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Energy Research and Development Division

FINAL PROJECT REPORT

Development and Testing of the Next-Generation Residential Space Conditioning System for California

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PREFACE

The California Energy Commission's (CEC) Energy Research and Development Division supports energy research and development programs to spur innovation in energy efficiency, renewable energy and advanced clean generation, energy-related environmental protection, energy transmission and distribution and transportation.

In 2012, the Electric Program Investment Charge (EPIC) was established by the California Public Utilities Commission to fund public investments in research to create and advance new energy solutions, foster regional innovation and bring ideas from the lab to the marketplace. The CEC and the state's three largest investor-owned utilities—Pacific Gas and Electric Company, San Diego Gas & Electric Company and Southern California Edison Company—were selected to administer the EPIC funds and advance novel technologies, tools, and strategies that provide benefits to their electric ratepayers.

The CEC is committed to ensuring public participation in its research and development programs that promote greater reliability, lower costs, and increase safety for the California electric ratepayer and include:

- Providing societal benefits.
- Reducing greenhouse gas emission in the electricity sector at the lowest possible cost.
- Supporting California's loading order to meet energy needs first with energy efficiency and demand response, next with renewable energy (distributed generation and utility scale), and finally with clean, conventional electricity supply.
- Supporting low-emission vehicles and transportation.
- Providing economic development.
- Using ratepayer funds efficiently.

Development and Testing of the Next-Generation Residential Space Conditioning System for California is the final report for Contract Number: EPC-14-021 conducted by the Electric Power Research Institute. The information from this project contributes to the Energy Research and Development Division's EPIC Program.

For more information about the Energy Research and Development Division, please visit the CEC's research website (www.energy.ca.gov/research/) or contact the CEC at ERDD@energy.ca.gov.

ABSTRACT

To optimize conditioning systems for greater energy efficiency, utility integration, and homeowner comfort, the Next-Generation Residential Space Conditioning System for California integrates advanced heating, ventilation, and air conditioning technologies. The researchers evaluated an alternative refrigerant as a possible future enhancement and assessed duct losses for single- versus multi-zone duct configurations with variable capacity equipment.

In this project, three leading United States laboratories provide experimental results to the industry on optimizing the system for efficiency, utility integration, and homeowner comfort. Key findings from the laboratory evaluation included:

- Cooling energy savings ranged between 22 and 32 percent for California climates using variable speed compressors and fans compared to a single speed system that meets California's minimum baseline requirement.
- The tested integrated system satisfies more than 90 percent of annual heating need for most of the state without electrical or natural gas backup.
- Demand response capability with variable capacity equipment enables utilities to reduce peak demand while reducing customer discomfort.
- Zonal control, integrated ventilation, and intelligent heating integrated with the Next-Generation Residential Space Conditioning System offer air delivery versatility that may result in additional energy cost savings and increased homeowner comfort.
- Revising ducting standards and providing more efficient control strategies would improve the integration of heat pumps connected to attic ductwork for hot and dry California climates.
- R-32 (global warming potential 675) as an alternative drop-in refrigerant in a variable capacity heat pump improves system efficiency while reducing refrigerant charge compared to R-410A (global warming potential 2100).

The Next-Generation Residential Space Conditioning Systems was field tested in three homes in investor-owned utility service territories with all systems functioning appropriately. Customer feedback was positive. Users appreciated how quiet the units were and how quickly they cooled or heated the space, as well as their ability to control the temperature in individual spaces. Daikin/Goodman has featured the technologies used in this study in their product lineup.

Keywords: variable-capacity heat pump, variable-capacity compressor, variable-speed blower, demand response, alternative refrigerant, heat recovery ventilator, zonal control, dual fuel, duct delivery effectiveness, coefficient of performance, residential ace conditioning

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EXECUTIVE SUMMARY

Introduction

Cooling and heating of homes to achieve comfortable temperature and humidity levels account for a large portion of California ratepayers' electricity bills, accounting for 48 percent of the residential energy in the United States and 31 percent in California. Every kilowatt-hour consumed by conventional air-conditioning systems requires the production of more energy at the power plant, resulting in more greenhouse gases and straining the capacity of electricity transmission and distribution systems. Many technologies that deliver efficiency exist, but they are not integrated into a single heating, ventilation, and air-conditioning (HVAC) system, nor are they optimized for California climates. This project evaluated a promising variable capacity heat pump with additional heating and cooling technologies that could provide higher efficiency space conditioning for California conditions.

Project Purpose

Project researchers developed, tested, and assessed the Next-Generation Space Conditioning System (Next-Gen RSCS) for heating and cooling California homes in the interest of optimizing conditioning systems for efficiency, utility integration, and homeowner comfort. The Next-Gen RSCS integrates multiple advanced HVAC technologies including: a variable capacity compressor, variable speed blower fan, automated demand response, fault detection and diagnostics, intelligent dual fuel heating (gas/electric), integrated ventilation, and zonal control. In addition to these features, the researchers evaluated an alternative refrigerant, R-32, as a possible future product enhancement. The project team also assessed air duct losses for single and multi-zone duct configurations with variable capacity equipment.

Project Process

Researchers completed this project in three phases. The first two phases were laboratory testing of selected Next-Gen RSCS features, and the third phase was field testing commercially available units in representative California climates. The laboratory evaluation was conducted in three independent facilities, Electric Power Research Institute, Pacific Gas and Electric Company (PG&E), and Western Cooling Efficiency Center, where each facility tested a 2-ton ducted split-system air-source heat pump unit provided by Daikin/Goodman. The laboratories' experimental setups evaluated the cooling and heating performance of the units under real-world operating conditions, based on standard conditions and California climate zones.

The research team field-tested Next-Gen RSCS in three homes in California investor-owned utility service territories: West Sacramento, PG&E; Chino Hills, Southern California Edison; and San Diego, San Diego Gas & Electric. A 4-ton system with new ducts was installed in each of the homes with instrumentation to collect data on the heating and cooling performance, as well as the functionality of the system with zonal control, demand response, and dual fuel heating capability.

Project Results

Key findings from the laboratory evaluation of the Next-Gen RSCS included:

- Variable capacity heat pump performed at higher system efficiency than their rated seasonal energy efficiency ratio levels when operating at lower speed settings.
- Cooling energy savings ranged between 22 and 32 percent for California climates using variable speed compressors and blowers/fans compared to a single speed system that meets California's minimum baseline requirement of 14 seasonal energy efficiency ratio.
- The system satisfied more than 90 percent of annual heating load for most of California without electrical or natural gas backup.
- Demand response capability with variable capacity equipment allowed utilities to reduce peak demand while maintaining customer comfort.
- Using R-32 (global warming potential 675) as an alternative drop-in refrigerant, or as a direct replacement refrigerant without further modifying the HVAC equipment, in a variable capacity heat pump improved system efficiency while reducing refrigerant charge compared to R-410A (global warming potential 2100). In laboratory testing, R-32 demonstrated an ability to be an effective, low global warming potential replacement for R-410A as a drop-in refrigerant in the variable capacity heat pump from an equipment performance and functionality perspective. R-32 reduced system charge by 29 percent compared to R-410A.
- Using R-32 in HVAC equipment offered a potential mechanism for peak power reduction in the warmest California climates, while reducing refrigerant charge.
- Zonal control, integrated ventilation, and intelligent heating integrated with the Next-Gen RSCS offered the customer energy cost savings and system versatility.
- While dual fuel functionality (electric heat pump with gas furnace backup) added versatility to Next-Gen RSCS, utility rates are primary drivers for encouraging heat pump use. Heat pumps are well suited for California climate zones since they can meet almost all loads in the heating season without backup.
- With integrated ventilation, the Next-Gen RSCS provided 1 to 4 percent additional cooling energy savings and 1 percent additional heating capacity for a variable capacity heat pump with heat recovery ventilator.
- Revising ducting standards and providing more efficient control strategies would improve the integration of heat pumps connected to attic ductwork for hot and dry California climates. Zoning should be required for variable capacity heat pumps with ducts in unconditioned space. Zoning can significantly reduce duct losses at low speed.

The field evaluation provided the following results:

- The variable capacity, dual fuel (intelligent heating), automatic demand response, and zonal elements of the Next-Gen RSCS all functioned appropriately in the field, although zonal control added to the complexity of the system operation for setup with more than two zones. Variable capacity heat pump (seasonal energy efficiency ratio 21/HSPF 9.6) field data shows efficiency improvements over baseline (single speed, seasonal energy efficiency ratio 14/HSPF 8.2)

- The project field testing demonstrated a clear need and energy saving benefit for zoning when installing a variable capacity heat pump with ductwork located in unconditioned space, which can significantly affect the variable capacity heat pump system performance. A minimum of two zones is recommended. Duct location and insulation will affect choice of control algorithms.
- Customer feedback on the Next-Gen RSCS was positive. Users appreciated how quiet the units were and how quickly they cooled or heated the space, as well as their ability to control the temperature in individual spaces (zonal control).

Energy cost savings of 50 percent are technologically achievable when implementing the Next-Gen RSCS compared to commonly used residential HVAC systems.

The researchers prepared and executed a technology transfer plan that identified important product features, possible barriers to their adoption, and actions to overcome these barriers. These actions included presentations, publications and interactions with key market participants including presentations at the American Society of Heating, Refrigerating, and Air-Conditioning Engineers, American Commission for an Energy-Efficient Economy, Electric Power Research Institute, and California Energy Commission; meetings and symposia; involvement with American Society of Heating, Refrigerating, and Air-Conditioning Engineers; and Title 24 standards-setting activities.

A production readiness plan was prepared for the Next-Gen RSCS including a discussion of product features and production requirements. The proprietary nature of manufacturers' information made it difficult to obtain production investment information, but a large share of the investment is likely to be associated with the variable speed features including the inverters and associated controls. Having a major manufacturer as a project partner greatly improves the likelihood of successful commercialization of the Next-Gen RSCS. Daikin/Goodman has featured the technologies used in this study in their product lineup, including variable capacity compressors, variable speed indoor blowers, demand response, fault detection and diagnostics, and integrated ventilation.

Recommendations for future development of the Next-Gen RSCS technology include: examining the cost effectiveness of each feature for the Next-Gen RSCS in California; configuring the Next-Gen RSCS into three different models (base, intermediate, and premium); refining zonal control with variable speed operation; limiting heat transfer to ducts in unconditioned spaces; developing an intelligent heating controller that permits users to select their preferred heating mode of operation; and refining the sensitivity of the fault detection and diagnostics of the Next-Gen RSCS.

Field measurements continued after the project term ended and a report will be developed by the project team. This report is available upon request by contacting ERDD@energy.ca.gov.

Benefits to California

Due to the higher efficiency of the Next-Gen RSCS, ratepayers of California could enjoy electricity savings of 50 percent, greater electricity reliability, lower cooling and heating costs, increased comfort, and increased safety over the toxicity and flammability of other refrigerants by integrating advanced energy technologies and intelligent controls into a single HVAC system. Inverter technology enables compressors and fans to run at capacities and speeds that match the load of a residence—instead of inefficiently switching between on and off states like traditional HVAC systems. Intelligent control of different "zones" within a residence prevents the conditioning of unoccupied spaces and the convenience of setting different temperature setpoints for different zones. The ability to intelligently switch between electricity and natural gas to heat a home provides customers with added choice to manage their energy and utility bills. HVAC systems that respond to demand-response signals from electric utilities enable the utilities to conserve capacity during peak demands for electricity, creating a more reliable grid. The variable capacity heat pump feature of the Next-Gen RSCS can maintain the customer thermal comfort during a demand response event. And advanced fault detection and diagnostics enhance the reliability of the Next-Gen RSCS. The lower carbon footprint of alternative refrigerants increases the health of all California residents through greater energy efficiency.

When compared to commonly used residential HVAC systems, Next-Gen RSCS could provide savings of 475 gigawatt-hours per year, or $83 million annually, in California. This assumes 20 percent of the market installs the new system and an average residential electric rate of $.176 per kilowatt-hour. The Next-Gen RSCS could also contribute to saving 1 to 1.5 gigawatts in peak demand and reducing carbon dioxide emissions by more than 157 million kilograms (157,000 metric tons) per year in California.

Introducing the next-generation residential space-conditioning system into the residential HVAC market will constitute an evolution in the way that residential buildings are conditioned to ensure the comfort of occupants. Integrated fresh-air ventilation will improve the quality of the air in conditioned areas, improving comfort. Residents will also benefit from participating in utility demand-response programs, which lower utility bills and increase grid stability by curtailing the consumption of energy when high demand for electricity would otherwise tax transmission and distribution systems.

CHAPTER 1: Project Overview and General Project Tasks (Task 1)

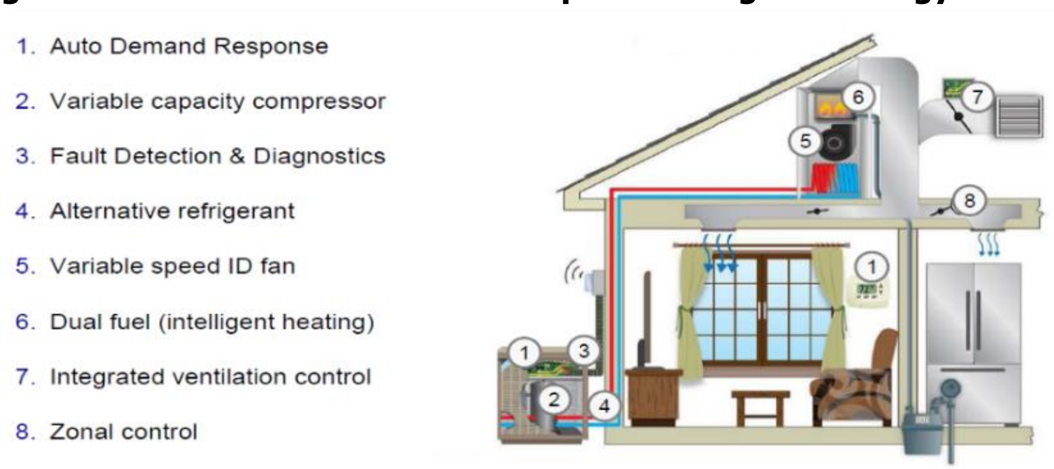
The Issue

Cooling and heating of buildings to achieve comfortable temperature and humidity levels account for a large portion of California ratepayers' electricity bills. Every kilowatt-hour consumed by conventional air-conditioning systems requires the production of even more energy at the power plant, resulting in the emission of greenhouse gases and putting a strain on electricity transmission and distribution systems. The building sector of California awaits energy-efficiency technologies that will decrease the operating cost of space-conditioning equipment, increase the comfort of consumers, and enhance the reliability of electricity in California, thereby achieving the goal of "big bold energy efficiency strategies" identified in California's Long Term Energy Efficiency Strategic Plan of September 2008. Many of the technologies that deliver efficiency exist, but they are not integrated into a single heating, ventilation, and air-conditioning (HVAC) system, nor are they optimized for California climates.

Project Goals and Objectives

The purpose of EPIC Project EPC-14-021 was to develop, test, and model a prototype Next-Generation Residential Space-Conditioning System (Next-Gen RSCS) that integrated several advanced technologies and is optimized for the California climate (Figure 1).

