
			Savings	8508
			Baseline	11952
			% Savings	71%

DEMAND REDUCTION

The supply fan control technology provides energy reductions at partial cooling loads. However, at full cooling, AC unit supply fans operated at full speed, providing no energy reduction. This full cooling operation would usually coincide with higher outside air temperatures, when there would be a call for demand reduction.

The manufacturer of the system tested in this report stated that packaged AC units provide more efficient cooling when their fans operate at 90% flow, rather than baseline 100% flow. This project did not test that hypothesis, because field conditions did not provide cooling loads that could be controlled closely enough to compare sensible cooling performance at different fan speeds.

The technologies available on the market have the capability to lock out the second stage of cooling during a demand response event. Supply fan speed control would add fan savings to the compressor savings during these events. For example, AC-19 supply fan would decrease from 1,524 to 905W, or 41%, if second stage of cooling were locked out.

INDOOR AIR QUALITY

In addition to providing space temperature control, the packaged AC unit introduces outside air into the conditioned space. The supply fan speed control technologies available on the market also control the position of the return air (RA) and OSA dampers, which prevents the reduction of OSA flow when the supply fan speed is reduced. Air flow is measured in Cubic Feet per Minute CFM. There are several methods used to reset OSA CFM with changing fan speed:

- Temperature balance – The mixed air temperature is dependent upon the percentage of OSA CFM, the RA temperature, and the OSA temperature. Theoretically, because the OSA, RA, and mixed air temperatures are measured by the control systems available, the OSA amount can be adjusted by a control algorithm. In reality, RTUs do not have a good location for accurate reading of mixed air temperature. Sensors are either too close to the upstream face of the cooling coil, or in a location where incomplete mixing of RA and OSA has occurred. Also, as OSA temperature approaches RA temperature, as on temperate days, the temperature difference is too low for accurate control.
- Damper position – Controls can open OSA dampers to a specified percentage of fully open. CFM is not directly proportional to damper position, though. A large proportion of OSA would be introduced into the system when the OSA damper is opened a small percentage.
- Air balancing – If the control technology has discrete fan speeds, an air balance contractor can set the OSA damper position to provide proper CFM at each of these steps. These damper positions can then be entered into the control system, to provide proper ventilation at each operating condition.

Of the three methods listed above for controlling OSA flow, the air balance method is the most reliable. It does, however, add the cost of the air balancing contractor to the project, which may create a market barrier.

DEMAND CONTROL VENTILATION

The available supply fan speed control technologies can provide demand control ventilation (DCV). DCV is a control strategy that adjusts the amount of OSA introduced to a space, indirectly based upon occupancy. Because an AC unit's control system cannot directly measure the number of people in a space, it uses CO_2 levels to approximate this occupancy. Control systems compare space CO_2 levels with OSA levels, and control the OSA damper position to maintain CO_2 level 600 parts per million (ppm) above OSA levels. As the space's number of occupants increases, the OSA amount increases. CO_2 sensors are normally located between 3 and 6 feet above the floor (the breathing zone).

The 2010 Title 24, recognizes the benefit of DCV, and requires this control in systems serving spaces with densities greater than 25 people per 1,000 square feet (sf). Classrooms, call centers, healthcare buildings, and public areas of social service buildings are exempted from this requirement. Title 24 sets the minimum OSA amount at 0.15 CFM/sf, so DCV cannot be applied to office spaces. Table 14 provides a few examples from the 2010 California Mechanical Code of high occupancy areas where DCV may be applied.

TABLE 14. TITLE 24 OUTSIDE AIR REQUIREMENTS

Room Type	CFM/sf	Occupant Density (person/1,000 sf)	CFM/Person	Total CFM/sf
Stage, Studio Area	0.06	70	10	0.76
Mall Common Areas	0.06	40	7.5	0.36
Conference Room	0.06	50	5	0.31
Worship Area	0.06	120	5	0.66

DCV was reviewed as part of this study. However, the CO_2 levels measured in the RA systems of the AC units never reached a level where DCV would be a factor. Table 15 shows the maximum CO_2 levels measured in RA of each AC unit. DCV control strategies are based upon an OSA CO_2 level of 400 ppm and a space CO_2 level of 1,000 ppm. During this test, OSA CO_2 levels averaged 66 ppm, with maximum measured levels of 100 ppm.

