Emerging Technologies

# Laboratory Testing of Variable Speed Compressor and Fan Controls for RTU Optimization

*ET15SCE7030*

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## Acknowledgments

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## Disclaimer

# EXECUTIVE SUMMARY

This report describes a laboratory evaluation to optimize the performance and efficiency of a packaged air-conditioning and heating roof top unit (RTU) with a variable speed electrically communicated motor (ECM) controlling the blower speed and a retrofitted variable speed drive controlling the compressor speed. This technology is being considered as a method to save energy associated with air conditioning at part load conditions. It also may have potential to reduce peak electricity demand in buildings with over-sized equipment or when combined with condenser-air evaporative pre-cooling technology.

Equipment capable of modulating delivered air conditioning capacity to a building is important because the building load changes based upon use and outdoor air temperature. Furthermore, engineers specify air conditioning equipment to meet the air conditioning load on a "design day", meaning the summer peak temperature that will only be exceeded 1% of the time. The result is the other 99% of the hours in the year the equipment will be oversized, a problem exacerbated by the fact that the capacity of the air conditioner increases at the outdoor temperature decreases, resulting in compressor cycling. Variable fan and compressor controls allow the unit to operate at reduced capacity when possible with reduced refrigerant flow and air flow, while benefiting from using the heat exchangers sized for the full capacity system. This technology has the potential to save large amounts of energy, but could also reduce peak demand in cases where the RTU was oversized to meet the demand of a peak day. In a study of 145 RTUs in Northern California, Felts and Bailey, found that "In at least 40% of the cases, the unit size could be dropped by 50% or more" while still meeting peak demand [1]. Adding a variable speed drive and RTU optimization controls could "right size" the unit for the building and significantly increase its efficiency. The technology could also reduce peak demand when combined with condenser-air evaporative pre-cooling technology, which drops the inlet air temperature to the condenser so the RTU observes a part-load condition.

This project seeks to conduct laboratory testing and analysis of RTU retrofit control strategies. While multiple manufacturers make controllers that fit into this category, Western Cooling Efficiency Center (WCEC) evaluated the potential energy and demand savings that could be achieved from these products if the control strategy were optimized. WCEC did not evaluate a specific commercial product. The control strategies investigated here could be implemented by any retrofit controller manufacturer or even the existing RTU controller or building management system.

WCEC retrofitted a 4-Ton packaged RTU with a variable frequency drive (VFD) on the compressor. The VFD allowed control of the compressor to operate between 30-72 Hz (50-120% of its normal operating speed). The RTU also contained a selectable speed electronically-commuted (ECM) evaporator fan motor. Testing was performed at each of the five available speed settings. A total of 63 tests were performed across a range of compressor speeds, evaporator fan speeds, and outside air dry bulb temperatures. Return air conditions were held constant throughout the testing. Instrumentation and ducting were added to the RTU but no other physical changes were made. For each steady-state test, total capacity, sensible capacity, power, and coefficient of performance were recorded and analyzed.

The results show that the part-load energy saving potential of this technology is significant. The baseline RTU tested in the WCEC laboratory consumed 4.16kW of power and had an operating COP of 3.06 at typical rating conditions (Table 1Table 5). When the outdoor air temperature drops to 75°F, the baseline RTU would continue to operate at the same compressor and fan speed. The capacity would increase 15%, which is likely to be completely unnecessary because the thermal load on the building would decrease simultaneously, causing additional cycling of the air conditioner. The power would decrease 16% and the COP would increase 37%. With the advanced controller, the compressor and fan speed could be reduced at the part load condition (Table 1). The scenario shown reduces the power over the baseline operation at 75°F by 25% and increases the COP by 11%.

**TABLE 1: EXAMPLE OPERATING SCENARIO AND ENERGY SAVING POTENTIAL**

| | DESIGN CONDITION (BASELINE) | PART LOAD CONDITION (BASELINE) | PART LOAD CONDITION (ADVANCED CONTROLS) |
|---|---|---|---|
| Outdoor Air Temperature (°F) | 95 | 75 | 75 |
| Compressor Speed (%) | 100 | 100 | 80 |
| Fan Speed | Medium | Medium | Medium-Low |
| Sensible Heat Ratio | 76% | 71% | 73% |
| System Capacity (Btu/hr) | 43,466 | 49,928 | 41,112 |
| System Power (kW) | 4.16 | 3.50 | 2.61 |
| System COP | 3.06 | 4.18 | 4.62 |

This technology would be extremely beneficial when combined with an evaporative condenser-air pre-cooler technology. In dry California climates, evaporative condenser-air pre-coolers reduce the effective outdoor temperature seen by the condensing unit. At a peak condition, it is possible to reduce the outdoor air temperature 20°F or more. Using a combination of an evaporative condenser-air pre-cooler and fan and compressor speed controls could reduce peak electricity demand by more than 35%. Achieving these results would require retrofitting the existing RTU with evaporative pre-cooler, variable speed drive(s), and controller. A package installation of these technologies would increases savings in comparison to installing a single technology and reduce the transaction cost associated with the RTU retrofit. However, there are challenges associated with this technology that result in market barriers. A significant barrier is that no known manufacturer is manufacturing a comprehensive pre-cooler and RTU optimizer solution. WCEC hopes to reduce this barrier by identifying potential manufacturers to participate in technology development and deployment. Secondly, installation of the system would require a licensed electrician, which increases installation costs. Lastly, installation of a pre-cooler faces the barriers associated with evaporative pre-coolers including access to water and sewer lines and regular maintenance requirements.

WCEC recommends further development of this technology, including engaging with potential manufacturing partners. Furthermore, WCEC recommends conducting laboratory and field tests of the technology in combination with an evaporative pre-cooler technology that has already been tested with positive results.

# ABBREVIATIONS AND ACRONYMS

| | |
|---|---|
| AHRI | Air-Conditioning, Heating, and Refrigeration Institute |
| ASHRAE | American Society for Heating, Refrigeration, and Air Conditioning Engineers |
| CFM | Cubic Feet per Minute |
| COP | Coefficient of Performance |
| CS | Cold Side |
| DB | Dry Bulb |
| EA | Exhaust Air |
| ECM | Electrically Commutated Motor |
| HS | Hot Side |
| OA | Outside Air |
| NI | National Instruments |
| RA | Return Air |
| RTU | Roof Top Unit |
| SA | Supply Air |
| SCE | Southern California Edison |
| SEER | Seasonal Energy Efficiency Ratio |
| SHR | Sensible Heat Ratio |
| RTD | Resistance Temperature Device |

| | |
|---|---|
| VFD | Variable Frequency Drive |
| WB | Wet Bulb |
| WCEC | Western Cooling Efficiency Center |

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# INTRODUCTION

This report describes a laboratory evaluation to optimize the performance and efficiency of a packaged air-conditioning and heating roof top unit (RTU) with a variable speed electrically communicated motor (ECM) controlling the blower speed and a retrofitted variable speed drive controlling the compressor speed. This technology is being considered as a method to save energy associated with air conditioning at part load conditions. It is also may have potential to reduce peak electricity demand in buildings with over-sized equipment or when combined with condenser-air evaporative pre-cooling technology.

## BACKGROUND

Packaged air-conditioning and heating RTUs are ubiquitous in California and provide an estimated 75% of the cooling to commercial buildings in this state [2]. While no specific data set on the subject was found, Western Cooling Efficiency Center (WCEC) has observed that existing RTUs capable of providing continuously variable cooling capacity to the building are rare. This capability requires installation of a variable speed compressor and evaporator fan. WCEC has observed existing installations of equipment with staged compressors, but this falls short of delivering true variable capacity air conditioning to the building. It should be noted that other types of variable capacity cooling systems for buildings exist in the remaining 25% of buildings not cooled by packaged RTUs. These technologies include variable refrigerant flow (VRF) or chilled water distribution systems.

Equipment capable of modulating delivered air conditioning capacity to a building is important because the building load changes based upon use and outdoor air temperature. Furthermore, engineers specify air conditioning equipment to meet the air conditioning load on a "design day", meaning the summer peak temperature that will only be exceeded 1% of the time. The result is the other 99% of the hours in the year the equipment will be oversized. This problem is exacerbated by the fact that the capacity of the air conditioner increases at the outdoor temperature decreases, resulting in compressor cycling.

Variable fan and compressor controls allow the unit to operate at reduced capacity when possible with reduced refrigerant flow and air flow, while benefiting from using the heat exchangers sized for the full capacity system. This technology has the potential to save large amounts of energy. It could also reduce peak demand in cases where the RTU was oversized even for a peak day. In a study of 145 RTUs in Northern California, Felts and Bailey, found that "In at least 40% of the cases, the unit size could be dropped by 50% or more" while still meeting peak demand [1]. Adding a variable speed drive and RTU optimization controls could "right size" the unit for the building and significantly increase its efficiency. The technology could also reduce peak demand when combined with condenser-air evaporative pre-cooling technology, which drops the inlet air temperature to the condenser so the RTU observes a part-load condition.

# ASSESSMENT OBJECTIVES

The objective of this project was to conduct laboratory testing and analysis of RTU retrofit control strategies. WCEC evaluated the potential energy and demand savings that could be achieved from these products if the control strategy were optimized. WCEC did not evaluate a specific commercial product; although multiple manufacturers make controllers that fit into this category. The control strategies investigated here could be implemented by any retrofit controller manufacturer or even the existing RTU controller or building management system.

# TECHNOLOGY DESCRIPTION

There are variety of aftermarket products aimed at reducing cooling energy usage and/or peak demand in packaged roof top units (RTUs) by incorporating variable speed drives on compressors and/or fans. Generally speaking, these products control the supply fan speed on an air conditioner to reduce capacity and increase efficiency at non-peak times. One "side-effect" of this strategy is that the reduced airflow increases latent cooling of the air stream, generally considered unnecessary in dry Western Climates. At least one known product reduces the compressor speed to match the fan speed. Generally speaking, reducing the RTU compressor speed should increase evaporator temperatures and reduce latent cooling. In addition, reducing the speed of an RTU's compressor effectively results in a smaller capacity RTU with oversized heat exchangers. Reducing the RTU compressor speed has the potential to improve efficiency and provide demand reductions.

Preliminary research conducted by various utilities and research organizations suggests that variable fan and compressor speed technology has the potential to offer significant savings, but more research is necessary to identify and optimize the specific components and control strategies that provide the greatest benefit. The large number of independent variables complicate this analysis. The power draw, capacity, and efficiency of the air conditioner is affected by outdoor air dry bulb temperature, return air dry bulb and wet bulb temperatures, fan speed, and compressor speed. This project consists of parametric testing of a 'standard' packaged rooftop air conditioner operating at various evaporator fan and compressor speeds.