Figure 1: Next-Generation Heat Pump Enhancing Technology Features



Source: Electric Power Research Institute

Specific project goals were to:

- Develop a next-generation residential space-conditioning system that integrates the best energy-efficient technologies for California consumers.
- Test the system in independent laboratories in multiple phases to continue to develop the system and integrate more energy-saving technologies.

- Model various configurations of the next-generation system to optimize its performance for California consumers.
- Test the system at three field locations in real-world operating environments and compare its performance to traditional HVAC systems.
- Through multiple technology-transfer efforts, impart the findings of the project to stakeholders and the public.

Specific project objectives were to :

- Obtain performance data from laboratory testing of two prototype next-generation residential space-conditioning systems.
- Obtain performance data from field testing three prototype next-generation residential space-conditioning systems.
- Create an energy model of the next-generation residential space-conditioning systems.
- Achieve a minimum of 50 percent energy savings for residential HVAC.

Background

The Electric Program Investment Charge (EPIC) was created by the California Public Utilities Commission (CPUC) in December 2011 to support investments in clean energy technologies that provide benefits to the electricity ratepayers of Pacific Gas and Electric Company (PG&E), San Diego Gas & Electric Company (SDG&E), and Southern California Edison Company (SCE). The EPIC program funds clean energy research, demonstration and deployment projects that support California's energy policy goals and promote greater electricity reliability, lower costs, and increased safety. The California Energy Commission (CEC), through EPIC, is filling critical funding gaps within the energy innovation pipeline to advance technologies, tools, and strategies of near zero-net-energy residential homes and commercial buildings, high-efficient businesses, low-carbon localized generation, sustainable bioenergy systems, electrification of the transportation system, and a resilient grid supported by a highly flexible and robust distribution and transmission infrastructure. These smarter, safer energy advancements provide ratepayers with better electricity services, reduce air pollution, foster economic development, and help achieve the state's policy goals at the lowest possible cost. The project documented in this report was conducted to advance the development of advanced residential heating and cooling technologies in California.

Cooling and heating of buildings to achieve comfortable temperature and humidity levels account for 47 percent of the energy use in the United States and 31 percent in California (U.S. Energy Information Administration, 2009). Improving the efficiency of HVAC systems is therefore a primary strategy for reducing the overall energy consumption in California and reducing the greenhouse gasses emitted by the generation of electricity. Many technologies that deliver efficiency exist individually, including automatic demand response, variable-capacity compressors, use of alternative refrigerants, variable-speed fans, and dual fuel technology (intelligent heating) to name a few. Past research efforts on these technologies for improving residential space-conditioning performance have focused on the incremental improvements of each individual technology rather than the combined performance that make up the entire residential HVAC system (for example, the cooling equipment or the duct system).

Project Overview

Improving energy efficiency is a primary strategy for reducing energy consumption and reducing the carbon dioxide (CO_2) emitted by the generation of electricity. The portfolio of advanced efficiency solutions for the next-generation HVAC system includes multiple efficiency solutions, such as variable-capacity compressors and variable-speed fans using state-of-the-art inverter technology; integrated ventilation to harness fresh air for "free cooling"; intelligent dual-fuel technology to decrease energy cost and empower consumers to choose between electricity and natural gas; zonal control to prevent conditioning of unoccupied rooms; demand-response interactivity to grid flexibility and reliability; advanced fault detection and diagnosis to ensure proper installation, operation, and maintenance; and alternative refrigerants for improved operation and large reductions in global warming.

This project examined the benefit of integrating such advanced energy-efficient and intelligent technologies into a single optimized residential HVAC system, including the effects of the conditioning equipment and the ductwork. Energy efficient technologies currently available are optimized for outdoor conditions that represent a national "average" climate condition but do not address the specific concerns for climate zones that have higher-than-average temperatures or low humidity or both. The only current mandatory test for high temperature is a maximum-operating-conditions test at 115°F (46°C), for which performance information is not published by manufacturers. There is a need for affordable next-generation space-conditioning systems that integrate the individual energy-efficient technologies and components available worldwide or in the research and development (R&D) phase to strive for optimal performance in a variety of climates. This will result in overall decreases in operating cost and increases in energy efficiency, comfort, and reliability for consumers living in different climates of the United States.

Additionally, the project used modeling to verify the efficacy of prototype designs and customize the system for California climates. The successful integration of the proposed energy solutions—and the market clout of Daikin/Goodman coupled with other manufacturers entering the market—will help influence changes to California's Title 24 building codes and to national standards. Such codes and standards will eventually engender the manufacture and marketing of similar products, with economy of scale reducing the cost of next-generation space-conditioning systems.

Project Scope

The project was split into three phases to achieve its objectives. Phase 1 and Phase 2 entailed laboratory evaluation of the system's eight technology features at three independent facilities: Electric Power Research Institute's (EPRI's) Thermal Testing Laboratory in Knoxville, Tennessee; PG&E's Applied Technology Services in San Ramon, California; and University of California (UC) Davis' Western Cooling Efficiency Center (WCEC) in Davis, California.

Phase 3 involved field evaluation of the Next-Generation Residential Space Conditioning System (Next-Gen RSCS) at three occupied residential buildings. Each unit was installed in one of the three California investor-owned utility (IOU) service territories: PG&E, SCE, SDG&E. The results from phases 1 and 2 informed the optimization configurations for Phase 3 field-evaluation prototypes.

The technology evaluation was distributed among the three labs based on their facilities and expertise, outlined in Table 1. Although the testing was conducted independently, there was some redundancy to provide "round-robin" testing.

Table 1: Technology Attributes Tested by Three Labs, Phases 1 and 2

Phase	Technology	EPRI	PG&E	WCEC
1	Variable-Capacity Compressor	✓	✓	✓
1	Variable-Speed Blower	✓	✓	✓
1	Integrated Ventilation	✓		
1	Demand Response	✓	✓	
1	Dual Fuel (intelligent heating)		✓	
1	Duct-loss assessment for single-zone			✓
2	Alternative Refrigerants	✓	✓	
2	Fault Detection & Diagnostics		✓	
2	Zonal Control	✓		
2	Duct-loss assessment for multi-zone			✓

Source: EPRI

Next-Generation Residential Space Conditioning System Features

A review of the literature pertaining to the potential benefits and liabilities of the Next-Gen RSCS features was performed as part of the technology transfer planning effort and is summarized in Table 2.

The Product Features section of Chapter 6 provides more details regarding the work reported on each system feature.

The testing and analysis in the current program and the literature review provided information on the potential operating savings resulting from implementation of the Next-Gen RSCS. In summary, the energy savings for each feature determined by this project, and in a cursory review of the literature review, were as follows:

- Variable speed — Current study: 22 percent to 32 percent energy savings; other studies: 35 percent to 40 percent energy savings.
- Alternative refrigerant — Current study: R-32 — 1.2 percent to 3.0 percent improvement in cooling efficiency; other studies: 1 percent to 9 percent energy savings.
- Dual fuel — Current Study: for average electric and gas prices, highest energy cost savings: 22 percent for climate zone 3 (Oakland); other studies: no quantitative results found.
- Fault detection and diagnosis — Current study: no quantitative study results; other studies: up to 55 percent energy savings.

- Reduced duct losses, improved delivery effectiveness — Current study delivery effectiveness improved by 0 percent (at full speed) and up to 50 percent (at part load); other studies: 10 percent to 25 percent energy savings due to reduced duct losses.
- Integrated ventilation control — Current study: 1.3 percent to 3.8 percent energy savings; other studies: 30 percent or more energy efficiency improvement.
- Zonal control — Current study: 10 percent load reduction would result in 12.8 percent power reduction; other studies: 15 percent to 20 percent.
- Auto demand response — Current study: 50 percent power reduction resulted in only a 38.2 percent capacity reduction, 70 percent power reduction resulted in only a 61.8 percent capacity reduction; savings depend on specific utility and customer conditions.

Table 2: Potential Benefits and Liabilities of Next Gen System Features

Feature	Benefits	Liabilities
Variable Speed	Greater comfort, lower energy costs, improved noise, smoother operation, possible reliability improvements	Higher equipment cost and complexity, harmonics/noise
Alternative Refrigerants	Lower global warming potential (GWP), performance improvement and lower energy cost	Possible higher equipment and servicing cost, flammability issues, possible higher insurance premiums
Dual Fuel	Lowest possible heating costs based on fuel selection	Higher equipment costs
Fault Detection and Diagnostics	Improved maintenance and reliability, lower energy costs	Higher cost of control systems, possibility of unnecessary service calls
Reduced Duct Losses, Improved Delivery Effectiveness	Lower energy costs	Cost of sealing and insulating ducts
Integrated Ventilation Control	Lower energy costs	Higher equipment costs
Zonal Control	Better controllability, greater comfort	Higher equipment costs
Auto Demand Response	Lower energy bills while using the system during peak summer/winter periods, greater comfort	Higher equipment costs

Source: EPRI

Anticipated Benefits for California

Due to the high efficiency of the next-generation HVAC system, ratepayers of California will enjoy greater electricity reliability, lower cooling and heating costs, increased comfort, and increased safety from the integration of advanced energy technologies and intelligent controls into a single HVAC system. Inverter technology enables compressors and fans to run at capacities and speeds that match the load of a residence, instead of inefficiently switching between ON and OFF states like traditional HVAC systems. Intelligent control of different "zones" within a residence prevents the conditioning of unoccupied spaces. The ability to intelligently switch between electricity and natural gas to heat a home saves energy and lowers utility bills. HVAC systems that respond to demand-response signals from electric utilities enable the utilities to conserve capacity during peak demands for electricity, creating a more reliable grid. And advanced fault detection and diagnostics enhance the reliability of the HVAC system. The lower carbon footprint and toxicity of alternative refrigerants such as R-32 increases the safety of all California residents.

Introduction of the next-generation residential space-conditioning system into the residential HVAC market will constitute an evolution in the way that residential buildings are conditioned to ensure the comfort of occupants. Integrated fresh-air ventilation will improve the quality of the air in conditioned areas, improving comfort. Residents will also benefit from participating in utility demand-response programs, which lowers utility bills and increases grid stability by curtailing the consumption of energy when high demand for electricity would otherwise tax transmission and distribution systems.

Chapter 5 provides qualitative and quantitative information on the potential energy, demand and environmental benefits resulting from this project. Energy cost savings of 50 percent are achievable when implementing the Next-Gen RSCS compared to commonly used residential HVAC systems. This could result in savings of 475 gigawatt-hours per year (GWh/yr) or $83 million annually in California, assuming market penetration of 20 percent and an average residential electric rate of $.176 per kilowatt-hour (kWh). The Next-Gen RSCS could also contribute to saving 1 to 1.5 GW in peak demand and reducing CO_2 emissions by more than 157 million kilograms (kg) per year in California.

Project Specifics

The project term was from July 2015 to June 2019.

Lead Organization and Partners

Electric Power Research Institute was responsible for project management, laboratory testing and technology evaluation of the features identified in Table 1 and described in chapters 2 and 3, field testing of three Next Gen units as described in Chapter 4, evaluation of project results as described in Chapter 5, technology transfer efforts described in Chapter 6, and preparation of a production readiness plan as described in Chapter 7.

Western Cooling Efficiency Center (WCEC) at the University of California, Davis, was responsible for the laboratory testing and associated technology evaluations of the features identified in Table 1; Pacific Gas and Electric's Applied Technology Services Department was responsible for the laboratory testing and associated technology evaluations of the features identified in Table 1; Daikin/Goodman (technology provider), supplied laboratory and field test units incorporating the features depicted in Figure 1 and assisted in understanding the operation of these units.

Technical Advisory Committee

The technical advisory committee (TAC) members represented utilities and research/industrial professionals with relevant experience to advise regarding the project's scope and direction. Two TAC meetings were held, in May and August of 2016, via webinar. The TAC members included: Jerine Ahmed, SCE; Mark Fernandes, Los Angeles Department of Water and Power; Marshall Hunt, PG&E; Peter Klint, Eversource; Jim Parks, Sacramento Municipal Utility District; Christian Weber, PG&E; Kate Zeng, SDG&E from utility companies and research and industry professionals Mangesh Basarkar, Consultant; Pat Phelan, Professor, Arizona State University; Reinhard Rademacher, Professor and Director, University of Maryland; Tom Smolarek, President, Cypress Ltd; and John Suzukida, Lanex Consulting.

CHAPTER 2: Phase 1 Laboratory Evaluation (Task 2)

Because next-generation space conditioning systems' combined technologies have the potential to greatly increase energy efficiency, integrate utilities to reduce peak demand, and increase homeowner comfort, this project was designed to develop a next-generation residential space-conditioning system that integrates advanced energy-efficient and intelligent technologies into a single optimized residential HVAC system. The project also tested and refined system and model various configurations of the next-generation system to optimize its performance. Phase 1 of this project consisted of evaluating six advanced technologies: a variable-capacity compressor, a variable-capacity blower, integrated ventilation, automated demand response, intelligent dual fuel, and duct loss for single-zone configurations. Evaluations were conducted in three independent laboratories. The following sections describe the features tested, the laboratory configurations, and the results.

Technology Features Evaluation

Each of the three labs received the same model 2-ton ducted split-system air-source heat pump unit from Daikin/Goodman. The system tested in Phase 1 of this project included off-the-shelf components of current production models of a variable speed system. The components included:

- Outdoor Unit: Daikin
  - 2-ton rated cooling capacity heat pump with inverter drive compressor
  - R-410a refrigerant
  - Rated seasonal energy efficiency ratio (SEER) 19-21 / heating season performance factor (HSPF) 9.6-10.0
- Furnace: Daikin
  - 80,000 British thermal unit (Btu)/hr modulating burner, ½-horsepower variable speed blower
  - Rated annual fuel utilization efficiency (AFUE) 97
- Indoor Coil: Daikin
- Thermostat: Honeywell ComfortNet

A manufacturer's certified technician installed the test unit in the laboratory to ensure proper installation.

For the Phase 1 evaluation, the units contained R-410A as the refrigerant. All labs evaluated the variable-capacity features of the heat pump system (variable-capacity compressor and variable-speed blower). EPRI's lab then conducted a deeper assessment of integrated ventilation and demand response capability of the system while PG&E tested the demand response and dual-fuel (intelligent heating) features. WCEC assessed the duct-losses associated with a variable-capacity heat pump system integrated in a residential building both experimentally and with a mathematical model. During Phase 1, WCEC tested and quantified the duct losses for a single-zone configuration (Chapter 2). During Phase 2, WCEC expanded its experimental and modeling evaluation for a multi-zone configuration (Chapter 3). The following describes the key features and testing strategies for variable-capacity systems.

Variable-Capacity Space Conditioning

Variable capacity space conditioning refers to the ability of a system to modulate the cooling or heating output in response to thermal loads and occupants of the conditioned space. Variable capacity is accomplished through the implementation of a variable speed compressor, a variable speed blower and fan, and a control system within an HVAC system. Variable speed compressors and variable speed fan systems are two next-generation technologies that have the potential to save a substantial amount of thermal energy (Tassou et al., 1983). Both these methods are becoming standard on new commercial/industrial HVAC systems. There is currently a push to install variable speed components in residential split systems with the idea that many residential systems are oversized for their load, so adding variable speed drives should reduce the energy consumed by the fan and compressor—and reduce system short cycling. Variable capacity offers multiple potential benefits to the customer and utility over baseline HVAC equipment including improved seasonal energy efficiency and improved demand response capability. During Phase 1, multiple aspects of the variable capacity system were evaluated within the laboratory setups including variable capacity compressor, variable speed blower, unit control system, and demand response capability.