TABLE 15. RETURN AIR MAXIMUM MEASURED CARBON DIOXIDE LEVELS

UNIT#	MAXIMUM MEASURED RETURN AIR CO_2 LEVEL (PPM)
AC-18	358
AC-19	303
AC-20	323

MARKET BARRIERS

Retrofit control systems for RTUs are provided by several manufacturers. The installation of these systems is straightforward for a trained HVAC technician. Some potential market barriers to this technology include:

- Replacement with a new RTU. If an existing RTU is nearing the end of its useful economic life, it can be replaced by a newer, more efficient unit. New RTUs are available with variable speed supply fans and compressors. The retrofit controls are applicable to an existing RTU that will remain in service for several more years.
- Air balance cost – A properly installed and commissioned retrofit controls system should be air balanced at each supply fan speed, and the OSA setting at each speed should be programmed. This involves the labor cost of measuring and balancing the existing system.

CONCLUSIONS

The field assessment showed RTU supply fan energy savings when a retrofit system controlled fan speed based upon cooling demand. Application of this technology did not adversely affect the performance of the AC systems. Care must be taken to ensure that proper outside air CFM is introduced into the conditioned space, if a retrofit control system is installed.

Additional energy savings can be gained by using a retrofit control system to provide DCV, in market sectors where this is allowed. Assembly areas such as auditoria, theaters, meeting rooms, and churches are applications where DCV would apply. Office spaces would not be candidates for DCV, due to the low occupant density.

RECOMMENDATIONS

Based upon the results of this assessment, retrofit control systems that adjust the RTU's supply fan speed in response to cooling and heating demand is recommended.

While this assessment only studied the performance of varying supply fan speed in one application, in one climate zone, its energy savings would be reflected in other market segments. To transfer this technology into the EE program, energy simulations need to be performed to calculate energy savings for each market segment, in each climate zone.

A work paper was submitted to the California Public Utilities Commission's (CPUC) Energy Division (ED) by Portland Energy Conservation Incorporated (PECI) in June 2011, on SCE's behalf. This work paper presented the results of an eQuest computer simulation of stepped fan control for RTUs. The paper's results covered all of the climate zones in SCE territory. Assembly, Education, Restaurant, Retail, and Office applications were modeled in that analysis. The energy model included DCV for market sectors where Title 24 allows this control strategy. The model maintained proper OSA flow when the supply fan speed was reduced.

Once the review of that work paper is complete, its underlying eQuest model can be altered to estimate the energy savings of stepped fan control without DCV, for the commercial market sectors that use RTUs. Those analytical results can be compared to actual measured savings of this report.

APPENDIX A – INSTRUMENTATION

Table 16 provides the specifications and calibration dates for all sensors used in this project. Calibration of all instruments occurred prior to conducting any tests.

TABLE 16. SPECIFICATIONS, CALIBRATION DATES, LOCATIONS, AND CORRESPONDING MONITORING POINTS FOR SENSORS

SENSOR TYPE	MAKE/MODEL	ACCURACY (NIST TRACEABLE)	CORRESPONDING KEY MONITORING POINTS
Compressor Current	Veris H921	±2%	• AC-18 (1 comp) • AC-19 (2 comp) • AC-20 (1 comp)
Fan Current	Veris H921	±2%	• AC-18, AC-19, AC-20
Outside Air Temperature, Dry Bulb and Wet Bulb	BAPI BA/10K-2-H200-O-WP	±0.3 °C (±0.5 °F),±2% Relative Humidity	• Mounted to outside of AC unit
Return Air, Mixed Air, and Supply Air Temperatures	BAPI BA/10K-2-D-8-WP	±0.3 °C (±0.5 °F)	• AC-18 • AC-19 • AC-20
Outside Air CO_2 Level	Veris CWE	±30 ppm, 0 – 2000 ppm range	• Mounted to outside of AC unit
Return Air CO_2 Level	Veris CWE	±30 ppm, 0 – 2000 ppm range	• AC-18 • AC-19 • AC-20
Power Factor	Fluke #435	±0.1%, 0 – 1 range	• AC-18

REFERENCES

1. 2010 California Energy Code, California Title 24, Part 6, California Building Standards Commission.
2. 2010 California Mechanical Code, California Title 24, Part 4, California Building Standards Commission
3. ASHRAE Climatic Data for Region X Arizona, California, Hawaii, Nevada, November 1994

[i] 2003 Commercial End Use Survey, California Energy Commission