# TECHNICAL APPROACH/TEST METHODOLOGY

## OVERVIEW

WCEC retrofitted a new 4-ton packaged RTU purchased in 2013 with a variable frequency drive (VFD) on the compressor. The RTU met the 2008 federal minimum efficiency standards and contained R-410A as the refrigerant. The VFD allowed control of the compressor to operate between 30-72 Hz (50-120% of its normal operating speed). The RTU also contained a selectable speed electronically-commuted (ECM) evaporator fan motor and testing was performed at each of the five available speed settings (1,200-1,800 CFM). Tests were performed across a range of compressor speeds, evaporator fan speeds, and outside air dry bulb temperatures. Return air conditions were held constant throughout the testing. Instrumentation and ducting were added to the RTU but no other physical changes were made. The RTU was run in recirculation mode only with no outdoor ventilation air intake. In actual installations, a RTU modified to run with multiple fan speeds would also require a modulating damper to control outdoor ventilation air flow.

The experimental protocol was based on the 2008 AHRI 210/240 standard 'Performance Rating of Unitary Air-Conditioning and & Air-Source Heat Pump Equipment' [3]. This standard specifies that packaged air conditioners be tested in an environmental chamber with controlled cold side (indoor condition) and hot side (outside condition) air loops. The indoor air is to be controlled for flow rate, temperature, and humidity whereas the outdoor air is controlled only for flow rate and temperature. All of the testing performed by the WCEC utilized the AHRI specified standard return air condition of 80°F dry bulb, 67°F wet bulb. Outside air temperature was controlled to 75°F, 85°F, and 95°F, depending on the requirement for each test. Each test was run for minimum one hour and controlled to within the tolerances specified by AHRI 210/240.

Each test was performed at a constant fan and compressor speed setting for the duration of the test. Compressor speed settings were chosen to cover the entire operable range at a given fan speed setting. At low fan speeds, the compressor could be turned down to 50% but was not operated above 100% to prevent risk of evaporator coil icing. At high fan speeds the lower limit on the compressor speed was set by the system's ability to provide dehumidification, since the cold side air loop had no other means of drying the air and thus required slight dehumidification from the RTU to maintain a 67°F wet bulb temperature.

## ENVIRONMENTAL CHAMBER

Figure 1 illustrates the design of the WCEC's environmental testing chambers. Two environmental chambers maintain both the simulated outdoor air condition and the indoor air condition. The outdoor air chamber is capable of heating, cooling, humidification, and de-humidification. The indoor air chamber is capable of heating and humidification. The temperature and dew point of the inlet and exhaust air from both chambers were measured with a 4-wire RTD temperature probe and chilled-mirror hygrometer, respectively. Airflow measurements for both chambers were measured using a differential pressure measurement across a calibrated nozzle box.

## CHAMBER CONTROL

The outdoor air chamber was used to control the dry bulb temperature supplied to the condensing unit. The control process for the outdoor air chamber was as follows (Figure 1):

1. The differential pressure across the condenser of test unit was measured in free-air (no ducting attached). This pressure was used a proxy measurement for airflow.
2. The exhaust air from the test unit was ducted to the chamber exit. The test unit was turned on.
3. The conditioning ducting was configured to run in re-circulation mode. The outside air intake and exhaust dampers were closed and the recirculation damper was opened.
4. Because only temperature control was required, the damper to the dryer was closed and the recirculation air was routed through the bypass with the humidifier disabled.
5. The speed on the chamber blower was increased until the differential pressure across the condenser of the test unit matched the measurement in Step 1. The blower speed was fixed for the remainder of the tests.
6. A control loop adjusted the cold water flow through a proportional valve serving the cooling coil until the target dry bulb temperature was reached.

The indoor air conditioning loop was used to heat and humidify the air supplied by the test unit. The control process for the indoor air chamber was as follows:

1. The supply and return to the test unit were ducted and the test unit was turned on.
2. The bypass damper for the humidifier was closed and all air was passed through the humidifier.
3. The speed on the chamber blower was set by following the procedure described in section 6.1.3.3.1.1 of ANSI/AHRI Standard 210/240-2008 [3].
4. The differential pressure across the humidification path was measured.
5. A control loop closed the damper for the humidifier until the target dew point was reached.
6. A control loop opened the damper for the humidification bypass to maintain the differential pressure measured in Step 4. This maintained the conditioning system at a fixed resistance.
7. A control loop adjusted the hot water flow through a proportional valve serving the heating coil until the target dry bulb temperature was reached.

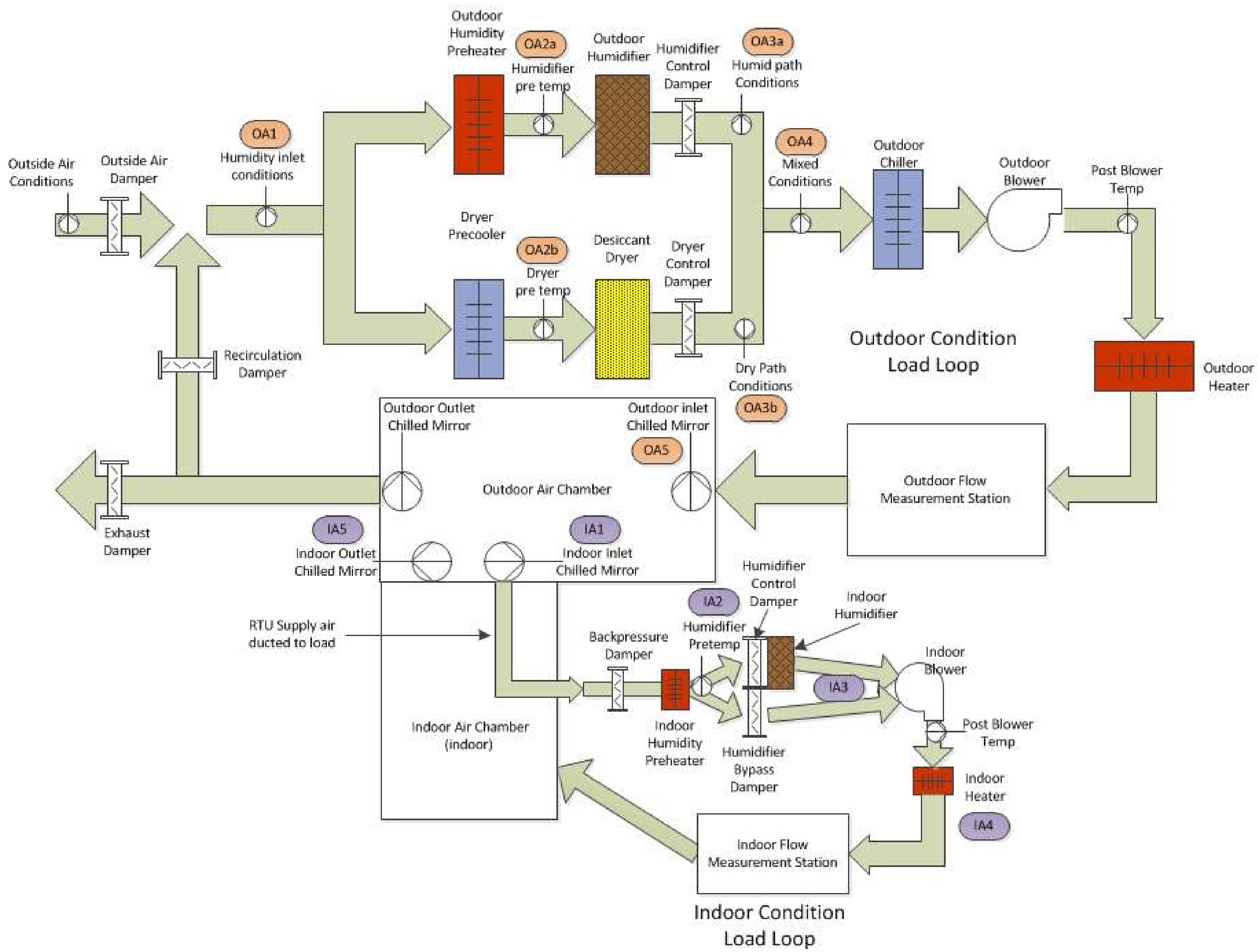


**FIGURE 1: SCHEMATIC OF TEST CHAMBERS AND BOTH INDOOR AND OUTDOOR CONDITIONING LOOPS**

# TEST PLAN

The testing procedures used were based on 2008 AHRI 210/240 standard 'Performance Rating of Unitary Air-Conditioning and & Air-Source Heat Pump Equipment' [3], with minor adjustments made to accommodate the variable speed compressor and evaporator fan operation. All tests were conducted for a minimum of one hour, with final results calculated based upon the last half hour of operation. Steady state conditions were ensured by adhering to the temperature tolerances (both dry bulb and wet bulb) set forth in AHRI 210/240.

Using flexible ducting, the RTU was installed in the hot ('outdoor') chamber with supply and return paths ducted into the cold side chamber. The condenser air was drawn in freely from within the hot side chamber and then exhausted through a flexible duct hose into the hot-side conditioning loop. Hot side airflow was calibrated to be equivalent to the airflow which would normally be drawn through the condenser coil by the RTU's internal fan. The hot side blower ('outdoor blower') has a variable speed drive and was set to maintain the same pressure drop across the condenser coil as was measured with the unit operating without any attached ducting on the condenser exhaust (the configuration it would be in for a typical rooftop installation).

The RTU's supply and return air streams were ducted to the cold side loop and, using guidelines set forth in ASHRAE Standard 41.2-1987 'Standard Methods for Laboratory Airflow Measurement', temperature pressure and flow measurements were conducted. A mixing device was installed upstream of the supply temperature measurement to ensure that the air stream was not stratified and accurate temperature measurements could be obtained across a range of flow rates.

A total of sixty-three tests were conducted (Table 2) varying the following parameters for the RTU: outdoor air temperature, compressor speed, and fan speed. The indoor air condition was 80°F/67°F DB/WB for all tests.