Ventilation Requirements and Heat Recovery Ventilator

To improve overall building performance and energy use, residential building codes and energy efficient constructions are implementing improvements in the building envelope. Improving the insulation and tightness of the building envelope can significantly reduce the natural ventilation and exchange of fresh air within the occupied space. ASHRAE 62.2 "Ventilation and Acceptable Indoor Air Quality in Residential Buildings" outlines proper fresh air requirements for residential applications. Multiple forms of mechanical ventilation have been developed to provide the occupied space with necessary levels of fresh air. During Phase 1, a heat recovery ventilator (HRV) was evaluated as part of the Next-Generation Space Conditioning system. An HRV provides fresh air ventilation efficiently by exchanging heat between a fresh air and exhaust air stream. In warm outdoor conditions, the fresh air temperature from an HRV is reduced before being introduced to the conditioned space or HVAC system. Meanwhile, for cold outdoor conditions, the fresh air temperature from an HRV is increased. The improvement in fresh air temperature allows for a reduced ventilation load on the overall space conditioning system, while maintaining appropriate fresh air ventilation requirements.

Variable Capacity System Connected to Ductwork System

While variable speed equipment has been shown to achieve very high efficiencies, it is not clear how these systems perform when connected to a standard duct system in the attic, which can often reach more than 120 degrees Fahrenheit (49 degrees Celsius) in the summer. With the large loads imposed on ductwork running through an attic, combined with a longer residence time of the conditioned air in the ducts due to part-load cooling, the efficiency of the system will be reduced. Current performance standards for residential systems do not address the relationship between the air conditioning system and the ductwork. With the introduction of variable speed systems, questions regarding their efficiency at part-load will need to be resolved to properly determine their efficacy and to create alternative control strategies for optimizing performance.

Zonal control (that is, maintaining individual temperature set-points in different zones of a building) is another strategy that has been emphasized as a means to optimize total HVAC energy cost (Gupta et al., 2016). While this strategy is commonplace in most commercial buildings in the United States (F. Jazizadeh, et al., 2014), the majority of single-family houses in the United States have HVAC systems typically controlled by a single, centrally located thermostat (Alles, 2006). There has been some interest in developing airflow control strategies for zone-based temperature control in multi-zone residential systems (Foster et al., 1993), but their effect on the efficiency of the air-conditioning system is not well understood, especially when including the duct-losses. Each zone may have supply ducts with multiple branches, thus the available surface area for heat gain from the attic will be different for each zone. This implies that the same flow rate through each supply trunk will render different delivery efficiencies through the ducts. Phase 1 of this project assessed the duct losses associated with the space-conditioning system operating in a multi-zone ducted application, with a focus on the effects of variable capacity and airflow on duct thermal losses and delivered capacity in multi-zone operation.

Steady-State and Load-Based Laboratory Strategies

Steady-state laboratory testing has been used to evaluate the performance and efficiency of HVAC equipment for numerous decades. Steady-state testing, which is also referred to as psychrometric testing, involves maintaining the indoor and outdoor conditions surrounding the HVAC equipment at steady-state and operating the HVAC equipment at a fixed level of output. Evaluating a variable capacity system under steady-state operation consists of fixing the output of the unit at certain levels, such as minimum and maximum output. Steady-state testing characterizes the performance of the system under specific outputs and air conditions. Steady-state testing eliminates a potentially crucial aspect of the operation of variable capacity equipment, namely the control system.

A load-based laboratory evaluation consists of imposing a thermal load on the indoor zone and allowing the unit controller to determine the appropriate output of the system. In load-based evaluations, the indoor zone is not maintained at steady-state conditions by the test setup, but rather the unit itself is responsible for maintaining appropriate conditions based on the unit setpoints and imposed load. Load-based evaluations are similar in nature to calorimetric testing, which is used to evaluate certain types of HVAC equipment. As opposed to steady-state testing, which fixes the level of operation, load-based testing allows the unit controls to modulate and adjust the unit output in response to the imposed thermal load on the indoor zone. Load-based evaluations examine the control system of the variable capacity unit and provide a more complete understanding of overall real-world operating performance under certain scenarios of operation.

Comparison of Performance Data Across Three Laboratories

As a benchmark, the three laboratories tested the cooling performance at the same set of conditions. The set consisted of five test points with similar outdoor, indoor, and external static pressure conditions. The results of the similar set of cooling performance data are provided in Table 3. In the table for each test condition, each of the involved labs is highlighted with a different color. Although efforts were made for similar equipment operation across the three labs for this set of tests, indoor unit airflow differed across the three labs due to general equipment setup and the approach used to evaluate the variable capacity space conditioning equipment. Considering the slight discrepancies across the three laboratories, the results of similar test conditions are highly comparable across the three laboratory setups.

Table 3: Comparison of Steady-State Performance Across Laboratories

Test Mode	OA Tdb (°F)	RA Tdb (°F)	RA Twb (°F)	Airflow (CFM)	Ext. Res. (IW)*	Capacity (Tons)	Power (kW) Indoor Unit	Power (kW) Outdoor Unit	Power (kW) Total	Coefficient of Performance Equip.
Cooling High	**95**	**80**	**67**	758	0.45	1.98	0.18	1.74	1.92	3.63
				579	0.45	1.88	0.14	1.70	1.84	3.59
				822	0.45	1.97	0.24	1.74	1.98	3.51
Cooling High	**95**	**75**	**62**	761	0.41	1.74	0.17	1.60	1.77	3.46
				574	0.45	1.61	0.14	1.69	1.83	3.10
				823	0.45	1.81	0.24	1.73	1.97	3.24
Cooling Int.	**95**	**75**	**62**	534	0.22	0.89	0.07	0.65	0.72	4.33
				472	0.30	0.92	0.08	0.67	0.76	4.27
				527	0.21	0.93	0.08	0.67	0.75	4.36
Cooling Low	**95**	**75**	**62**	562	0.28	0.41	0.09	0.29	0.38	3.82
				417	0.23	0.47	0.06	0.33	0.39	4.17
				528	0.23	0.52	0.09	0.32	0.41	4.52
Cooling High	**115**	**75**	**62**	753	0.46	1.38	0.18	2.09	2.27	2.13
				577	0.45	1.42	0.14	2.11	2.25	2.23
				822	0.45	1.56	0.24	2.16	2.40	2.28

***: Inches of Water**

Source: EPRI

Electric Power Research Institute Thermal Lab Activities and Results

The EPRI Thermal Testing Laboratory is located at EPRI's Knoxville, Tennessee facilities. The following paragraphs describe the experimental configuration, measurements and results of variable capacity compressor and blower, integrated ventilation and demand response testing conducted at this laboratory in Phase 1.

Laboratory Setup

Steady-State Performance Setup

A steady-state laboratory evaluation was conducted on a 20 SEER, 10 HSPF variable capacity split heat pump in heating and cooling operation over an outdoor temperature range of 15°F (–9°C) to 62°F (17°C) for heating operation and 65°F (–15°C) to 115°F (46°C) for cooling operation. The steady-state testing was conducted using an assumed indoor external static pressure curve and appropriate indoor conditions. The laboratory setup consisted of a two-zone, thermal chamber with one zone serving as a simulated indoor zone and the other serving as an outdoor zone. Figure 2 provides the detailed experimental schematic of the laboratory test setup.

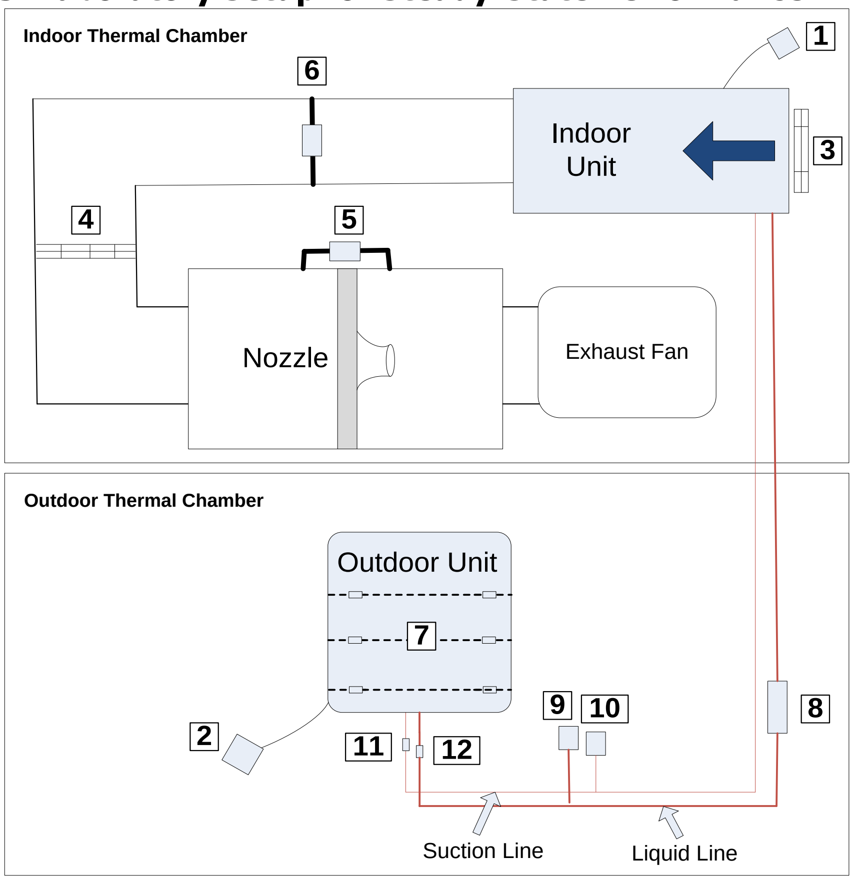
Figure 2: EPRI Thermal Chambers Used in Experimental Setup

Source: EPRI

Air-side and refrigerant-side measurements were conducted, which allowed for air-side and refrigerant-side capacity calculations. An exhaust fan on the indoor setup allowed the external static pressure on the indoor unit to be adjusted in accordance with the assumed external static pressure curve. Laboratory measurements included return air conditions, supply air conditions, outdoor air conditions, indoor airflow, external static pressure, indoor and outdoor unit power, refrigerant suction, discharge temperature and pressure, and refrigerant mass flow. The return and supply air measurements were conducted with a 9-point matrix setup, while the outdoor temperature measurement was conducted with a 10-point grid surrounding the face of the outdoor unit.

Table 6 provides a detailed list of the measurements and instrumentation used in the experimental setup. Numerical indicators shown in Figure 3 correspond with the actual location of measurement in the test setup and the location values shown in Table 5. The table provides the instrumentation and associated nominal accuracy used for each measurement.

Figure 3: Laboratory Setup for Steady-State Performance Evaluation



Source: EPRI

Based on the experimental measurements recorded within the laboratory setup, air-side capacity was determined for each steady-state test. Air-side capacity and corresponding air-side unit efficiency were used as the primary means of evaluating the unit performance. Refrigerant-side capacity was used to verify the air-side capacity calculation. The following equations provide the calculation of air-side capacity and efficiency, presented as energy efficiency ratio:

$$Capacity\ (Air)\ \left(\frac{Btu}{h}\right)$$

$$=\ Air\ Mass\ Flow\ \left(\frac{lbm}{h}\right) x\ (Return\ Air\ Enthaply - Supply\ Air\ Enthalpy)\ (\frac{Btu}{lbm})$$

$$Energy\ Efficiency\ Ratio\ \left(\frac{Btu}{Wh}\right) = \frac{Capacity\ (\frac{Btu}{h})}{Total\ Power\ Consumption\ (W)}$$

Table 4: Instrumentation Description of Experimental Setup

Location	Measurement(s)	Instrumentation (Accuracy)
1	Indoor Unit Power	Shark Meter 200T (±0.2%)
2	Outdoor Unit Power	Shark Meter 200T (±0.2%)
3	Return Air Temperature	9 Point Omega T-Type Thermocouple Grid (±0.5°C)
	Return Air Humidity	GE Chilled Mirror Hygrometer (±0.2°C dew point)
4	Supply Air Temperature	9 Point Omega T-Type Thermocouple Grid (±0.5°C)
	Supply Air Humidity	GE Chilled Mirror Hygrometer (±0.2°C dew point)
5	Pressure Drop Across Nozzle	Dwyer Series 2000 (±2%)
6	Indoor External Static Pressure	Dwyer Series 2000 (±2%)
7	Outdoor Air Temperature	10 Point Omega T-Type Thermocouple Grid (±0.5°C)
8	Refrigerant Mass Flow	MicroMotion Coriolis (±0.1% Liquid; ±0.35% Gas)
9	Liquid Line Pressure	Setra 207 Pressure Transducer (±0.13%)
10	Suction Line Pressure	Setra 207 Pressure Transducer (±0.13%)
11	Suction Line Temperature	Omega T-Type Thermocouple (±0.5°C)
12	Liquid Line Temperature	Omega T-Type Thermocouple (±0.5°C)

Source: EPRI

Table 5 provides a matrix of the cooling and heating test conditions imposed on the variable capacity heat pump.

Table 5: Test Matrix for Steady-State Performance Evaluation

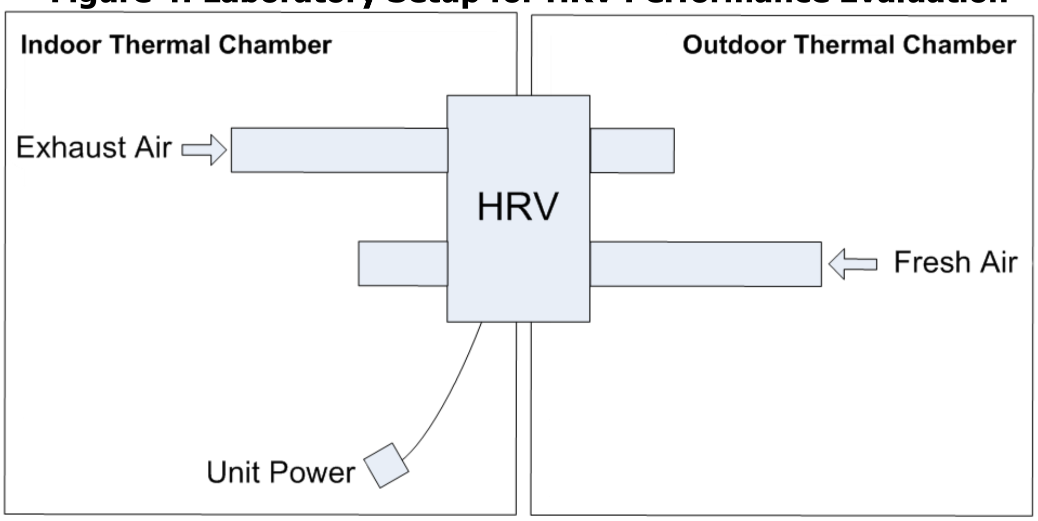
Mode of Operation	Cooling Operation	Heating Operation
Indoor Conditions (dry bulb/wet bulb) (˚F)	75/63	70/57
Outdoor Temperature (˚F)	65, 75, 85, 95, 105, 115	62, 47, 35, 25, 15
System Operating Level	Maximum Output, Intermediate Output, Minimum Output	Maximum Output, Intermediate Output, Minimum Output

Source: EPRI

For both cooling and heating operation, the variable capacity system was operated at three fixed levels of output: maximum, intermediate, and minimum. The maximum and minimum conditions correspond with the maximum and minimum operating boundaries of the system, while the intermediate level falls between these two limits. The indoor condition for cooling operation was a dry-bulb temperature of 75°F (24°C) and a wet-bulb of 63°F (17°C), while the heating operation tests were conducted at a dry-bulb temperature of 70°F (21°C) and a wet-bulb of 57°F (14°C). These indoor conditions were assumed to be realistic operating conditions of the unit in a field or real-world environment.

In conjunction with the steady-state laboratory test setup, the heat recovery ventilator was evaluated over a range of indoor and outdoor air conditions (Figure 4). The HRV was designed to be operated under a fixed airflow, which could be adjusted during equipment setup. In the laboratory evaluation, the HRV was evaluated under the maximum airflow setting. The HRV used the same indoor and outdoor chamber setup as the steady-state setup, as well as comparable instrumentation for exhaust air temperature, fresh air temperature, supply air temperature, and unit power consumption. The measurements of the HRV setup allowed for an evaluation of the performance and HRV effectiveness.

Figure 4: Laboratory Setup for HRV Performance Evaluation

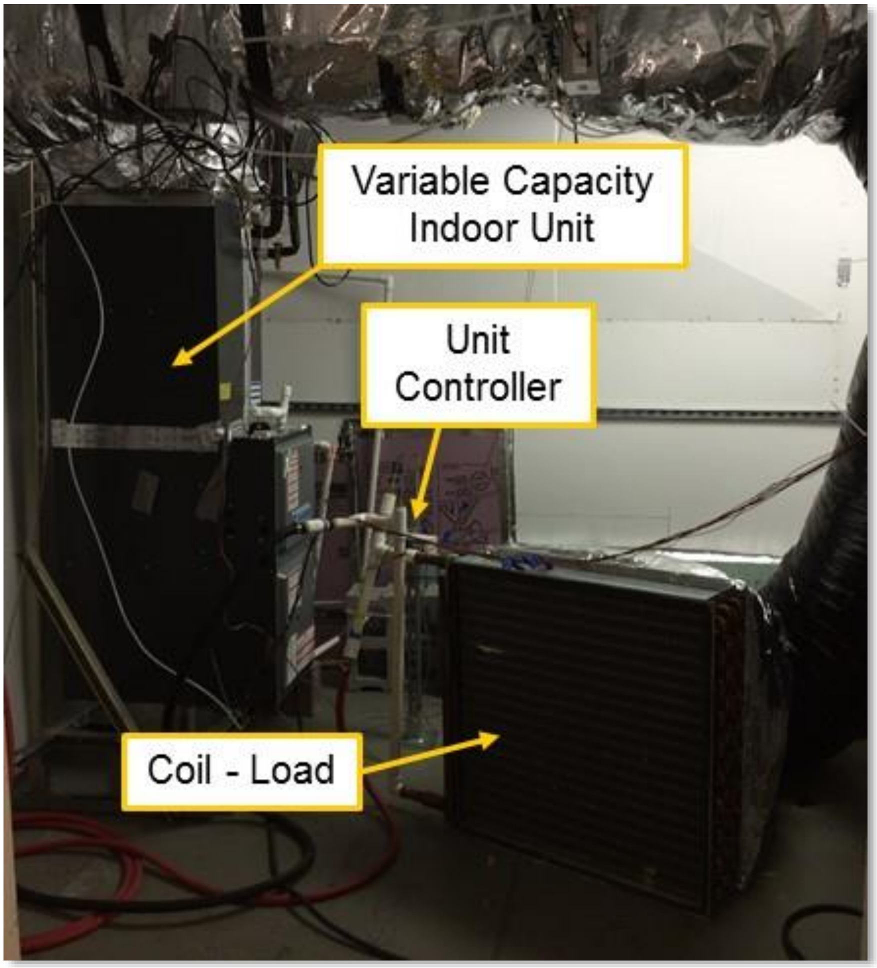


Source: EPRI

Load-Based Performance Setup

The load-based laboratory setup used to evaluate the variable capacity heat pump used a similar setup to the steady-state laboratory evaluation. The previously described indoor and outdoor thermal chambers, experimental measurements, and instrumentation were used in the load-based setup. Along with this equipment, the load-based setup implemented a hydronic coil in series with the indoor unit airflow, which served as a load mechanism to the indoor zone. In the load-based setup, the variable capacity heat pump's standard unit controller was installed within the indoor zone. The standard unit controller operated the variable capacity heat pump as designed by the manufacturer, based on the imposed load and unit settings. e 5 provides an image of the indoor zone for the load-base evaluation.

Figure 5: Layout of Load-Based Evaluation in Indoor Chamber

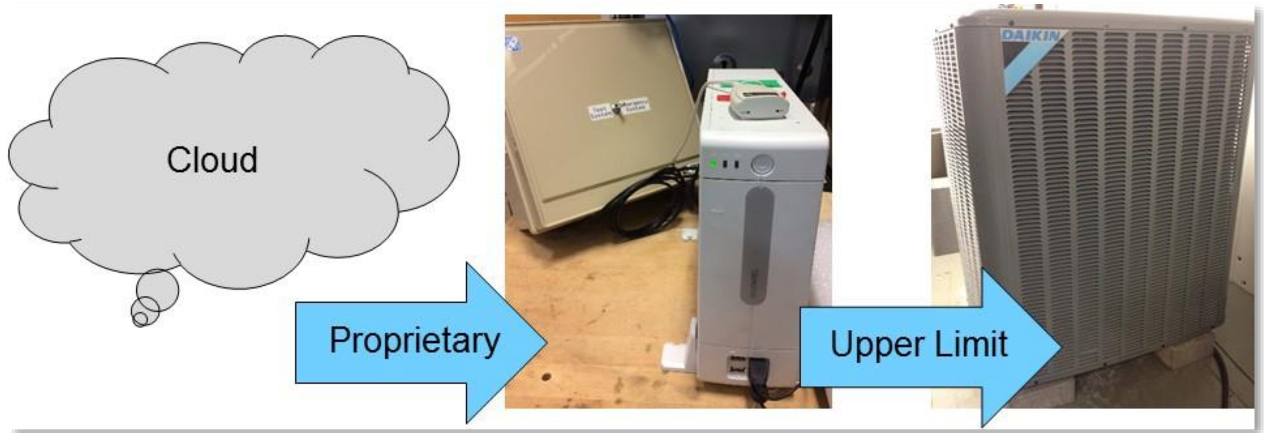


Source: EPRI

Demand Response Setup

To evaluate the demand response (DR) capabilities of the variable capacity heat pump system, computer hardware, which received the DR signal and controlled the DR operation of the system, was implemented within the experimental setup. Along with the DR hardware, the previously discussed indoor and outdoor thermal chambers and air-side instrumentation provided an environment for evaluating the performance of the system under simulated demand response events. The demand response setup used an open automated demand response (OpenADR) infrastructure. A cloud-based service issued a DR event start time, time duration, and payload value to the test unit's DR computer hardware (Figure 6). After receiving the DR signal, the computer hardware adjusted the unit's operation accordingly based on a predetermined upper limit. The upper limit refers to the load capabilities of the equipment and does not directly refer to the operation level of the system. The upper limit in the DR setup was adjustable between 0 percent and 100 percent. An upper limit of 0 percent would not allow the unit to operate during the DR event, while a setting of 100 percent would allow for full operation of the variable capacity system. If the predetermined upper limit was set at 30 percent, the unit would not be allowed to operate above 30 percent during the active DR event.

Figure 6: Laboratory Setup for Demand Response Evaluation

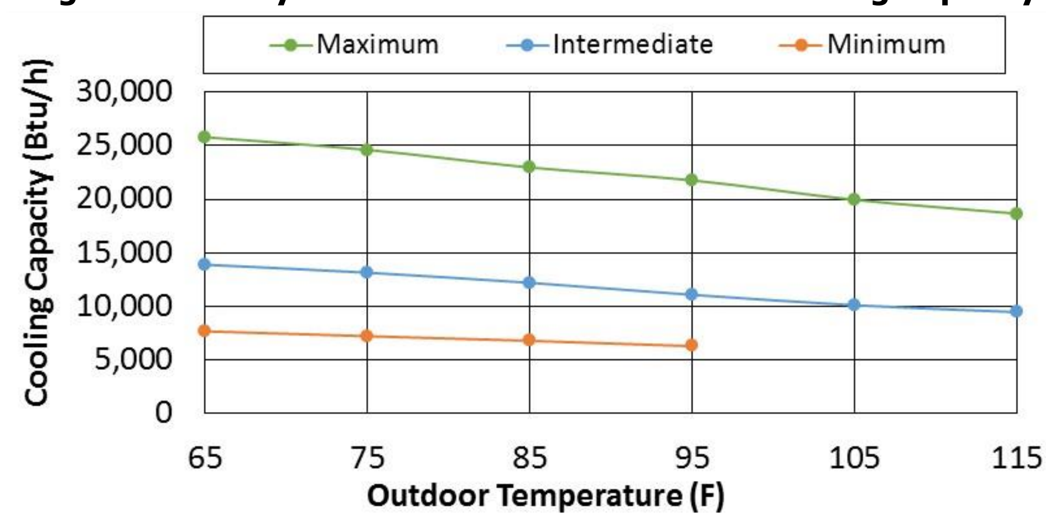


Source: EPRI

Steady-State Results

Figure 7 provides the steady-state cooling capacity results of the variable capacity heat pump over the laboratory test plan conditions. The results are presented as a function of outdoor temperature and operation level. The minimum and intermediate operating levels were approximately 30 percent and 50 percent of the maximum operating level, respectively. In a real-world or field operation, the variable capacity heat pump should be able to operate at capacities between the maximum and minimum output curves.

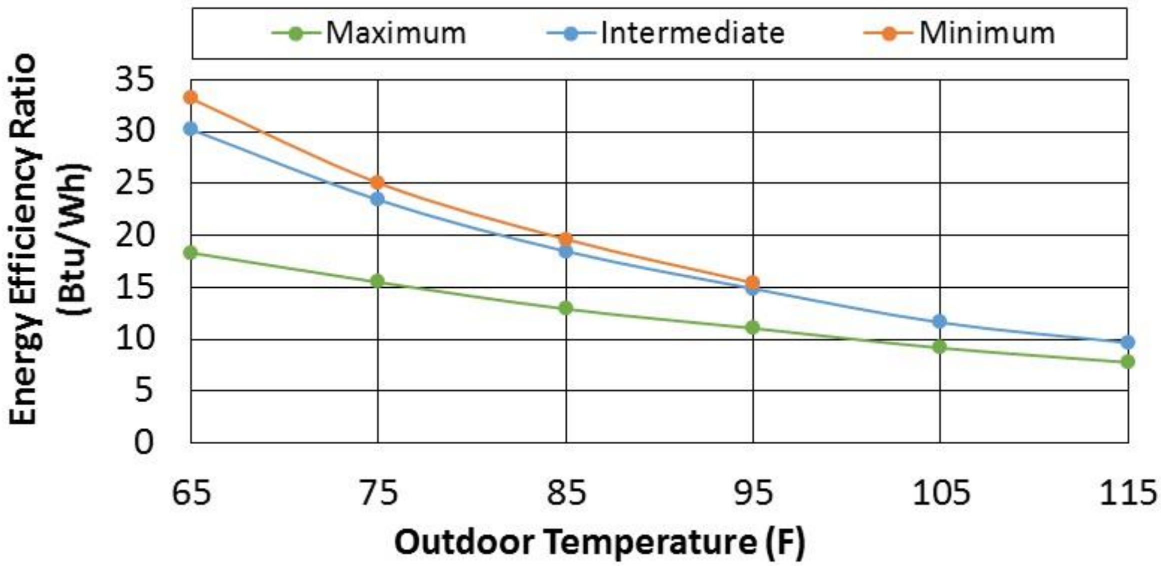
Figure 7: Steady State Performance Results – Cooling Capacity



Source: EPRI

Figure 8 provides the corresponding cooling efficiency data to the cooling capacity data presented in Figure 7. Efficiency is presented as the energy efficiency ratio (EER) as a function of outdoor temperature and operation level. The intermediate and minimum operation efficiency curves are largely comparable for a given outdoor test condition. Multiple trends are observed in examining the cooling efficiency as a function of outdoor temperature and operation level. As a function of outdoor temperature, the efficiency curve for a given level of operation increases with decreasing outdoor temperature. As a function of operation level, the cooling efficiency of the variable capacity unit increases with decreasing level of output. The highest recorded efficiency for a given outdoor condition is when the unit is operating at minimum operation, followed by intermediate operation. The efficiency increase from maximum to intermediate or minimum operation is less substantial at higher outdoor temperatures.

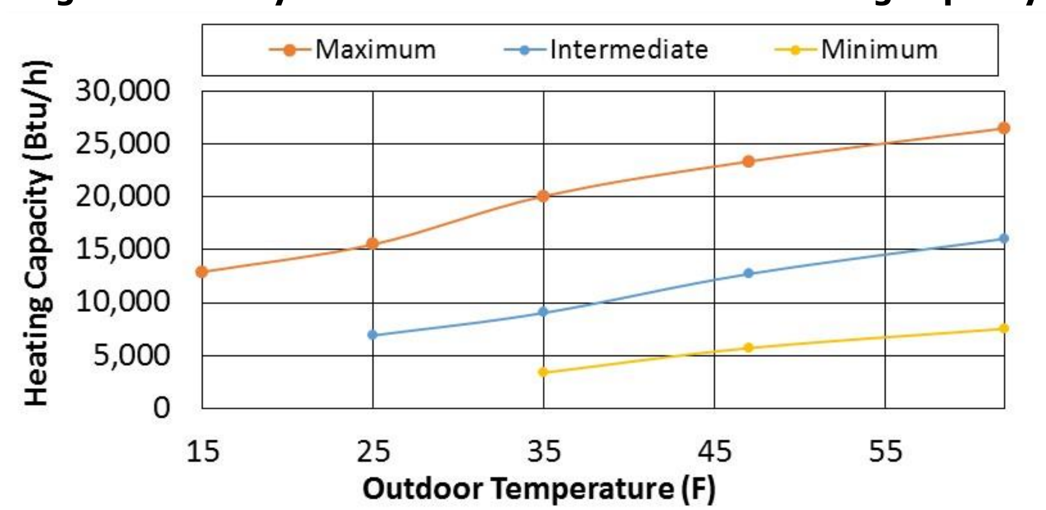
Figure 8: Steady State Performance Results – Cooling Efficiency



Source: EPRI

Figure 9 provides the steady-state heating capacity results as a function of outdoor temperature and operation level. The intermediate and minimum capacity curves are approximately 50 percent and 30 percent of the maximum heating capacity, respectively. In a real-world or field operation, the heating operation of the evaluated variable capacity heat pump would be capable of modulating between the maximum and minimum operation curves.