**TABLE 2: TEST MATRIX**

| TEST NUMBER | OUTDOOR AIR TEMPERATURE | COMPRESSOR SPEED (%) | FAN SPEED |
|---|---|---|---|
| 1.1, 1.2, 1.3, 1.4 | 95 | 60, 80, 100, 120 | High |
| 2.1, 2.2, 2.3, 2.4 | 95 | 60, 80, 100, 120 | Medium-High |
| 3.1, 3.2, 3.3. 3.4, 3.5 | 95 | 50, 60, 80, 100, 120 | Medium |
| 4.1, 4.2, 4.3, 4.4 | 95 | 50, 60, 80, 100 | Medium-Low |
| 5.1, 5.2, 5.3, 5.4 | 95 | 55, 60, 80, 100 | Low |
| 6.1, 6.2, 6.3, 6.4 | 85 | 60, 80, 100, 120 | High |
| 7.1, 7.2, 7.3, 7.4 | 85 | 60, 80, 100, 120 | Medium-High |
| 8.1, 8.2, 8.3. 8.4, 8.5 | 85 | 50, 60, 80, 100, 120 | Medium |
| 9.1, 9.2, 9.3, 9.4 | 85 | 50, 60, 80, 100 | Medium-Low |
| 10.1, 10.2, 10.3, 10.4 | 85 | 50, 60, 80, 100 | Low |
| 11.1, 11.2, 11.3, 11.4 | 75 | 60, 80, 100, 120 | High |
| 12.1, 12.2, 12.3, 12.4 | 75 | 60, 80, 100, 120 | Medium-High |
| 13.1, 13.2, 13.3. 13.4, 13.5 | 75 | 50, 60, 80, 100, 120 | Medium |
| 14.1, 14.2, 14.3, 14.4 | 75 | 50, 60, 80, 100 | Medium-Low |

| Test Number | Outdoor Air Temperature | Compressor Speed (%) | Fan Speed |
|---|---|---|---|
| 15.1, 15.2, 15.3, 15.4 | 75 | 50, 60, 80, 100 | Low |

## Fan Speed Control

Testing was designed to simulate the conditions of an RTU connected to a fixed ducting network. Thus, as evaporator fan speed was adjusted, an external circulator fan on the cold side loop was adjusted to maintain a second order pressure-flow relationship between all of the test points (Figure 2). The cold side loop's circulation fan (indoor fan) was controlled by a variable speed drive adjustable in tenths of hertz. The RTU's nominal flow rate specified by the manufacturer of 1550 CFM was used to develop the pressure-flow curve used for simulating a fixed ducting network. With the indoor fan speed set to maintain 1550 CFM through the RTU when operating at AHRI standard conditions (80°F dry bulb, 67°F wet bulb) the RTU's external static pressure (pressure difference between supply and return duct) was measured. A second order curve was fit through this point and (0,0). For all subsequent testing at varied RTU fan speeds and outdoor air (OA) temperature, the indoor blower speed was adjusted so that the cold side flow rate and external static pressure remained on the fixed ducting curve.

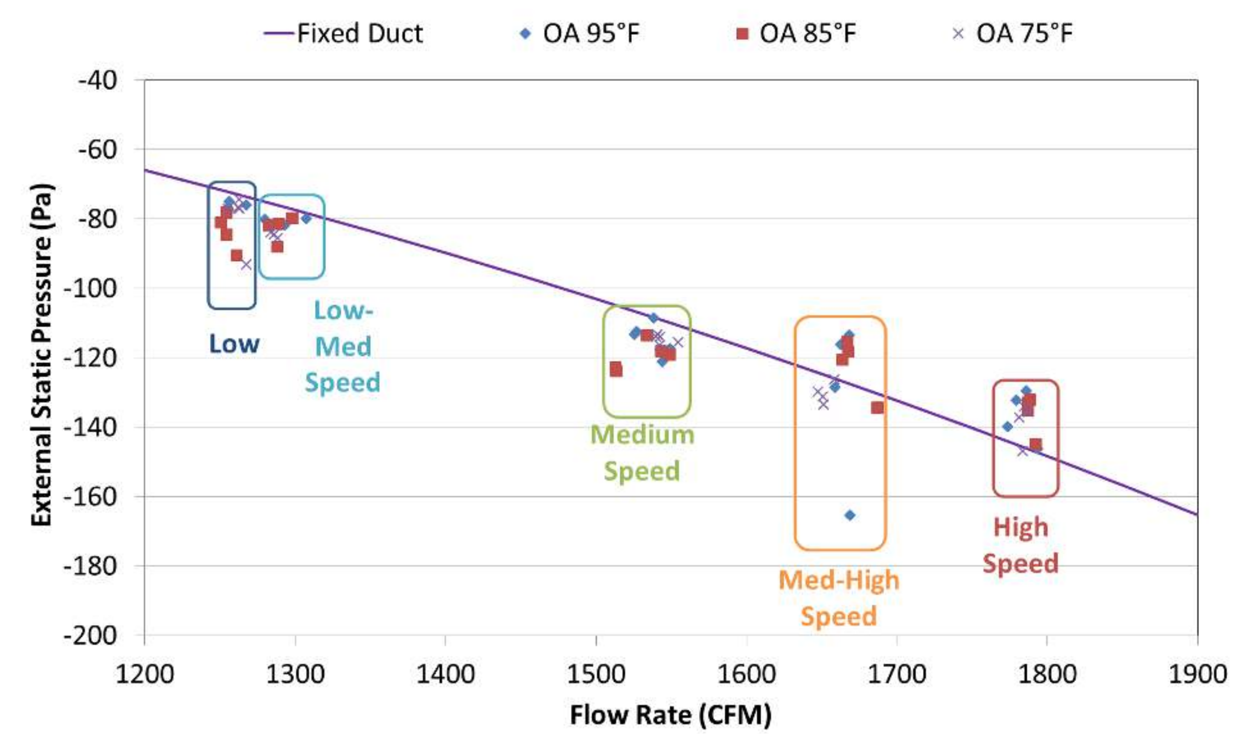


**Figure 2: Relationship between flow rate and external static pressure for varied RTU fan speeds**

## COMPRESSOR SPEED CONTROL

Compressor speed was varied between 50-120% of the compressor's normal operating frequency, where 100% speed equaled 60Hz. The lower limit of this range was based on the following observations: 1) the RTU's COP dropped off rapidly below 60-80% speed (varying slightly with other parameters),
2) the RTU's compressor current draw was high at low speeds, and 3) latent capacity was reduced to zero or near zero which made cold side humidity control difficult. The compressor had a low pressure switch to protect against damage to the compressor at low suction pressures, however, this was never tripped. The upper limit was chosen to help better understand the effects of varying compressor speed without damaging the compressor. The range of compressor speeds tested varied with evaporator fan speed, since low compressor speeds could only provide latent cooling capacity while operating at low air flow rates, and there was a perceived risk that operating at high compressor speeds at low flow rates could lead to evaporator coil icing. When possible, tests were conducted at compressor speed increments of 20 hertz, although exceptions were made due to the difficulties with extremely low or extremely high compressor speed operation listed above.

# INSTRUMENTATION PLAN

The RTU was placed inside the conditioned chamber and used for all tests (Figure 3). The measurements are color coded; light blue sensors measure differential pressure, orange sensors measure temperature, green sensors measure pressure, grey sensors measure air properties, purple sensors measure power, and the red sensor measures condensate generation (Figure 4).

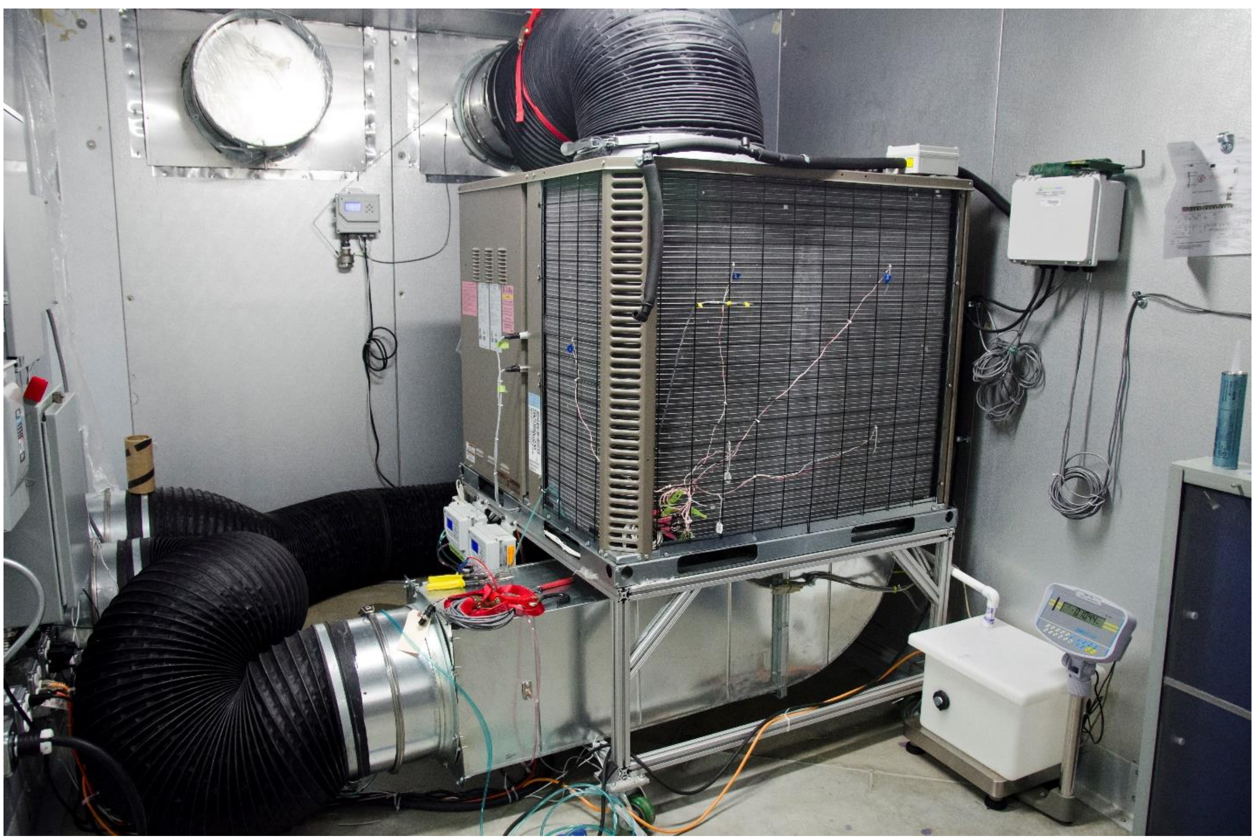

**FIGURE 3: TEST UNIT INSTALLED IN THE ENVIRONMENTAL CHAMBER**

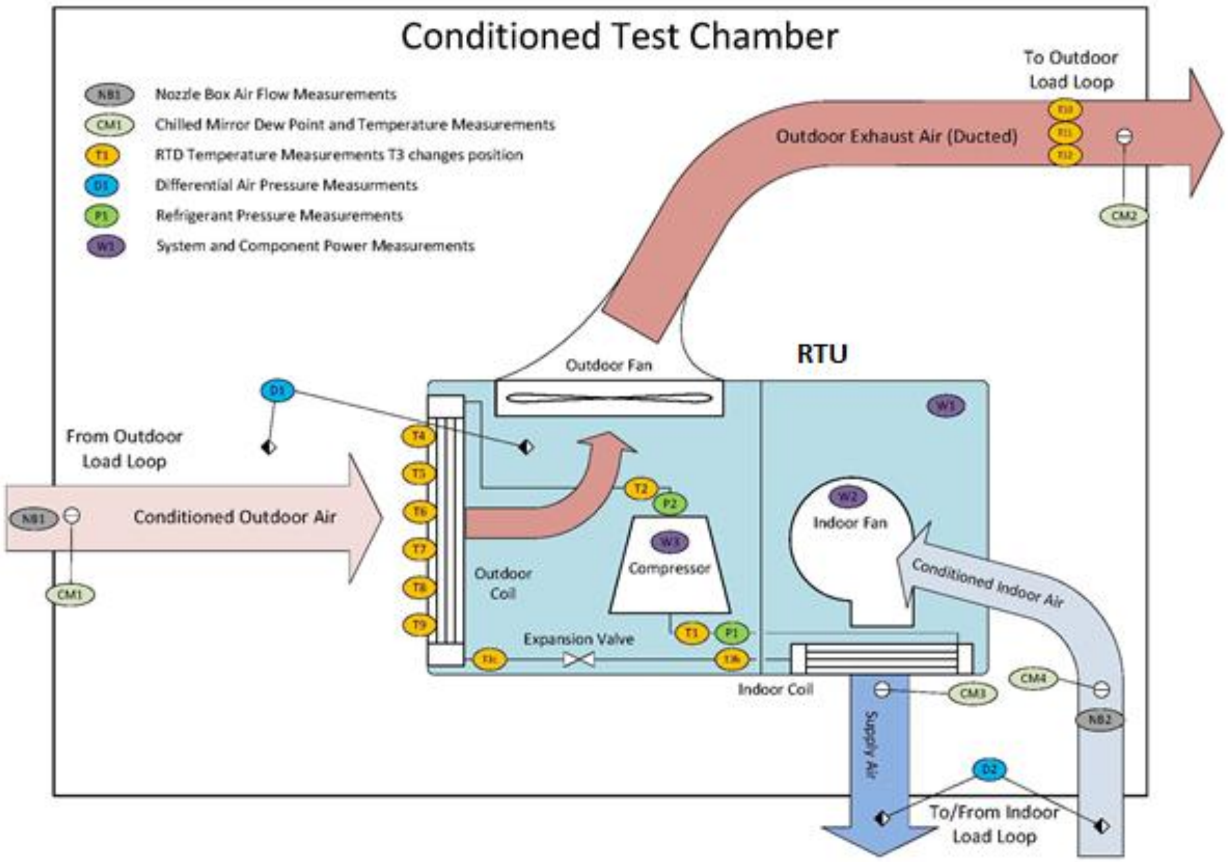


**FIGURE 4: MEASUREMENTS FOR TESTING APPARATUS**

## TABLE 3: TABLE OF INSTRUMENTS

| MEASUREMENT TYPE | SENSOR TYPE | MANUFACTURER AND MODEL # | ACCURACY | SIGNAL TYPE | DAQ CHANNEL | CALIBRATION DATE |
|---|---|---|---|---|---|---|
| Inlet Outdoor Air Temp | RTD | GE Optisonde | ±0.3°F | RS-232 | Serial | 03/13/2012 Serial #:0670312 |
| Inlet Outdoor Air Dew Point Temp | Chilled Mirror | GE Optisonde | ±0.4°F | RS-232 | Serial | 03/13/2012 Serial #:0670312 |
| Exhaust Outdoor Air Temp | RTD | GE Optisonde | ±0.3°F | RS-232 | Serial | 1/13/2014 Serial #:0051213 |
| Exhaust Outdoor Air Dew Point Temp | Chilled Mirror | GE Optisonde | ±0.4°F | RS-232 | Serial | 1/13/2014 Serial #:0051213 |
| Return Indoor Air Temp | RTD | GE Optisonde | ±0.3°F | RS-232 | Serial | 01/30/2014 Serial #:0291113 |
| Return Indoor Air Dew Point Temp | Chilled Mirror | GE Optisonde | ±0.4°F | RS-232 | Serial | 01/30/2014 Serial #:0291113 |
| Supply Indoor Air Temp | RTD | GE Optisonde | ±0.3°F | RS-232 | Serial | 3/18/2013 Serial #:0690113 |
| Supply Indoor Air Dew Point Temp | Chilled Mirror | GE Optisonde | ±0.4°F | RS-232 | Serial | 3/18/2013 Serial #:0690113 |
| Delta P Static (Condenser) | Differential Pressure | Energy Conservatory DG-500 | 1% of reading | RS-232 | Serial | |
| Delta P Static (RTU Fan) | Differential Pressure | Energy Conservatory DG-500 | 1% of reading | RS-232 | Serial | |
| Upstream Flow Nozzle Pressure (Indoor Side) | Differential Pressure | Energy Conservatory APT | 1% of reading | RS-232 | Serial | |
| Flow Nozzle Differential Pressure (Indoor Side) | Differential Pressure | Energy Conservatory APT | 1% of reading | RS-232 | Serial | 7/23/2013 Serial #CR6547 |
| Upstream Flow Nozzle Pressure (Outdoor Side) | Differential Pressure | Energy Conservatory APT | 1% of reading | RS-232 | Serial | |
| Flow Nozzle Differential Pressure (Outdoor Side) | Differential Pressure | Energy Conservatory APT | 1% of reading | RS-232 | Serial | |
| Indoor Chamber Static Pressure | Differential Pressure | Energy Conservatory APT | 1% of reading | RS-232 | Serial | |
| Outdoor Chamber Static Pressure | Differential Pressure | Energy Conservatory APT | 1% of reading | RS-232 | Serial | |
| Atmospheric Pressure | Atmospheric Pressure | OMEGADYNE PX409-26BI | ±0.08% BSL | 4-20mA | NI Compact DAQ Model #9203 | 3/19/2010 |
| RTU Compressor, Blower, and Total Power | True Power | Dent PowerScout 18™ | ±0.5% kW reading | RS-485 | Serial | 7/24/2013 Serial# PS18909134 |
| Condensate Generation | Weight | Adam Equipment-GBK 16A –Bench Scale | ±0.3 g ±0.006 lb | RS-232 | Serial | |

## REFRIGERANT MEASUREMENTS

Properties of the refrigerant were determined by measuring the temperature and pressure of the refrigerant before and after the compressor, and measuring the temperature after the condenser. The refrigerant properties were recorded for information only; they were not used to calculate system capacity. The RTDs used to measure the refrigerant temperatures were placed in contact with the refrigerant pipes and insulated.

## EVAPORATOR MEASUREMENTS

Dry bulb temperature, wet bulb temperature, and flow rate were controlled to provide return air at 80/67 (DB°F/WB°F) at the flow rate as described in the "Fan Speed Control" section. Return and supply temperatures and dew points were measured using a 4-wire RTD and chilled mirror hygrometer, respectively. Condensate generation was measured and recorded using a high accuracy bench scale.

## CONDENSER AIR MEASUREMENTS

The temperature of the air entering the condenser was measured using four RTDs spaced equally over the surface of the condenser. The temperature distribution of the RTDs was evaluated to ensure uniform temperature distribution of the inlet condenser air. The sensors values were averaged, and maximum and minimum readings were assured to be within ±1°F of the average.

## DIFFERENTIAL PRESSURE AND AIRFLOW MEASUREMENTS

The differential and static pressures for the environmental chambers were recorded using an Energy Conservatory APT-8, a data acquisition device with 8 differential pressure channels. For each chamber, the following values were measured and recorded: the static pressure upstream of the flow nozzle with respect to the laboratory, the differential pressure across the flow nozzle, and the static pressure of the chamber with respect to the laboratory.

Differential pressures for the RTU were measured with an Energy Conservatory DG-500, a data acquisition device with two differential pressure channels. These two channels were used to measure differential pressure across just condenser coil and evaporator fan with evaporator (total external static pressure). A baseline measurement across the condenser coil with no ducting attached was performed for the baseline test unit and with each of the pre-coolers tested. This measurement was matched during testing after the ductwork had been reattached to set the condenser air flow rate.

## CHAMBER CONDITIONS MEASUREMENTS

During all tests the inlet and exit conditions of both chambers were monitored with four GE Optisonde chilled mirror hygrometers. These sensors used an RTD to measure dry bulb temperature and air from a sampling grid to measure the dew point. Wet bulb temperature was then calculated from the dry bulb temperature and dew point. Data was digitally output via serial interface every second.

### POWER MEASUREMENTS

Measurements for the total power, compressor power, and fan power were recorded using a PowerScout 18 with a serial interface and Modbus protocol. It digitally output data every three seconds. Compressor power was measured prior to the variable speed drive so measurements include any drive losses. Baseline measurements were also performed without the variable speed drive installed and the compressor operating normally to provide a measurement of the variable speed drive's efficiency.

### DATA ACQUISITION SYSTEM

All signals were acquired using National Instruments (NI) hardware at 0.3 Hz or greater, averaged every 30 seconds using LabVIEW software, and logged to a text file.

## TOLERANCES

The goal for all tests was to adhere to the relevant tolerances specified in ANSI/AHRI Standard 210/240-2008, ANSI/AHRI Standard 340/360-2007, and ASHRAE 37-2009 [3] [4] [5]. Tolerances for both indoor and outdoor dry bulb and wet bulb tolerances are specified in these standards and were upheld during these tests.

The tolerances are listed in Table 8. There are two types of tolerances; the "range tolerance" and the "mean tolerance." The range tolerance specifies the maximum and minimum limits that the controlled variable was allowed, and the mean tolerance specifies the range that the average value of all recorded test points must fall within. The range and mean tolerance were met for a 30 minute period to allow the test equipment to reach steady state and for the immediately following 30 minute test period.

**TABLE 4: TEST TOLERANCES**

| TEST CONDITION | RANGE TOLERANCE | MEAN TOLERANCE |
|---|---|---|
| Dry Bulb Temp. (indoor and outdoor) | ±2°F | ±0.5°F |
| Wet Bulb Temp. (indoor) | ±1°F | ±0.3°F |
| Condenser Coil Pressure Drop | ±7% of set point | |

## EQUATIONS AND ERROR ANALYSIS

From the direct measurements taken capacity, coefficient of performance (total and sensible), and sensible heat ratio were calculated.