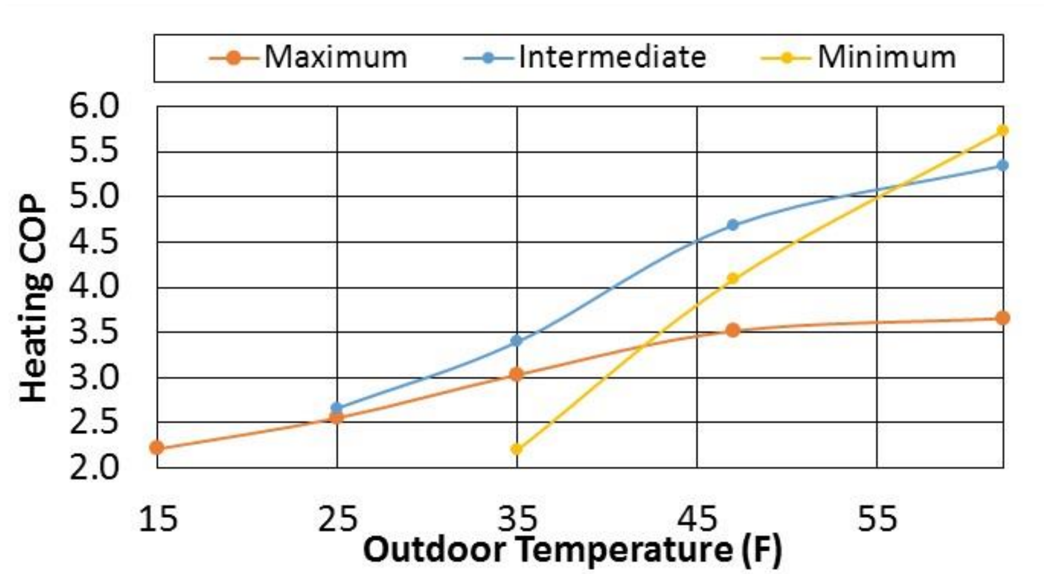
Figure 9: Steady State Performance Results – Heating Capacity



Source: EPRI

Figure 10 provides the corresponding heating efficiency (coefficient of performance) curves corresponding to the heating capacity data shown in e 9. Both figures present the data as a function of outdoor temperature and operation level. When comparing the intermediate efficiency curve to the maximum efficiency curve, similar trends may be observed to those of the previously discussed cooling efficiency curves. For similar test conditions, the efficiency at intermediate operation of the system is higher than when the system is operating at maximum output. As the outdoor temperature decreases, the efficiency increases from maximum to intermediate operation decreases. The efficiency curve for minimum operation only follows these discussed trends at the highest outdoor condition of 62°F (17°C). Below the outdoor condition of 62°F (17°C), the efficiency curve for minimum operation falls between or below the intermediate and maximum operation curves. The variable capacity unit demonstrated acceptable efficiency down to an outdoor temperature of 15°F (–9°C).

Figure 10: Steady State Performance Results – Heating Efficiency

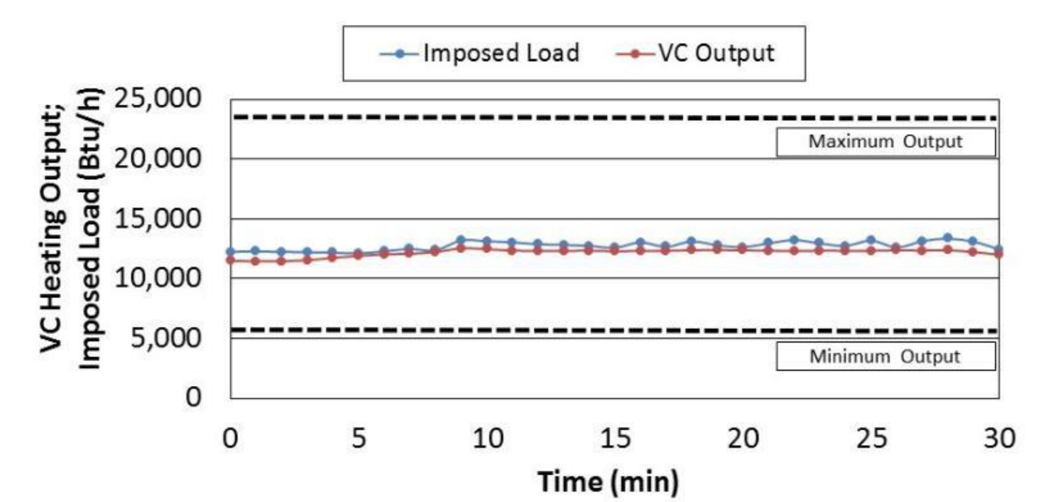


Source: EPRI

Load-Based Heat Pump Performance

To evaluate the controls of the variable capacity heat pump, two load-based laboratory test cases were imposed on the system in heating operation. During both load-based test cases, the outdoor zone conditions were held constant at approximately 47°F (8°C) dry-bulb, 43°F (6°C) wet-bulb. The first load-based test consisted of imposing a steady heating load on the indoor zone of the experimental setup for approximately 30 minutes. The results of the first test case are shown in Figure 11. As seen in the figure, the heating output of the variable capacity heat pump matched the imposed heating load at approximately 12,500 Btu/h. In Figure 11, the variable capacity heat pump demonstrates an ability to match an imposed heating load over a period of time.

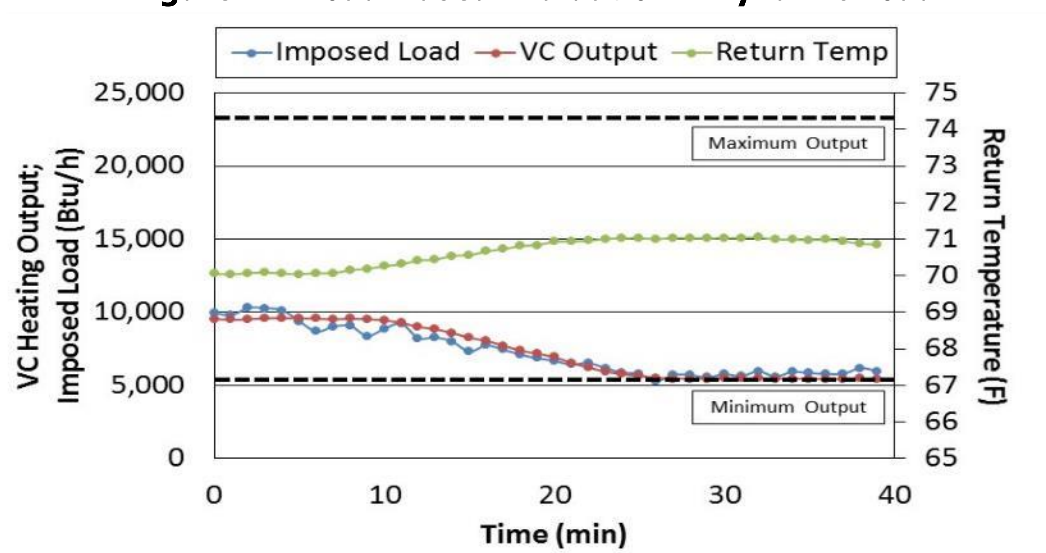
Figure 11: Load-Based Evaluation – Steady Imposed Load



Source: EPRI

The second load-based test consisted of first imposing a steady heating load on the experimental indoor zone and then dynamically modulating the imposed heating load. The results of the second test case are shown in Figure 12. The test case consisted of imposing a steady load of approximately 12,500 Btu/h and then modulating the load to approximately 6,000 Btu/h after a period of time. As seen in the figure, the heating output of the variable capacity heat pump modulated and tracked with the imposed load on the indoor zone. After a period of modulation, the variable capacity heat pump matched the lower imposed load of approximately 6,000 Btu/h. In Figure 12, the variable capacity heat pump demonstrates an ability to modulate heating output in accordance with a dynamic imposed heating load.

Figure 12: Load-Based Evaluation – Dynamic Load



Source: EPRI

Heat Recovery Ventilator Assessment

In conjunction with the steady-state performance evaluation, an assessment of a heat recovery ventilator was performed. The heat recovery ventilator was evaluated under similar indoor and outdoor air conditions as the variable capacity heat pump for a simulated cooling and heating season. The results of the laboratory assessment of the HRV are provided in Table 6. The primary metric used to quantify the performance of the HRV was sensible effectiveness. Sensible effectiveness refers to the ability of the HRV to transfer heat from the exhaust air stream to the fresh air stream. For cooling season conditions, the sensible effectiveness ranged from 0.62 to 0.78 for examined test points, while for heating season conditions, the sensible effectiveness ranged from 0.77 to 0.82 from the laboratory results. The laboratory results of the tested HRV are consistent with industry guidelines for a high efficiency HRV, which aim to provide sensible effectiveness values approaching 0.80.

Table 6: Laboratory Results of Heat Recovery Ventilator Assessment

Season	Outdoor Temp (F)	Return or Exhaust Temp (F)	Supply Temp (F)	Sensible Effectiveness
Cooling	115.1	75.0	83.9	0.78
	105.0	74.7	81.8	0.77
	95.1	75.1	80.7	0.72
	85.3	74.6	78.6	0.62
Heating	46.8	69.8	65.7	0.82
	35.0	69.9	64.1	0.83
	24.8	69.7	60.5	0.79
	14.8	70.0	57.1	0.77

Source: EPRI

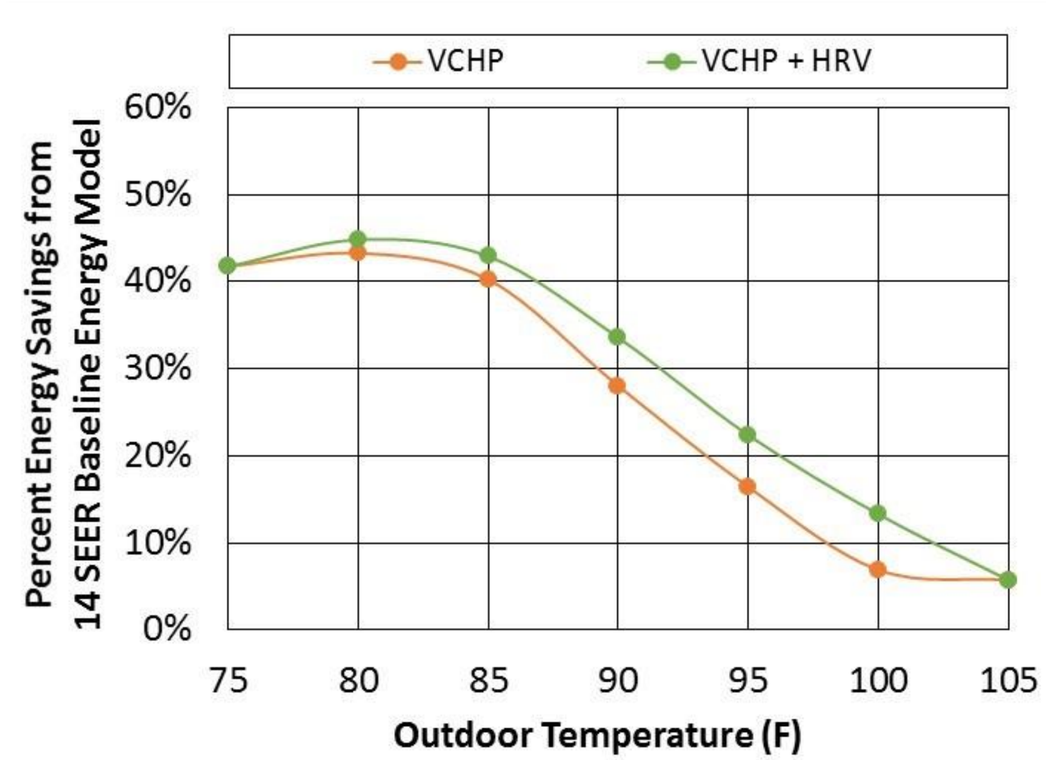
Investigation of Variable Capacity Heat Pump and Heat Recovery Ventilator

Using the results of the laboratory assessment for the variable capacity heat pump and the heat recovery ventilator, an energy model was developed that compared the performance of variable capacity heat pump (VCHP) and HRV to a baseline system. For the energy model comparison, the baseline system was assumed to consist of a 14 SEER air-conditioner, forced air ventilation, and a natural gas 80 percent AFUE furnace. To examine the performance of the VCHP and HRV in greater detail, California Climate Zone 10 was selected to investigate further. California Climate Zone 10 consists of a balanced heating and cooling climate with a cooling design condition of 101°F (38^{o}C) and a heating design condition of 35°F (2^{o}C).

Figure 13 presents the cooling energy savings versus outdoor temperature for the 20 SEER variable capacity system when compared to a baseline 14 SEER air-conditioner in California Climate Zone 10. The variable capacity system was investigated both with and without the implementation of an HRV. As seen in the figure, higher energy savings are expected at milder outdoor temperatures, while lower energy savings are expected at higher outdoor temperatures. This trend in cooling energy savings can be related back to the cooling efficiency curves of the variable capacity system. The addition of the HRV with the VCHP provides improved seasonal energy savings, as well as considerably increased energy savings at higher outdoor temperatures. Overall, the cooling season energy savings for Climate Zone 10 are approximately 30 percent for the investigated 20 SEER variable capacity system and approximately 34 percent for the variable capacity system and the evaluated HRV. At the

105°F (41°C) outdoor temperature, no energy savings were observed with the addition of the HRV because the air-conditioner was operating at 100 percent output based on the modeled conditions. The effect of the HRV in the energy model was determined by a reduced ventilation load as part of the overall building load on the HVAC system. The baseline unit in the energy model assumed a forced air ventilation load as part of the overall building load.

Figure 13: Comparison of Variable Capacity and Heat Recovery Ventilator Equipment to Baseline Air Conditioning



Source: EPRI

Continuing the cooling performance investigation of the examined 20 SEER VCHP and the HRV, the energy model was adjusted to evaluate the effect for all 16 California climate zones. Table 7 provides a compilation of the cooling energy savings for the 20 SEER VCHP and the VCHP coupled with the HRV unit. Similar trends observed previously in Figure 13 carry over to Table 7: Modeled Cooling Savings with Variable Capacity and Heat Recovery Ventilator Equipment. Higher energy savings were often seen for overall milder climates such as San Jose and Los Angeles, while the combination of the VCHP and the HRV offered the largest efficiency improvement over a VCHP in warmer climates such as Fresno and Twentynine Palms.