## CAPACITY

The capacity of the test unit in Btu/hr was calculated from the following equation:

**EQUATION 1: CAPACITY**

$$q = \frac{Q_e \times (h_i - h_o) \times 60}{v_{e,n} \times (1 + W_{e,n})}$$

Where:

$Q_e$ is the measured flow rate of the evaporator air in $ft^3$/min as described by ANSI/ASHRAE Standard 41.2-1987 [6], $h_i$ and $h_o$ are the enthalpy of the return and supply air, respectively, in Btu/lb, as measured by the chilled mirror hygrometers, $v_{e,n}$ is the specific volume of dry air at the evaporator side nozzle, measured in $ft^3$/lb, and $W_{e,n}$ is the humidity ratio of the air at the evaporator side nozzle in lbw/lba.

## COEFFICIENT OF PERFORMANCE

The coefficient of performance (COP) of the test unit is a dimensionless number that was determined as:

**EQUATION 2: COEFFICIENT OF PERFORMANCE**

$$COP = \frac{q}{P}$$

Where:

$q$ is the capacity of the test unit as measured in equation 2 and $P$ is the total power consumption of the unit in Btu/hr, including the compressor, variable speed drive, condenser fan, and blower.

## SENSIBLE HEAT RATIO

The sensible heat ratio (SHR) for each test was determined as:

**EQUATION 3: SENSIBLE HEAT RATIO**

$$SHR = c_p \frac{T_o - T_i}{h_o - h_i}$$

Where:

$c_p$ is the specific heat capacity of dry air (assumed to be constant), $T_o$ is the dry bulb supply temperature, $T_i$ is the dry bulb return temperature, $h_o$ is the supply enthalpy, and $h_i$ is the return enthalpy. The sensible heat ratio ranges from zero (all delivered capacity is latent cooling) to unity (all delivered capacity is sensible cooling).

### SENSIBLE COP

The sensible COP is a dimensionless measure of the RTU's efficiency at delivering sensible cooling.

**EQUATION 4: SENSIBLE COP**

$$COP_{sensible} = COP \times SHR$$

## MEASUREMENT UNCERTAINTY

The uncertainty of all calculations were conducted using the sequential perturbation method which is a numerical approach that utilizes a finite difference method to approximate the derivatives to represent the sensitivity of the calculated value to the variables used within the calculation [7]. This method is well accepted and used when the partial differentiation method of the propagation of error is complex, or the amount of variables used is very large. The process used for sequential perturbation involves calculating a result, $R_o$, based on measured values. After $R_o$ has been calculated, an independent variable within the equation for $R_o$ is perturbed by its respective uncertainty, and a new value, $R_i^+$ is calculated. Next, the same independent variable within $R_o$ is decreased by its respective uncertainty, and a new value, $R_i^-$ is calculated. The differences between $R_i^+$ and $R_o$, and $R_i^-$ and $R_o$ are calculated and the absolute values are averaged. The result is defined as $\delta R_i$. This process is repeated for every independent variable within $R_o$, and the final uncertainty is calculated as shown in Equation 4.

**EQUATION 5: UNCERTAINTY USING SEQUENTIAL PERTURBATION**

$$U_R = \pm\left[\sum_{i=1}^{L}\left(\delta {R_i}^2\right)\right]^{1/2}$$

The uncertainty for all tests were calculated using this method and are illustrated using error bars in the results section.

# RESULTS

The results obtained show several key trends and illustrate numerous opportunities to optimize the efficiency, capacity, and sensible heat ratio of traditional RTUs. Results of varying compressor and fan speed are shown for 95°F, 85°F, and 75°F outdoor air temperature (Figure 5-Figure 7). Results for medium fan speed only at all three outdoor air temperatures are shown in Figure 8.

## IMPACT OF FAN SPEED

Adjustment of evaporator fan speed has significant effects on both system COP (Figure 5A). Increasing fan speed requires greater electric power delivered to the evaporator fan motor and thereby increases the total power draw of the RTU. This increased fan power draw negatively affects the COP both by increasing the total energy into the RTU, and by introducing additional fan heat which slightly reduces the total net cooling capacity delivered by the system.

The effect on sensible heat ratio is opposite (Figure 5C). Higher fan speeds increase the rate of heat transfer between the evaporator coil and the airstream. This raises the average temperature of the evaporator coil resulting in less dehumidification. Although there is less dehumidification, the total capacity delivered is nearly the same at all fan speeds (except for a slight differences due to fan heat) and thus the sensible capacity and sensible COP are increased.

## IMPACT OF COMPRESSOR SPEED

Results indicated that a variable speed compressor enables three primary optimization strategies. Reduction of compressor speed from 100% to 80% increased the total COP of the unit for the low, medium-low, and medium fan speed cases (Figure 5A). For many applications this could provide energy savings if the RTU could be run at reduced capacity for longer run times. This is the case for most buildings during off-peak conditions, since cooling equipment is sized for peak temperatures.

The second benefit provided by a variable speed compressor is dynamic capacity modulation (Figure 5D). Although RTU's are typically only configured to operate at one or two preset cooling capacities, these capacities only occasionally match the actual capacity required to meet the building load. Test results show that a variable speed compressor drive can effectively modulate the RTUs delivered capacity from 30% - 116% of its nominal cooling capacity. This would enable an RTU to utilize a compressor speed which best meets the building load and also provides an efficiency or peak demand benefit when compared to traditional one or two stage RTUs.

Finally, test results indicate that incorporation of a variable speed drive enables control of a RTUs sensible heat ratio between approximately 70%-95% while operating at constant temperature conditions (Figure 5, bottom left). Dry climates often do not require significant dehumidification capacity, and thus compressor speed can be adjusted to maximize the sensible COP of an air conditioner. This reduces both the peak demand and total energy usage of the building. Alternatively, when conditions do require dehumidification, compressor speed (and fan speed) can be adjusted.

## Impact of Outdoor Air Temperature

The effect of outdoor air temperature is similar across all compressor and fan speeds (Figure 8). Increased outdoor air temperature consistently results in a reduction of COP and total capacity. Increasing the outdoor air temperature also increases the evaporator temperature, thereby increasing the sensible heat ratio. The compressor speed that maximizes COP appears to remain approximately constant regardless of outdoor air temperature.

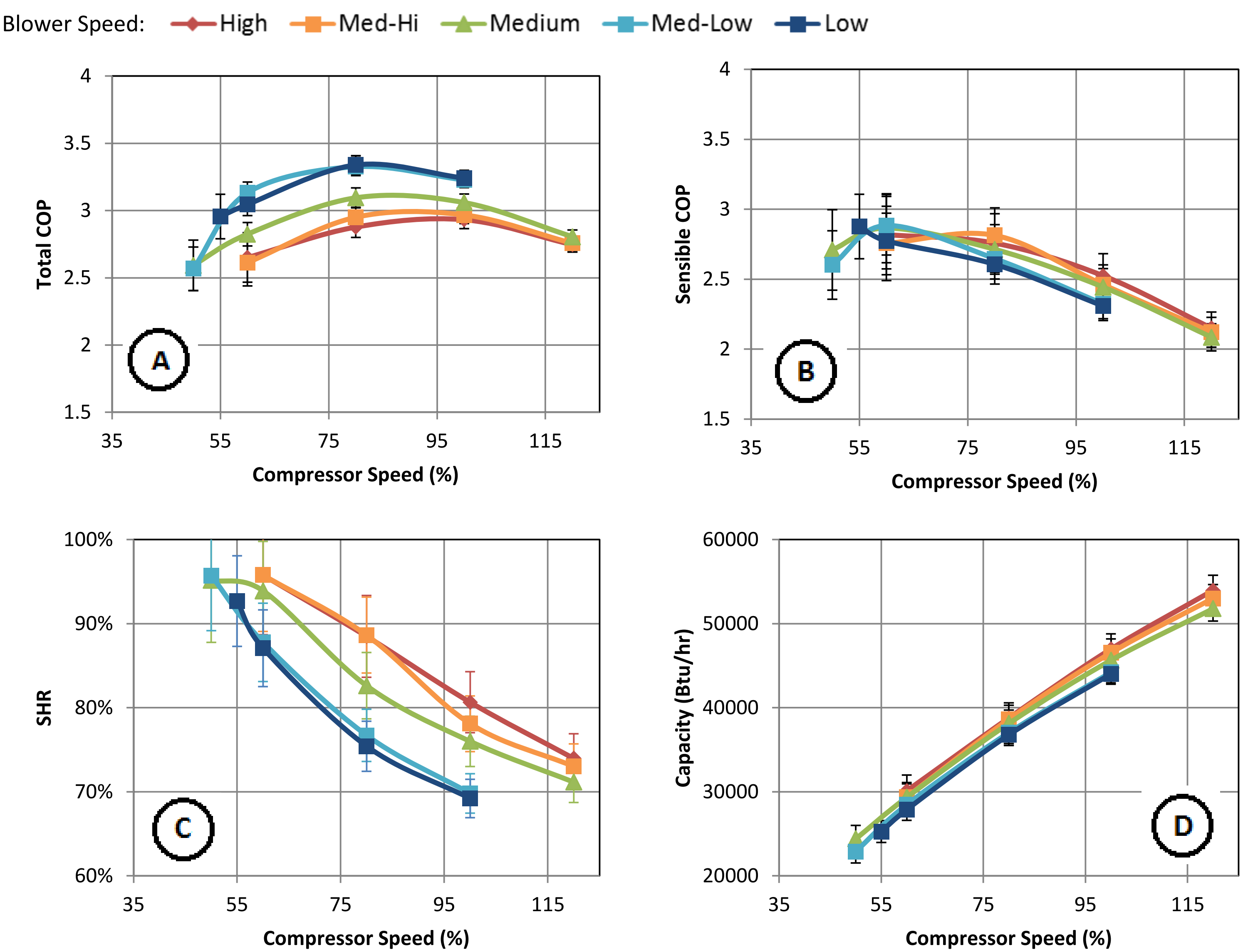


**FIGURE 5: COMPRESSOR SPEED VERSUS VARIOUS PERFORMANCE METRICS AT 95°F HOT SIDE DRY BULB**

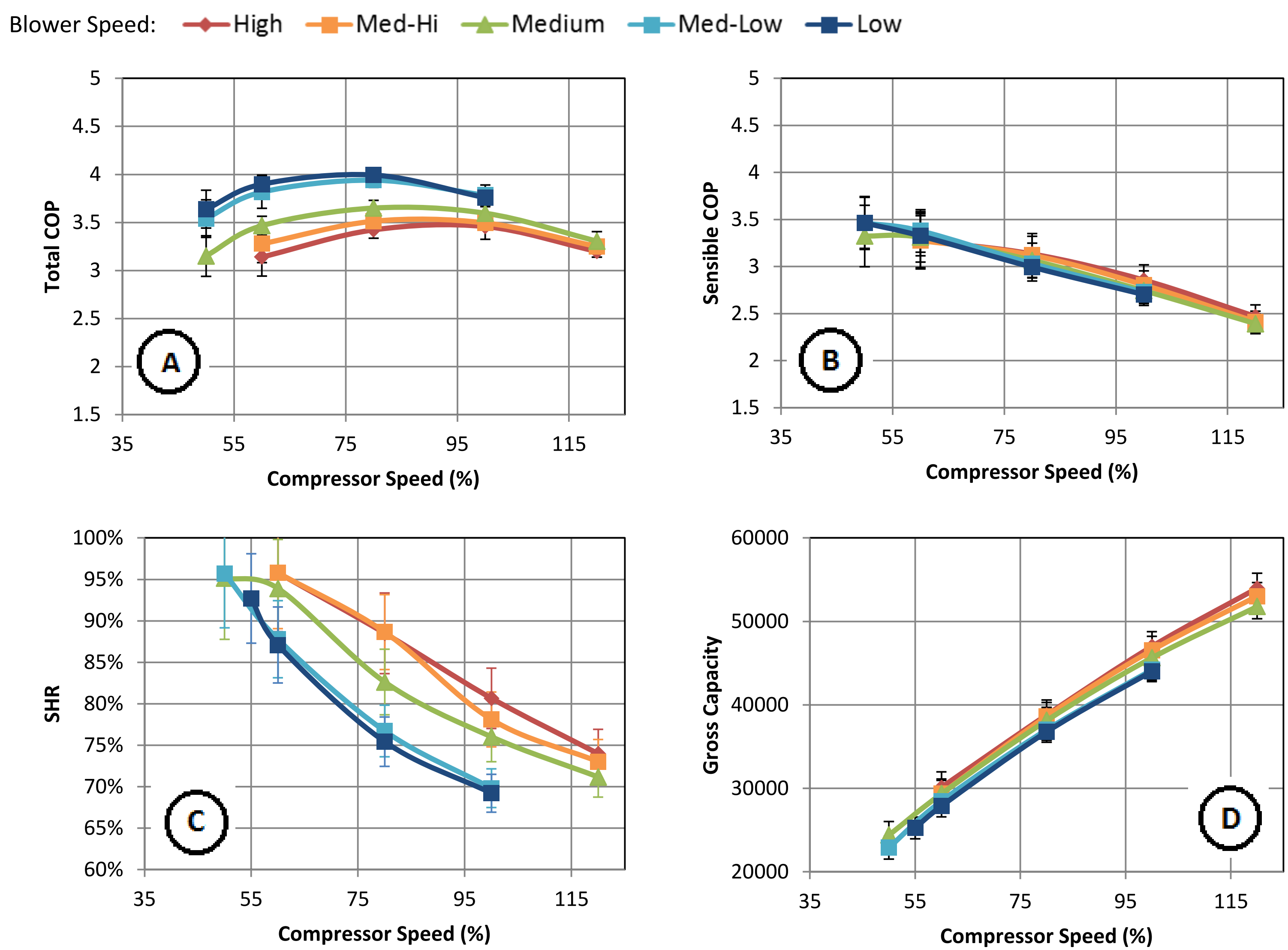


**FIGURE 6: COMPRESSOR SPEED VERSUS VARIOUS PERFORMANCE METRICS AT 85ºF OUTDOOR AIR TEMPERATURE**

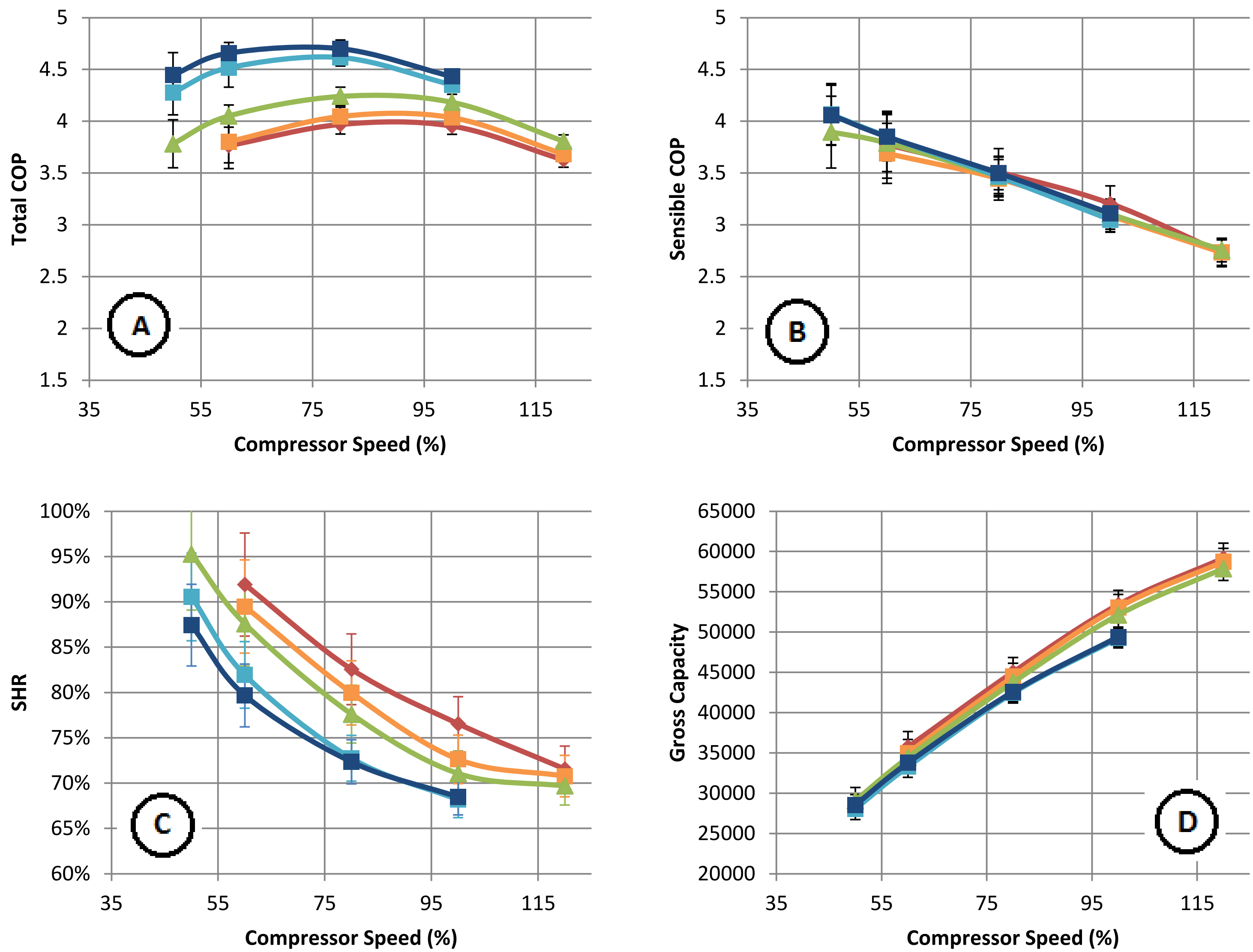


**FIGURE 7: COMPRESSOR SPEED VERSUS VARIOUS PERFORMANCE METRICS AT 75ºF OUTDOOR AIR TEMPERATURE**

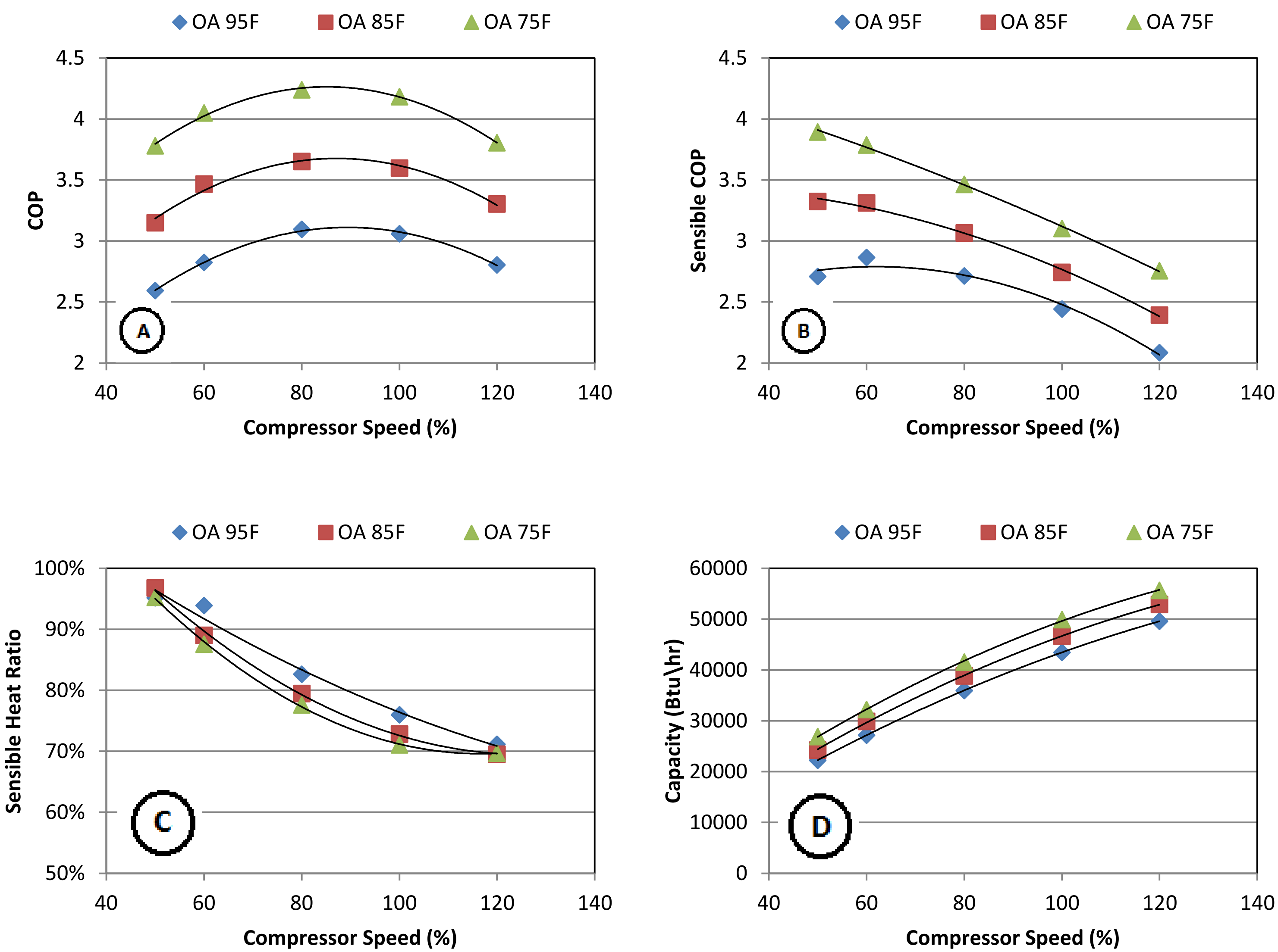


**FIGURE 8: COMPRESSOR SPEED VERSUS VARIOUS PERFORMANCE METRICS AT 75°F, 85°F, AND 95°F HOT SIDE DRY BULB**

## FAN AND COMPRESSOR SPEED CONTROL STRATEGIES

The test results show that matching fan and compressor speeds to the building load can be used maximize total COP. In actual operation, one control strategy would be to select the lowest possible compressor speed to meet the building load, while ensuring that the speed is high enough such that total efficiency is not compromised. Then, the optimal fan speed must be selected to control the sensible heat ratio and building humidity. The challenge is being able to select the optimal compressor and fan speed on any RTU without specifically knowing its operating efficiency, since actual RTU performance will differ between units. There does, however, appear to be certain trends which could be utilized by a control system to provide improved efficiency without requiring a separate analysis of the RTU and its installation.