Table 7: Modeled Cooling Savings with Variable Capacity and Heat Recovery Ventilator Equipment

California Climate Zone	Representative City	Annual Energy Savings – Cooling*	
		VCAC	VCAC + HRV
1	Arcata	—	-
2	Napa	32.3%	+2.3%
3	Oakland	25.5%	+1.8%
4	San Jose	29.6%	+1.8%
5	Santa Maria	28.8%	+1.6%
6	Los Angeles	30.2%	+1.4%
7	San Diego	28.3%	+1.3%
8	Long Beach	29.9%	+1.8%
9	Burbank	29.7%	+3.0%
10	Riverside	30.3%	+3.5%
11	Red Bluff	28.5%	+3.6%
12	Stockton	28.6%	+3.2%
13	Fresno	28.2%	+3.7%
14	Palmdale	25.7%	+3.7%
15	Blythe	22.4%	+3.6%
16	Bishop	28.2%	+3.8%

***Baseline Equipment consists of 14 SEER Air Conditioner and Forced Ventilation**

Source: EPRI

To investigate the heating ability of the VCHP and HRV, the energy model for the California climate zones was expanded to cover the heating season. Table 8 provides the percentage of the modeled annual heating load which could be satisfied solely by the VCHP for each California climate zone. The ability of the variable capacity to satisfy the annual thermal load is shown in Table 8 both with and without the integration of the HRV. For most California climates, the VCHP ability to satisfy the annual thermal load was 90 percent or higher. For the remaining percentage of the annual thermal load, a backup heating source would be needed to satisfy and maintain the conditioned space. For climates in which the VCHP could be used 97 percent or more of the time, the variable capacity heat pump could potentially be implemented without backup heat if designed appropriately for a given application in the climate zone.

Table 8: Heating Ability of Variable Capacity Heat Pump for California Climates

California Climate Zone	Representative City	Percentage of Modeled Annual Heating Load Satisfied by VCHP	
		VCHP	VCHP + HRV
1	Arcata	91.3%	+0.9%
2	Napa	91.1%	+0.9%
3	Oakland	96.9%	+0.3%
4	San Jose	94.2%	+0.7%
5	Santa Maria	91.6%	+0.7%
6	Los Angeles	99.0%	+0.1%
7	San Diego	99.1%	+0.0%
8	Long Beach	97.8%	+0.2%
9	Burbank	96.8%	+0.4%
10	Riverside	93.9%	+0.7%
11	Red Bluff	88.6%	+1.3%
12	Stockton	87.5%	+0.9%
13	Fresno	87.5%	+1.0%
14	Palmdale	88.9%	+1.0%
15	Blythe	95.5%	+0.7%
16	Bishop	59.6%	+0.8%

Source: EPRI

Demand Response Performance

Within the experimental DR setup, the variable capacity system was first operated under a steady-imposed load, which was approximately 90 percent of the unit's maximum capacity output. During the DR assessment, the unit was operated in cooling mode with a fixed outdoor temperature of 105°F (41°C). Two different DR upper load limits were placed on the variable capacity system, namely 60 percent and 30 percent, after the steady period of the unit operating at approximately 90 percent output. The results of the two DR tests are provided in Figure 14.

Figure 14: Demand Response Evaluation of Next-Generation Heat Pump – Unit power consumption (W) over time

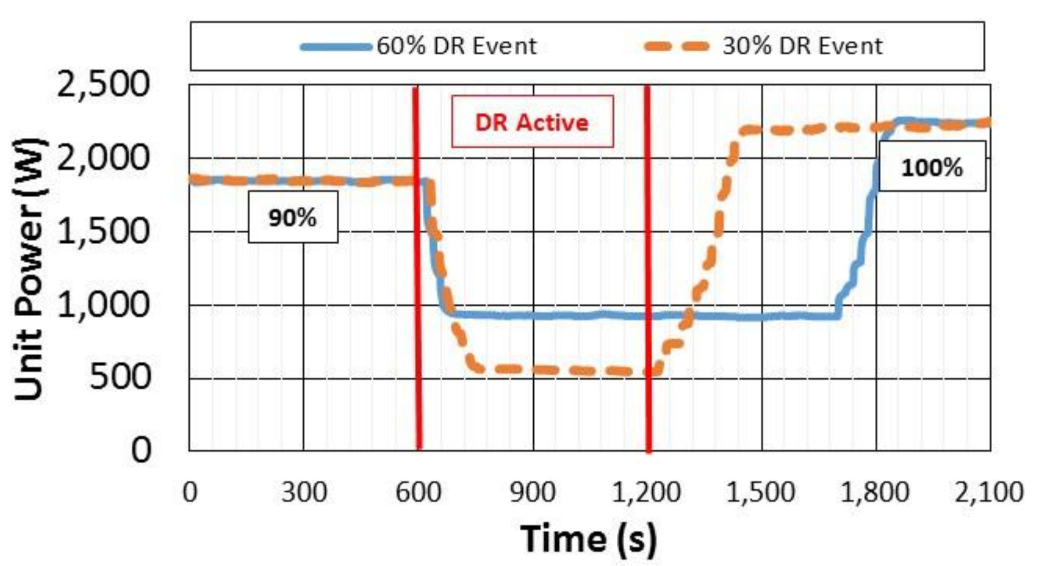
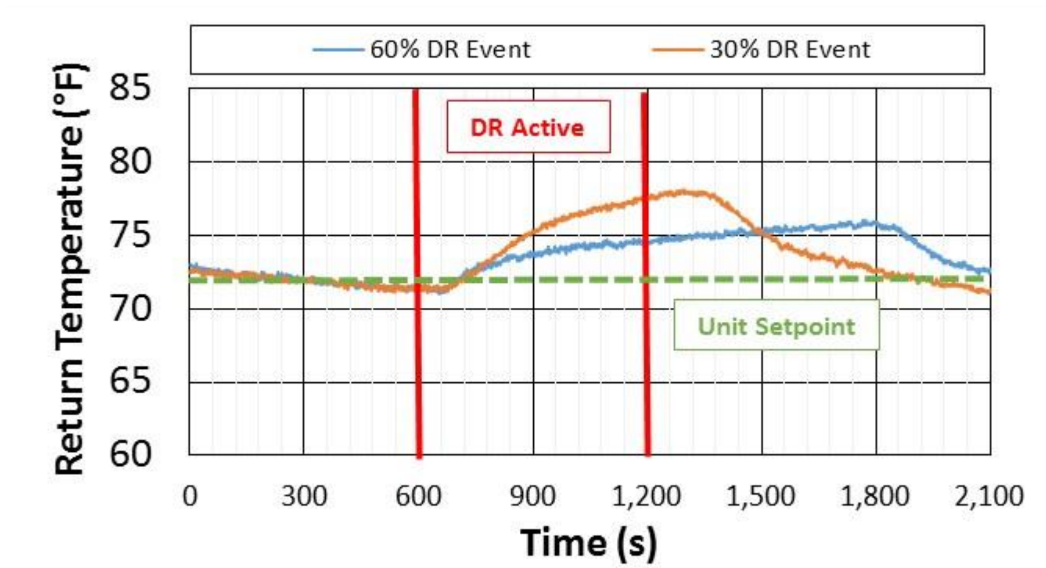


Figure 15: Demand Response Evaluation of Next-Generation Heat Pump - Return air temperature (F) over time for 30% and 60% DR events



Source: EPRI

Figure 14 provides the unit power consumption of the two DR test cases, while Figure 15 provides the return or indoor temperature of the experimental setup with a corresponding time of the testing period. As observed in Figure 14, the variable capacity equipment modulated to the appropriate DR upper limit shortly after activation and throughout the DR event. Upon termination of the DR event, the variable capacity unit resumed normal operation and responded to the condition of the experimental indoor space. Because the imposed cooling load of the indoor space was approximately 90 percent of the unit's maximum output, the return or indoor temperature in the space rose during the DR active period for both the 60 percent and 30 percent test cases.

Table 9 provides further details of the operation of the equipment while steady at 90 percent, the 60 percent DR test case, and the 30 percent DR test case. For the 60 percent test case, the power reduction of the HVAC system was approximately 50 percent, while the power reduction of the 30 percent test case was approximately 70 percent. During both the 60 percent and 30 percent test cases, the variable capacity unit continued to provide a level of cooling capacity to the space. The rise of indoor or return temperature for a given residential application will be dependent upon the building load, temperature setpoint, and building construction. In the experimental setpoint, the indoor zone was a near adiabatic thermal chamber and thus the indoor temperature rose quickly during the DR active interval.

Table 9: Response of Next-Gen Heat Pump in Demand Response Evaluation

	Unit Power (W)	Percent Power Reduction	Approximate Cooling Capacity (Btu/h)	Peak Return Temperature (F)	Percent Capacity Reduction
Baseline Case, Steady at 90%	1,866	—	17,000	71.8	—
60% DR Event, 50% Power	928	50.3%	10,500	75.8	38.2%
30% DR Event, 30% Power	558	70.1%	6,500	77.7	61.8%

Source: EPRI

Pacific Gas and Electric Company Laboratory Activities and Results

Phase 1 testing by PG&E involved laboratory testing of a prototype system under both cooling and heating modes, but with an emphasis on heating and the dual-fuel capabilities. The plan included testing at defined environmental conditions to create system performance maps that can be used for computer modelling. Additional testing included dynamic conditions to demonstrate the system controls for dual-fuel heating, and to evaluate the system response to demand response events.

The capacity of the system was measured by three distinct methods to assess stability and measurement accuracy:

- Indoor air-side cooling and heating
- Outdoor air-side cooling
- Indoor refrigerant-side cooling and heating

The primary capacity measurement is the indoor air-side measurement, and all of the capacity values presented in the report are from these measurements. Having redundant measurements provides test validation as long as they agree within a suitable tolerance (6 percent per ASHRAE Standard 37-2009).

All of the system components had their electric power and gas consumption measured separately, with the total energy input used with the measured capacity to determine the system efficiency.

Laboratory Setup

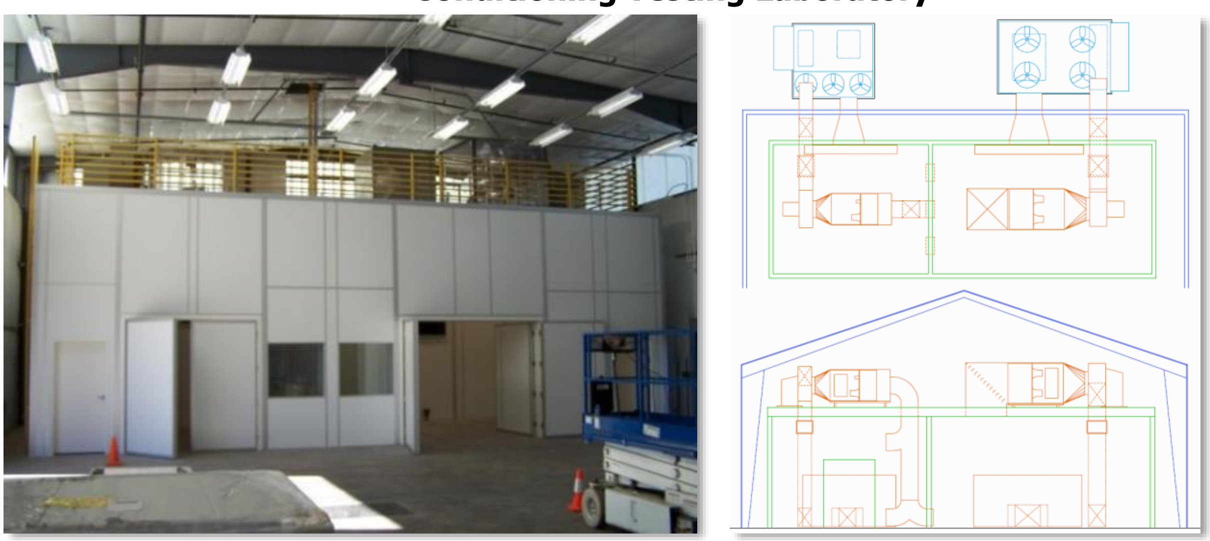
Laboratory Facility

All testing described in this report was performed in the HVAC testing apparatus in the Advanced Technology Performance Lab at PG&E's San Ramon Technology Center. The testing apparatus consists of two side-by-side environmental chambers designed in accordance with ASHRAE Standard 37-2009. Each chamber has an independent space conditioning system to control temperature and humidity, consisting of packaged commercial heat pump units with electric resistance heating elements to fine-tune the temperature, and separate electric resistance heated humidifiers. The packaged units are equipped with economizers that allow the test chambers to be flushed with outside air, which can provide stability during a test.

Each room has its own airflow measurement apparatus constructed according to ASHRAE standard design. This consists of a sealed box with a partition having several flow nozzles that can be opened or closed in combination to provide the required range of differential pressure for the current airflow. Variable-speed blowers on the outlets of each airflow station can be set to maintain the desired outlet static pressures or airflow rates and compensate for the added resistance of the flow measurement system and ductwork. The airflow stations are positioned on the roof of the test chambers to provide extra space for large test units.

The smaller of the two chambers (left side of Figure 16) is designated the "indoor room" and conditioned to maintain the required return air conditions to the test unit, and its airflow apparatus is used to measure the supply airflow from the indoor coil. The larger chamber, or "outdoor room," is used to maintain the required air conditions to the outdoor components, and its airflow apparatus is used to measure the outdoor unit exhaust airflow.

Figure 16: Pacific Gas and Electric Company Heating, Ventilation, and Air Conditioning Testing Laboratory



Source: EPRI

Experimental Setup

The test air conditioning system was installed as a split system with components separated into the two environmental chambers, as shown in Figure 17. The outdoor components (condensing units) were installed in the outdoor room, and the discharge of the condensing unit was connected to an airflow measurement station during cooling mode testing. The indoor components (furnace and coil) were installed in the indoor room and attached by ducting to the airflow measurement apparatus. A Coriolis mass flow meter was installed in the liquid line to measure refrigerant flow as an additional means of determining capacity. (Please refer to Appendix A for a complete list of instrumentation).

Figure 17: Test Unit Installation

Source: EPRI

For the indoor room, a natural gas supply was added to provide fuel to the furnace, and it was equipped with instrumentation to measure the flow rate and quantity of fuel consumed, as well as pressure and temperature measurements to convert to standard volume conditions. Since the test lab is unoccupied, the furnace exhaust products could be discharged into the test chamber and displaced by ventilation air from the space conditioning system.

When the laboratory was originally constructed, there was no provision for taking the outdoor room down to lower temperatures than what the packaged space conditioning system could achieve on its own. To achieve the low temperatures required for heat pump testing, an insulated sub-chamber was constructed to surround the outdoor unit, with dimensions that satisfied the clearance requirements of ASHRAE Standard 37 (a minimum of 3 feet to any air intake surface [sides], and a minimum of 6 feet from any air discharge [top]). The enclosure was equipped with a separate refrigeration system to achieve low temperatures, and the temperature setpoint was controlled by modulating airflow into the sub-chamber from the main chamber space conditioning system by a combination of butterfly dampers in supply ducts and varying the speed of the blower on the exhaust airflow measurement apparatus. Because the airflow apparatus would be used for temperature control, the ducting between it and the test unit would be disconnected and thus the outdoor unit airflow would not be measured during heat pump testing. (Only the indoor air side and refrigerant flow methods of capacity measurement can be done.) This also allowed the test unit itself to help cool the air in the sub-chamber when in heat pump mode.