It was observed throughout testing that a constant refrigerant suction pressure maximized COP for all air flow rates and a fixed outdoor air temperature (Figure 9 - Figure 11). The specific suction pressure which optimizes COP also varies some with outdoor air temperature as shown in Figure 12, which plots the variation of COP as a fucntion of suction pressure for medium fan speed only. By creating a regression of these maxima with respect to outdoor temperature an RTU controller could be programmed to optimize its COP by adjusting compressor speed until the desired suction pressure is reached at the measured outdoor air temperature. This provides a simple, easily implementable means to maximize the efficiency of an RTU across varying airflow rates and outdoor temperatures.

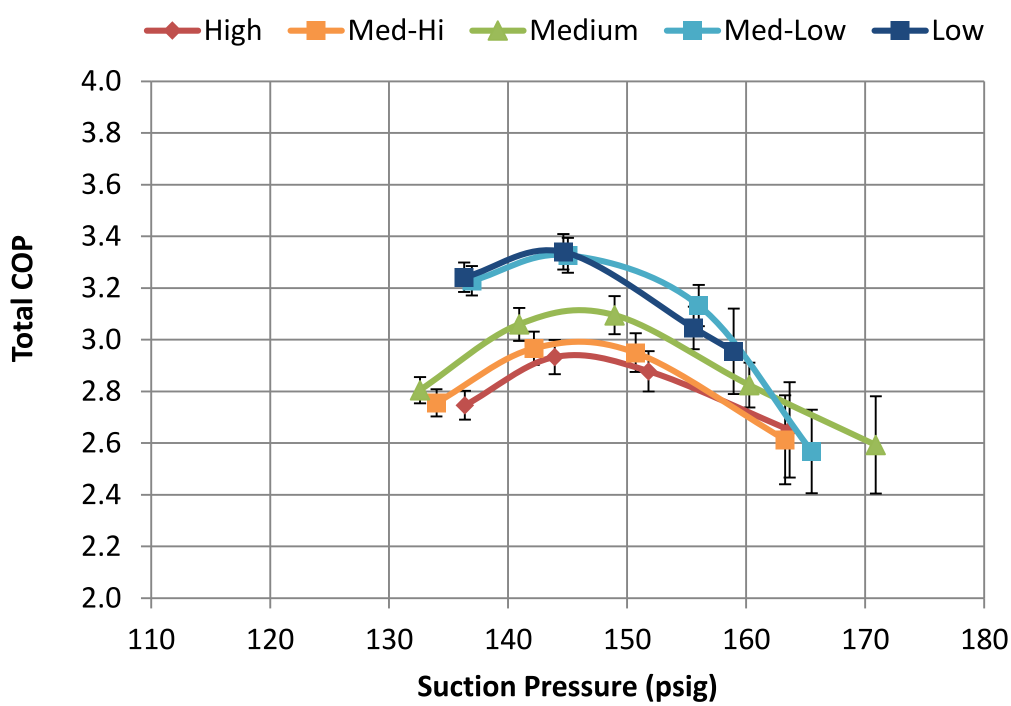


**FIGURE 9: COP VS SUCTION PRESSURE FOR 95ºF OUTDOOR AIR TEMPERATURE**

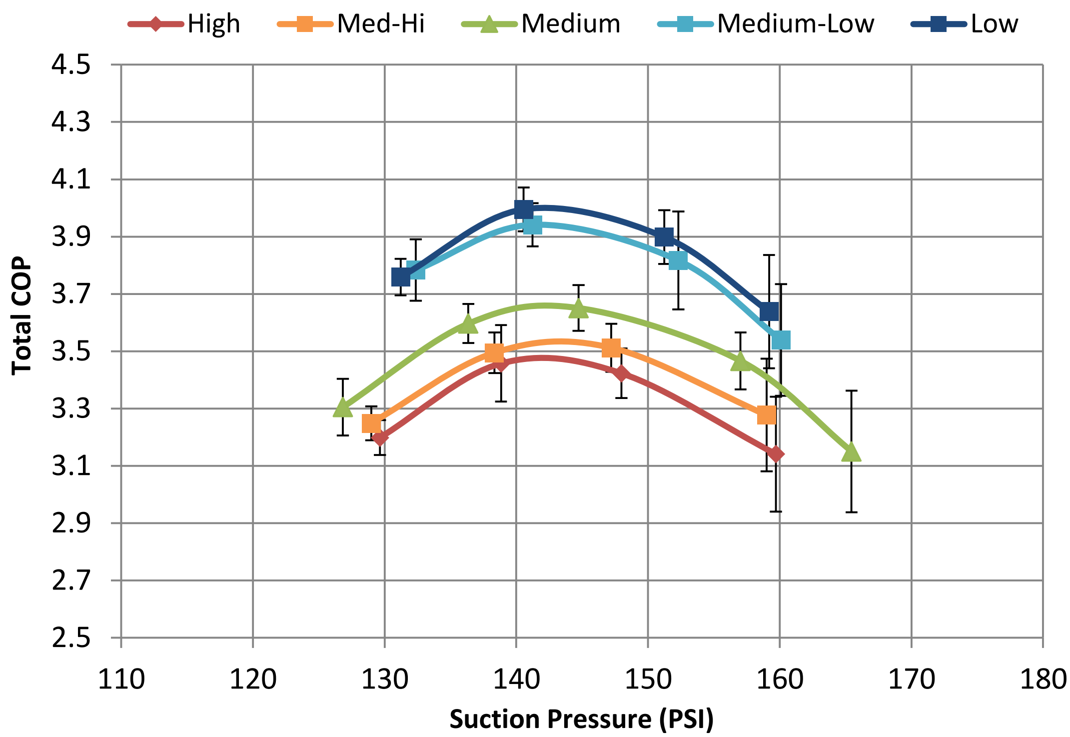


**FIGURE 10: COP VS SUCTION PRESSURE FOR 85°F OUTDOOR AIR TEMPERATURE**

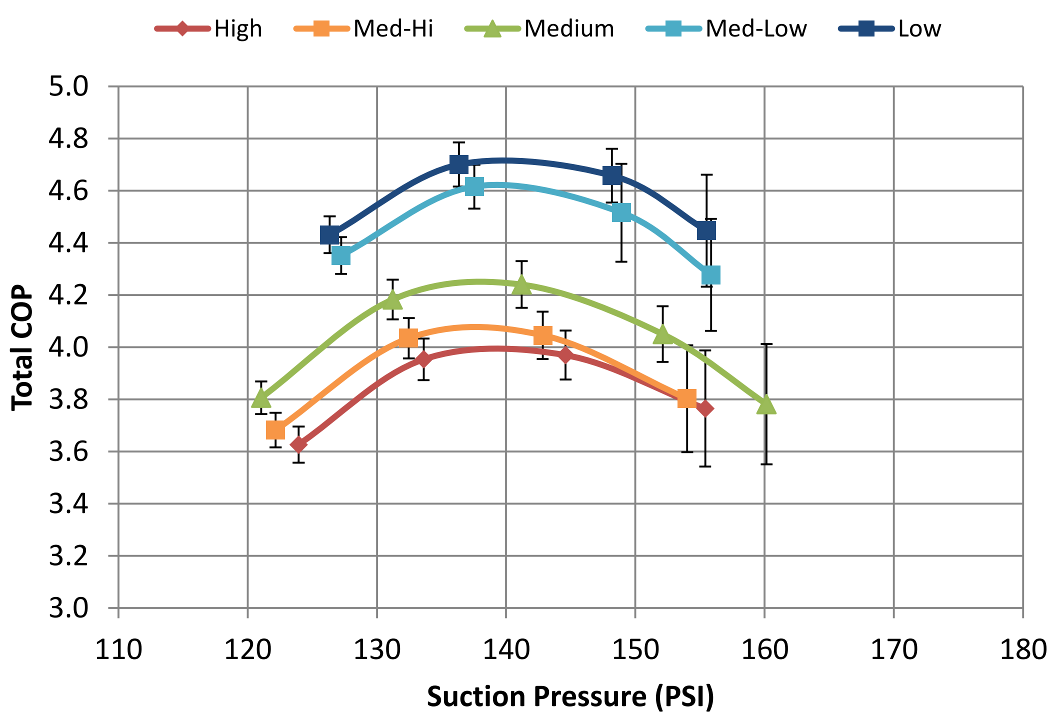


**FIGURE 11: COP VS SUCTION PRESSURE FOR 75°F OUTDOOR AIR TEMPERATURE**

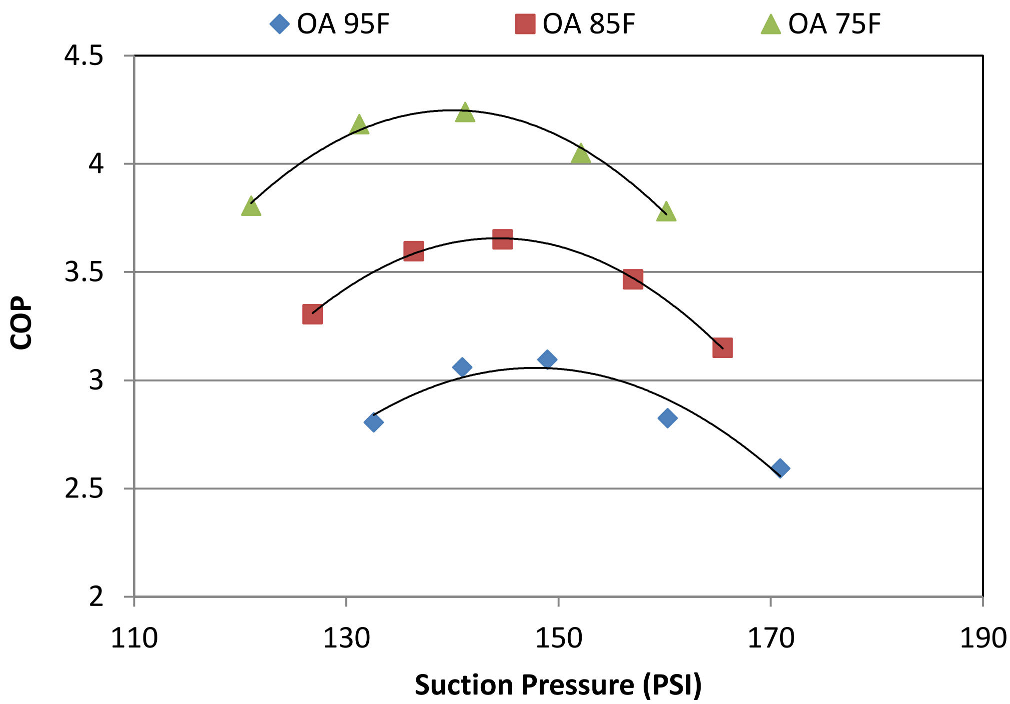


**FIGURE 12: TOTAL COP VS SUCTION PRESSURE**

During installation of a compressor/fan speed controller on any RTU, the ideal suction pressure could be determined by measuring system COP at various compressor speeds and developing a relationship between suction pressure and COP at the current ambient outdoor air temperature. Then, this suction pressure could be programmed as the optimal operating suction pressure, with a correction made for measured outdoor air temperature based on a pre-programmed relationship developed from this research.

In order to implement this strategy, a method is needed to estimate the COP of a RTU in the field. It is important that the exact COP is not required, but rather the relative efficiency as a function of compressor speed. This measurement could be made by using differential enthalpy between the supply and return duct as a proxy for system capacity. Enthalpy can be measured using temperature and humidity sensors, which although not always accurate on an absolute scale, can be calibrated to one another when the system is operating in fan only mode (supply and return enthalpy should be approximately equal), thus providing a precise differential enthalpy measurement.

In order to complete the calculation of COP, a measurement of compressor power is also needed. The cost of current transducers and associated electronics is low enough that direct measurement may be an option. Alternatively, other system parameters can be used as a proxy. Work done by compressing a gas is the change in pressure times the change in volume. The change in pressure of the refrigerant could be measured directly using pressure transducers on the suction and discharge lines. These transducers are low cost and already frequently incorporated into high end packaged units for fault detection purposes. The rate change in volume of the refrigerant is correlated to the compressor speed – which is controlled by the variable speed drive. Thus by using a differential pressure measurement across the compressor multiplied by the compressor's speed, an approximation of the compressor's electrical power draw can be calculated. This method of measuring a relative compressor power was validated with the laboratory experiments and was shown to be excellent linear approximation (Figure 13).

The results of the laboratory test indicate that a control strategy could be developed where an installer could install a VFD and controller on any RTU and perform a field commissioning process with simple temperature and pressure sensors in a short period of time (estimated at under an hour).

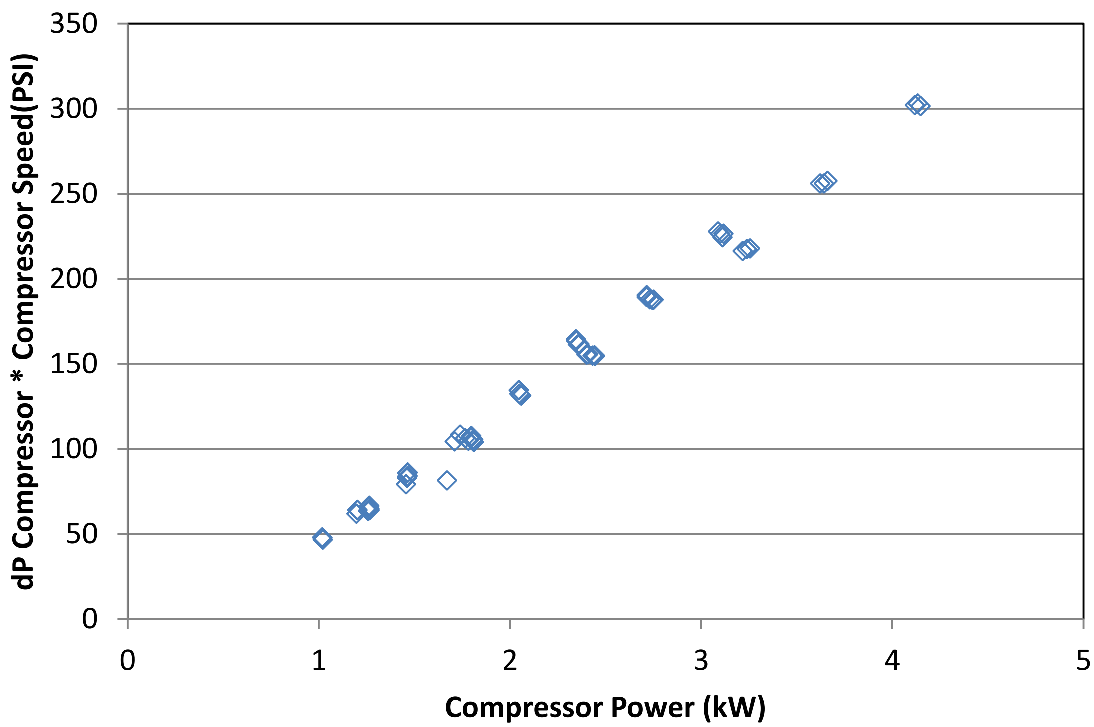