Data Acquisition

All of the instruments were connected to signal conditioning modules based on the National Instruments C-series architecture, connected to four Compact-RIO chassis. The modules included different units for resistance temperature detectors (RTDs), thermocouples, voltage, current, and pulse counting, plus both analog and digital output modules to control the room conditioning systems. Two of the Compact-RIO chassis were connected by serial cables to a weather station and to the power meter to record their measurements digitally. The default chassis internal scan rate for reading the module inputs is 10 Hertz (Hz), although the weather station and power meter updated once every second.

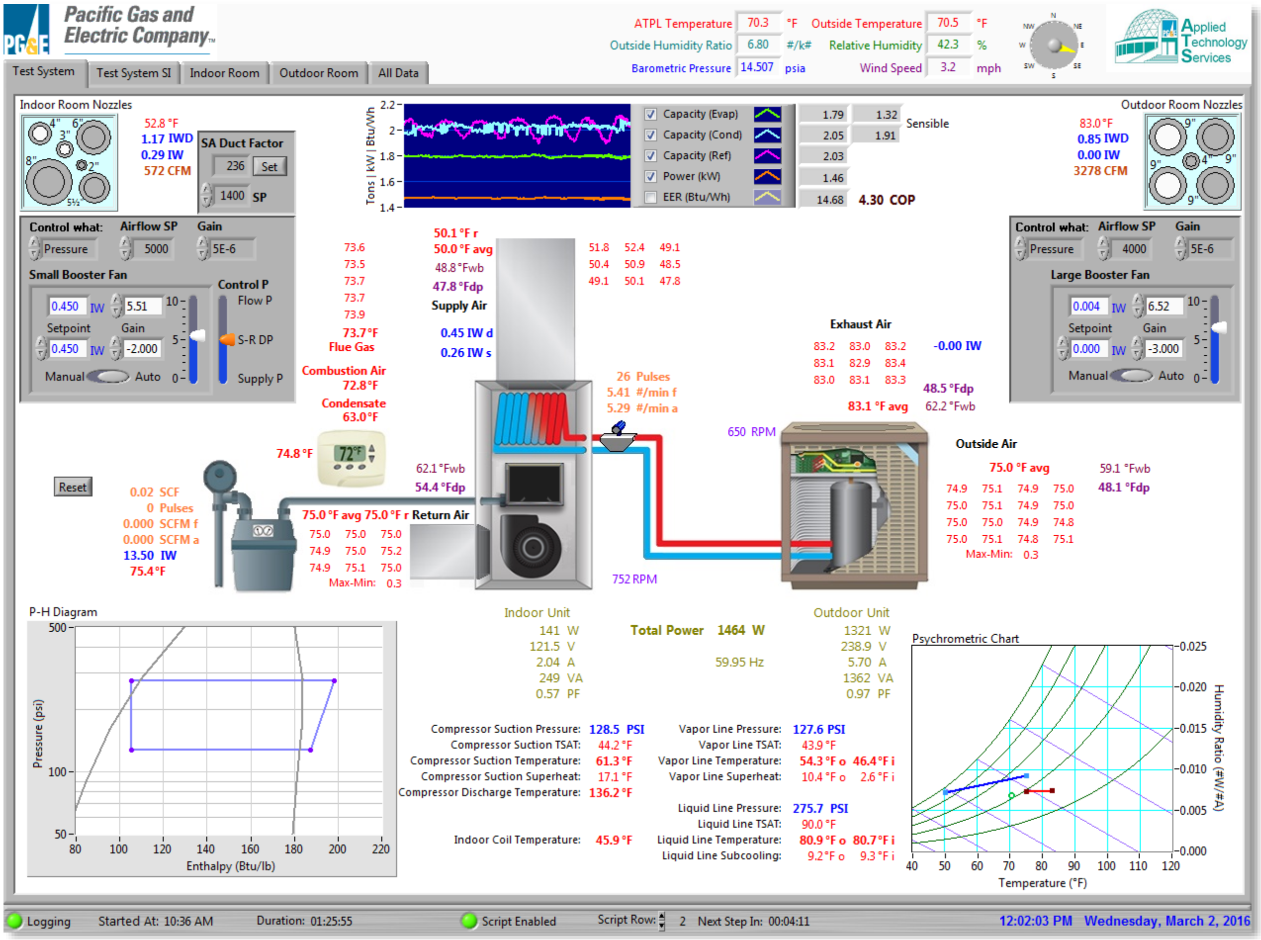
The four Compact-RIO chassis communicate over an Ethernet network to a central host computer, which ran a custom data acquisition and control program developed with National Instruments LabVIEW™ graphical programming language. The program acquired readings from the chassis at a rate of once per second, applied calibration scaling and maintained a running average for each measurement, and logged the averages to a file every 5 to 15 seconds. The scaled values and other calculated values were also displayed on screen in both text and graphical form and used to generate feedback control signals to the space conditioning systems, as shown in Figure 18. The program also included the ability to run scripts that could change the setpoints for the chamber conditioning systems at specific times of the day or after specific time intervals.

The logged data were saved in a text format that is easily imported into Microsoft Excel for analysis. A macro was developed to run on the raw data file to apply formatting, calculate statistics, and create trend charts. The result was then analyzed to isolate a period of stable operation. For most of the "standard" tests, the target period duration was 30 minutes, although shorter duration periods were accepted when thermal stability was not critical (for example, fan performance mapping), or on rare occasions when some operating anomaly reduced the acceptable data set. Once this period is identified, the statistics (average, standard deviation, range) were isolated to just this period and then copied to another spreadsheet with one row per test. Operating performance metrics were then calculated from these values, and the results were checked for the test tolerances specified in ASHRAE Standard 37.

To conduct fixed-condition tests, the manufacturer provided a software program that communicated through a cable between a computer's serial port and the outdoor unit and overrode the unit's thermostat. This program was used to put the system into three different fixed cooling modes (1: Low, 2: Intermediate, and 3: High) and three different heating modes by adjusting the "Heating_Test_Mode" setting, as shown in Figure 19. This program also had its own logging capability, but only for the temperature and pressure measurements. When recorded, this data log was combined with the log from the main program for trending confirmation.

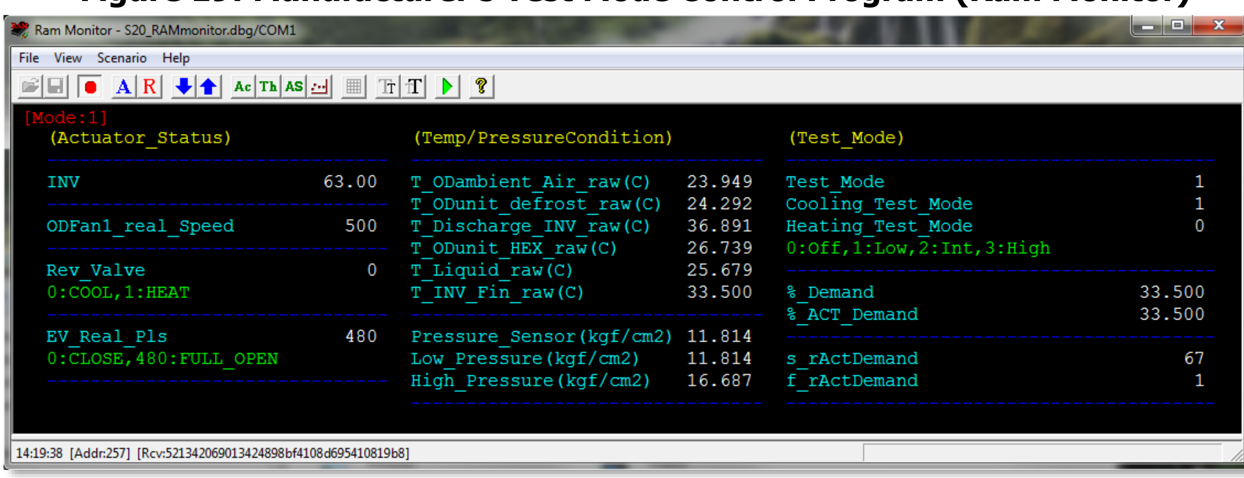
In addition to the connection to the main data acquisition system, the power meter was also connected through an Ethernet connection to a stand-alone laptop computer. This computer logged readings from the power meter and created a display of current readings, and both of these were uploaded every minute to cloud storage where the manufacturer's representatives could access them.

Figure 18: LabVIEW™ Data Acquisition Main Screen



Source: EPRI

Figure 19: Manufacturer's Test Mode Control Program (Ram Monitor)



Source: EPRI

Cooling Mode Testing

Plan for Standard Tests in Cooling Mode

The test plan followed the current rating standards, specifically Table 10 in AHRI Standard 210/240-2008, which is also incorporated into the Department of Energy (U.S. DOE) regulations (U.S. DOE Title 10). In addition to the standard tests, several performance mapping tests as a function of outdoor temperature were conducted to fill in the gaps in the performance trend, and also to examine performance under a more California-climate appropriate return air condition, specifically, 75°F dry bulb temperature (db)/62°F wet bulb temperature (wb) versus the AHRI standard 80°F db/67°F wb (Table 10).

Table 10: Planned Cooling Mode Standard Tests

Test Description	Air Entering Indoor Unit Temperature		Air Entering Outdoor Unit Temperature	Compressor Speed and Cooling Air Volume Rate
	Dry-Bulb (°F)	Wet-Bulb (°F)	Dry-Bulb (°F)	
AHRI A_2 Test	80	67	95	Maximum
AHRI B_2 Test	80	67	82	Maximum
AHRI E_V Test	80	67	87	Intermediate
AHRI B_1 Test	80	67	82	Minimum
AHRI F_1 Test	80	67	67	Minimum
AHRI G_1 Test	80	Dry Coil	67	Minimum
AHRI I_1 Test (Cyclic)	80	Dry Coil	67	Minimum
AHRI Maximum Conditions	80	67	115	Maximum
Performance Mapping AHRI Indoor Conditions	80	67	75, 85, 95*, 105, 115*	Maximum & Minimum
Performance Mapping California Indoor Dry Climate	75	62	75, 85, 95, 105, 115	Maximum & Minimum

*** Mapping test conditions already included in standard tests at maximum**

Source: EPRI

The majority of these were steady-state tests, where stable conditions were maintained for at least 30 minutes within specified tolerances. The exception was the I_1 test, which was an on/off cycling test at minimum speed. For variable speed systems like the test unit, the cycling period was specified to be 48 minutes off followed by 12 minutes on to complete an hour cycle. The cyclic test was conducted immediately following a G_1 test to record the steady state performance at the same conditions for comparison. Both of these tests were conducted at a return air dew point temperature that was below the operating temperature of the evaporator coil to avoid condensation, as the measurement of humidity in the supply duct was not considered to be accurate under unsteady conditions. The on/off cycling of the unit had to be done manually using the Ram Monitor program, toggling between Cooling_Test_Mode settings of 0 and 1.

The planned tests were also doubled from what was originally planned in that all the testing was proposed to be done using the indoor unit external resistance as specified in the current AHRI Standard at 0.10 inches of water (IW) (24.884 Pa). A proposed revision to the AHRI standard specifies a more realistic value of 0.45 IW (111.978 Pa), and so the entire list of test points was repeated at the higher resistance. The resistance specification only applies when the unit was operated at its maximum airflow setting. At the minimum and intermediate settings, the pressure would be reduced while maintaining a constant duct coefficient, defined as follows:

$$C_{DUCT} = \frac{CFM}{\sqrt{IW/\rho_{SA}}}$$ **Equation 1**

where ρ_{SA} is the supply air density. The test conducted at the maximum compressor speed and the corresponding indoor blower speed was done at the prescribed fixed external resistance, and the measured airflow rate through the indoor unit was then used to establish the duct coefficient. This was then used as a constant with this equation rearranged to calculate the appropriate external static pressure setpoint for the measured airflow rate.

Results for Standard Tests in Cooling Mode

The cooling mode standard tests were conducted between February 19 and March 24, 2016. Some earlier tests conducted between January 7 and February 17 had to be discarded after a large leak was discovered in some of the installed ductwork.

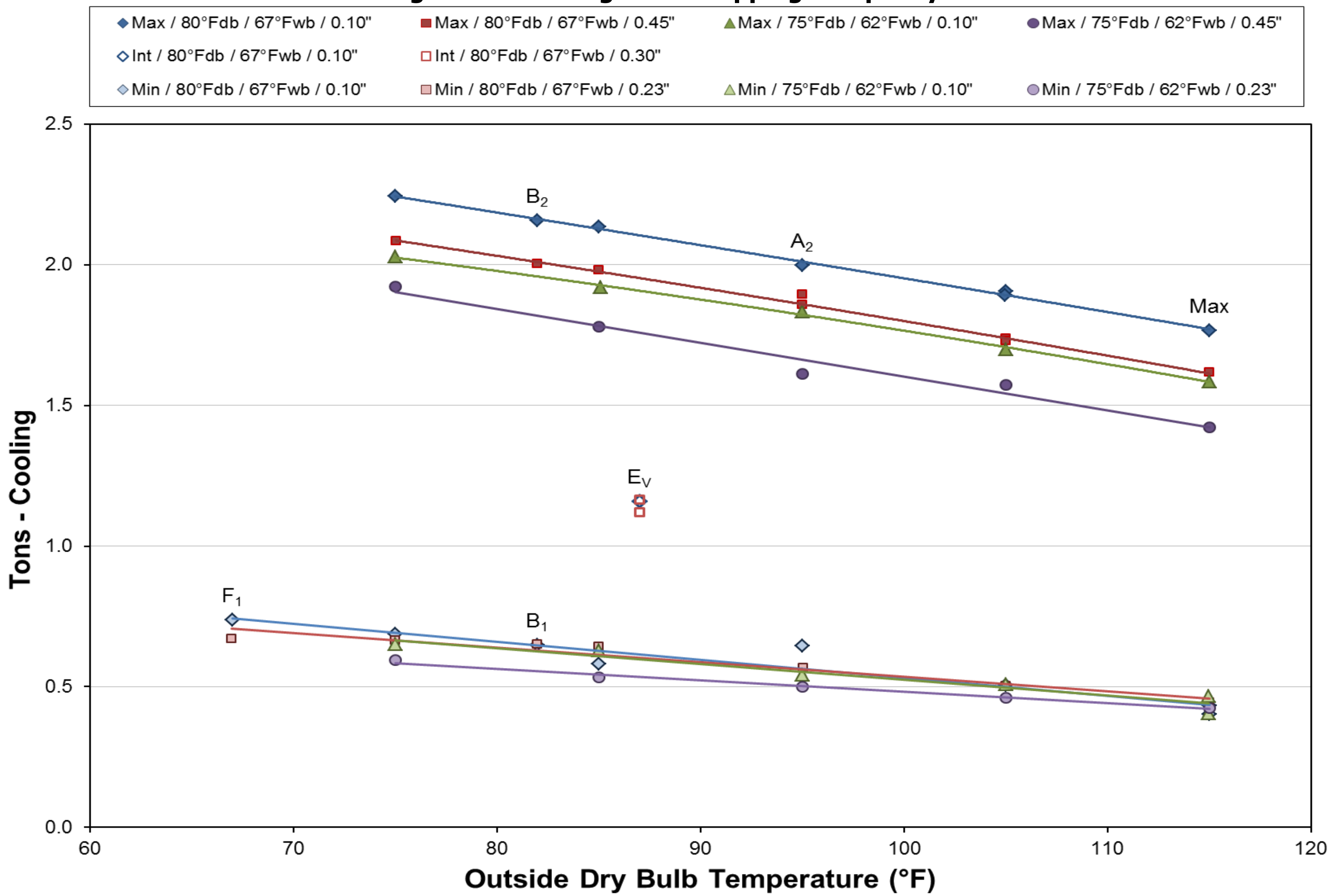
The results from this phase of testing are given in Table 11, but perhaps are better viewed in the following three figures. The three figures present key performance metrics (capacity, power, and efficiency) as a function of the outside temperature, divided into groups of return air conditions, external resistance, and compressor speed setting.