**FIGURE 13: COMPRESSOR POWER VS. PRESSURE DIFFERENCE * COMPRESSOR SPEED**

# CONCLUSIONS AND RECOMMENDATIONS

## ENERGY SAVINGS POTENTIAL

The part-load energy saving potential of this technology is significant. Let's examine one example, a RTU that is primarily designed to operate at an outdoor air temperature of 95°F. In this scenario, the RTU has no advanced controls and the fan and compressor run at fixed speed. At this condition, the RTU tested in the WCEC laboratory consumed 4.16kW of power and had an operating COP of 3.06 (Table 5). If the outdoor air temperature drops to 75°F, the baseline RTU would continue to operate at the same compressor and fan speed. The capacity would increase 15%, which is likely to be completely unnecessary because the thermal load on the building would decrease simultaneously, causing additional cycling of the air conditioner. The power would decrease 16% and the COP would increase 37%.

With the advanced controller, the compressor and fan speed could be reduced at the part load condition (Table 5). The result would be a reduced capacity from the baseline unit operation reflecting the reduced need for cooling the building at lower outdoor air temperatures. In the example shown, the capacity was chosen to match the design capacity delivered to the building at 95°F outdoor air temperature. In practice, even greater capacity reductions may be feasible. The scenario shown reduces the power over the baseline operation at 75°F by 25% and increases the COP by 11%.

**TABLE 5: EXAMPLE OPERATING SCENARIO AND ENERGY SAVING POTENTIAL**

| | DESIGN CONDITION (BASELINE) | PART LOAD CONDITION (BASELINE) | PART LOAD CONDITION (ADVANCED CONTROLS) |
|---|---|---|---|
| Outdoor Air Temperature (°F) | 95 | 75 | 75 |
| Compressor Speed (%) | 100 | 100 | 80 |
| Fan Speed | Medium | Medium | Medium-Low |
| Sensible Heat Ratio | 76% | 71% | 73% |
| System Capacity (Btu/hr) | 43,466 | 49,928 | 41,112 |
| System Power (kW) | 4.16 | 3.50 | 2.61 |
| System COP | 3.06 | 4.18 | 4.62 |

While the described scenario shows that savings only occur at part-load conditions, this technology would be extremely beneficial when combined with an evaporative condenser-air pre-cooler technology. In dry California climates, evaporative condenser-air pre-coolers reduce the effective outdoor temperature seen by the condensing unit. At a peak condition, it is possible to reduce the outdoor air temperature 20°F or more. Using a combination of evaporative condenser-air pre-cooler technologies and fan and compressor speed controls could reduce peak electricity demand by more than 35%.

The biggest market barrier for the technology is that there is no known manufacturer currently selling a controller to retrofit RTUs with the capabilities described here. WCEC is working to engage potential manufacturing partners that could deploy a product with these capabilities, particularly in combination with evaporative pre-cooling.

WCEC recommends further development of this technology, including engaging with potential manufacturing partners. Furthermore, WCEC recommends conducted a laboratory and field test of the technology in combination with an evaporative pre-cooler technology that has already been tested with positive results.

# REFERENCES

[1] D. R. F. a. P. Bailey, "The State of Affairs - Packaged Cooling Equipment in California," ACEE Summer Study on Energy Efficiency in Buildings, 2000.

[2] Itron, "California Commercial End-Use Survey," California Energy Commission, Sacramento, 2006.

[3] ANSI/AHRI, "Standard 210/240 "2008 Standard Rating of Unitary Air-Conditioning and Air-Source Heat Pump Equipment"," 2013.

[4] AHRI, "AHRI 340/360. Performance Rating of Commercial and Industrial Unitary Air-Conditioning and Heat Pump Equipment," AHRI, Arlington, 2007.

[5] ASHRAE, "ASHRAE 37-2009. Methods of Testing for Rating Electrically Driven Unitary Air-Conditioning and Heat Pump Equipment," ASHRAE, 2009.

[6] ASHRAE, "Standard 41.2-1987 - Standard Methods for Laboratory Air Flow Measurement," ASHRAE, Atlanta, 1987.

[7] R. S. Figliola and D. E. Beasley, Theory and Design for Mechanical Measurements, 3rd ed., John Wiley & Sons, Inc, 2000.

[8] J. Sunde, J. Wang and K. Vlasek, "Digi RTU Optimizer Case Study: New Technology to Maximum Energy Efficiency," ACEEE Summer Study on Energy Efficiency in Industry, 2011.

[9] E. Djunaedy, K. Van Den Wymelenberg, B. Acker and H. Thimmana, "Oversizing of HVAC System," University of Idaho, Boise, 2010.