Figure 20 presents cooling capacity in units of tons (12,000 British thermal units per hour [Btu/hr]), as calculated from the evaporator air side measurements. The A_2 test conducted at maximum speed, standard rating conditions, and the current standard's external resistance of 0.10 IW actually produced the rated 2-tons of capacity for the system. Deviating from these standard rating conditions to the other alternatives caused a reduction in capacity. The next step down was when it was operated at the proposed external resistance of 0.45 IW due to a combination of reduced airflow and higher blower power, for a reduction of about 6 percent. This was followed closely by the test done using the alternative return air, which had about an 8 percent decrease from the standard. Most of this reduction was due to a reduced latent capacity since this air contained less water vapor. The lowest of the four curves at maximum speed was the combination of the two effects, for a loss of 19 percent.

Figure 20 shows the corresponding trends of total power (indoor unit plus outdoor unit) for the same data groupings. Worth noting in Table 11 is that the portion of the total power derived from the outdoor unit was only a function of the outdoor temperature and was mostly unaffected by the different return air temperatures and external resistance. Figure 21 shows the ratio of capacity to power as the energy efficiency ratio (EER)[1] in Btu/Wh. The advantage of the variable speed system is demonstrated by this chart, which shows that at mild temperatures when the unit can be running at low speed, the efficiency improves dramatically.

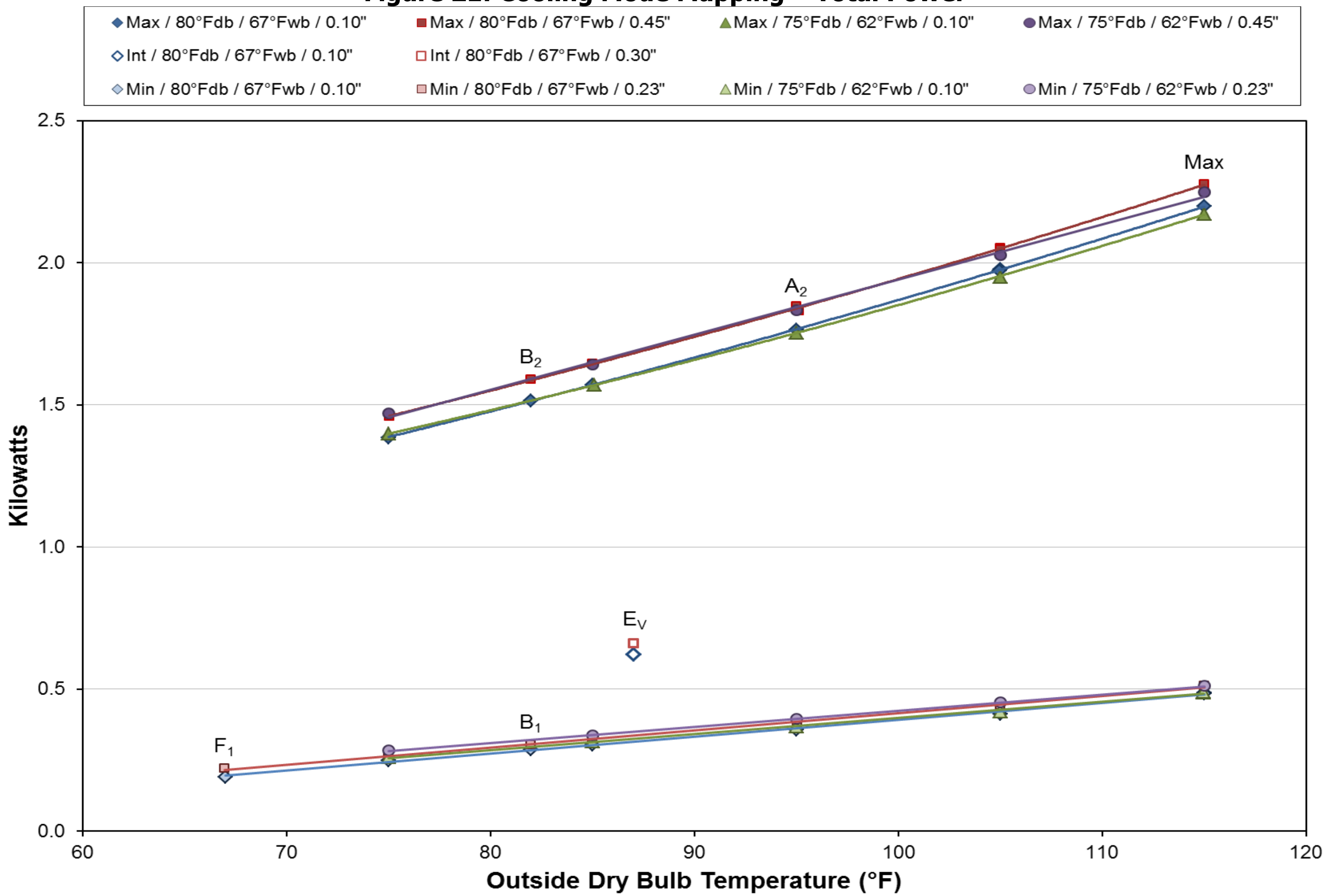
[1] Energy Efficiency Ratio (EER) (Btu/Wh) = 3.41214 x COP (Coefficient of Power). To convert the scale in Figure 21 to COP, divide by 3.42124.

Figure 20: Cooling Mode Mapping – Capacity



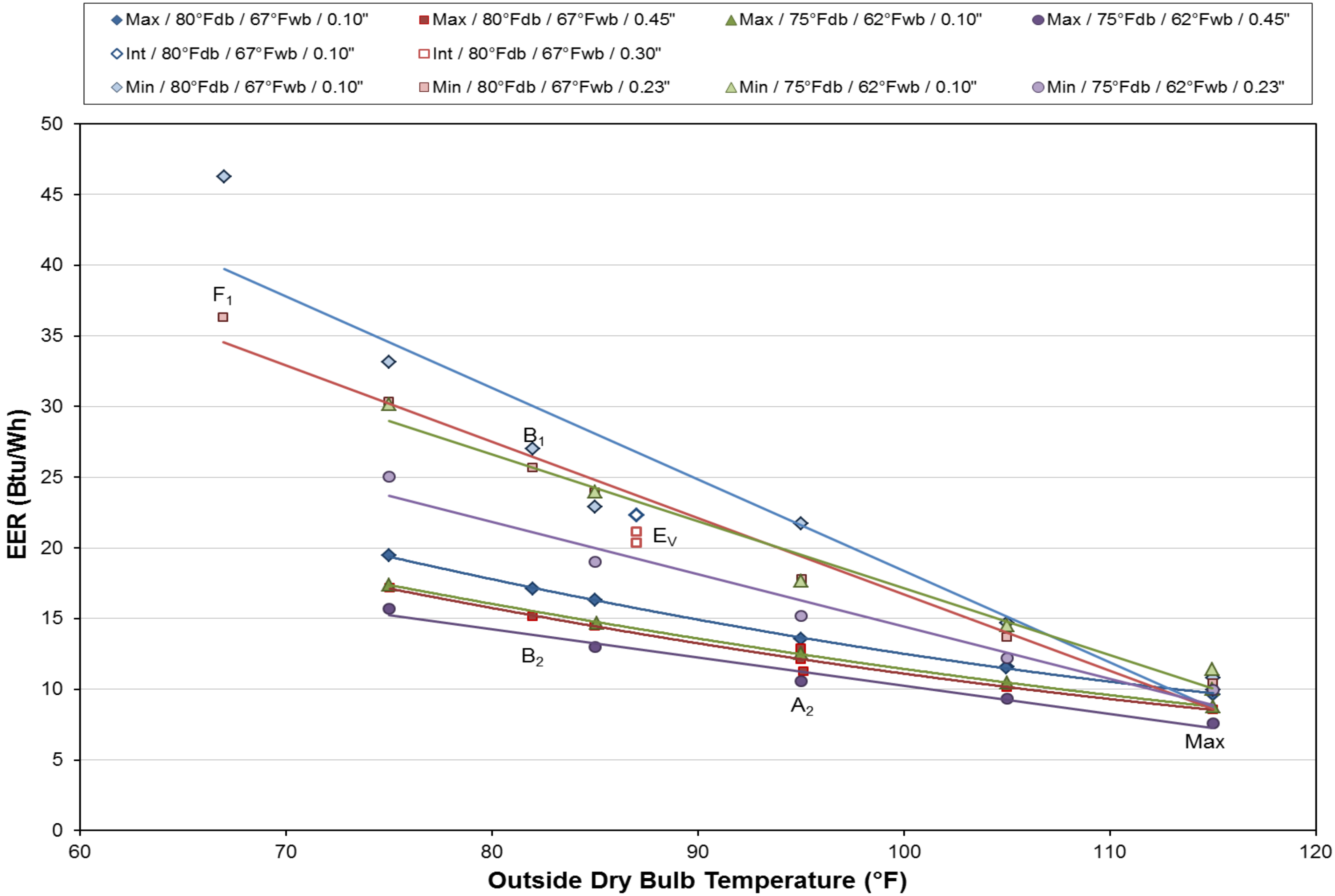
Source: EPRI

Figure 21: Cooling Mode Mapping – Total Power



Source: EPRI

Figure 22: Cooling Mode Mapping – Efficiency (EER)



Source: EPRI

One concern from these tests was that the indoor unit airflow was much less than would be expected for this system, even after fixing the supply duct leakage problem. For example, for the A_2 test at 0.10 IW, the measured airflow was around 600 CFM, when 700 to 800 CFM would be expected for a 2-ton rated system. There are apparently ways to adjust the airflow within either the furnace or the thermostat, but these were not examined or altered from what the installing contractor set. Since the system was still cooling at close to its rated capacity, the issue was not investigated.

Table 11: Cooling Mode Standard Test Results

Test Description	Compressor Speed and Cooling Air Volume Rate	Outside Air Tdb (°F)	Return Air Tdb (°F)	Return Air Twb (°F)	Ext. Res. (IW)	Cooling Capacity (Tons)	Power (kW) IDU	Power (kW) ODU	Power (kW) Total	EER (Btu/Wh)	Indoor Blower Airflow (CFM)	Indoor Blower Speed (RPM)	Outdoor Fan Airflow (CFM)	Outdoor Fan Speed (RPM)
A_2	Maximum	95	80	67	0.10	**2.00**	0.069	1.70	1.76	**13.6**	604	514	3,284	650
					0.45	1.88	0.143	1.70	1.84	12.2	579	758	3,278	650
B_2	Maximum	82	80	67	0.10	2.16	0.069	1.45	1.52	17.1	602	514	3,264	650
					0.45	2.00	0.142	1.45	1.59	15.1	577	757	3,259	650
E_V	Intermediate	87	80	67	0.10	1.16	0.052	0.57	0.62	22.3	504	466	2,563	500
					0.31	1.14	0.088	0.57	0.66	20.8	485	636	2,523	500
B_1	Minimum	82	80	67	0.10	0.65	0.043	0.25	0.29	27.0	437	429	2,491	500
					0.24	0.65	0.064	0.24	0.30	25.7	427	556	2,517	500
F_1	Minimum	67	80	67	0.10	0.74	0.043	0.15	0.19	46.3	436	429	2,470	500
					0.24	0.67	0.064	0.16	0.22	36.3	426	556	2,504	500
G_1	Minimum	67	80	61	0.10	0.54	0.042	0.13	0.17	37.4	443	426	2,500	500
					0.23	0.56	0.061	0.15	0.21	32.4	413	542	2,500	500
I_1	Minimum (Cyclic)	*67*	*80*	*61*	0.10	0.105			0.069	18.1	Cyclic			
					0.23	0.099			0.072	16.5				
Performance Mapping	Maximum	115	80	67	0.10	1.77	0.069	2.13	2.20	9.6	605	512	3,298	650
					0.45	1.62	0.142	2.13	2.27	8.5	578	755	3,291	650
			75	62	0.10	1.59	0.069	2.10	2.17	8.8	603	511	3,293	650
					0.45	1.42	0.142	2.11	2.25	7.6	577	754	3,284	650
		105	80	67	0.10	1.90	0.069	1.91	1.97	11.5	604	513	3,288	650
					0.45	1.73	0.142	1.91	2.05	10.2	577	755	3,290	650
			75	62	0.10	1.70	0.069	1.88	1.95	10.5	601	511	3,295	650
					0.45	1.57	0.142	1.89	2.03	9.3	575	753	3,288	650
		95	80	67	0.10	2.00	0.069	1.70	1.76	13.6	604	514	3,284	650
					0.45	1.88	0.143	1.70	1.84	12.2	579	758	3,278	650
			75	62	0.10	1.83	0.069	1.68	1.75	12.6	601	512	3,287	650
					0.45	1.61	0.142	1.69	1.83	10.6	574	754	3,281	650
		85	80	67	0.10	2.13	0.069	1.50	1.57	16.3	603	515	3,265	650
					0.45	1.98	0.142	1.50	1.64	14.5	578	757	3,269	650
			75	62	0.10	1.92	0.069	1.50	1.57	14.7	598	511	3,280	650
					0.45	1.78	0.141	1.50	1.64	13.0	570	752	3,274	650
		75	80	67	0.10	2.24	0.069	1.32	1.39	19.4	603	513	3,256	650
					0.45	2.09	0.142	1.32	1.46	17.2	577	756	3,260	650
			75	62	0.10	2.03	0.069	1.33	1.40	17.4	602	511	3,269	650
					0.45	1.86	0.141	1.33	1.47	15.2	571	752	3,270	650
	Minimum	115	80	67	0.10	0.42	0.042	0.44	0.48	10.3	441	427	2,556	500
					0.23	0.44	0.062	0.45	0.51	10.4	419	545	2,533	500
			75	62	0.10	0.44	0.042	0.45	0.49	10.7	436	423	2,572	500
					0.23	0.42	0.062	0.45	0.51	10.0	420	544	2,572	500
		105	80	67	0.10	0.50	0.042	0.37	0.41	14.7	440	427	2,549	500
					0.23	0.50	0.062	0.38	0.44	13.6	418	545	2,515	500
			75	62	0.10	0.51	0.042	0.38	0.42	14.5	438	424	2,573	500
					0.23	0.46	0.062	0.39	0.45	12.2	419	544	2,562	500
		95	80	67	0.10	0.64	0.042	0.31	0.36	21.7	439	427	2,543	500
					0.23	0.57	0.062	0.32	0.38	17.8	418	545	2,510	500
			75	62	0.10	0.54	0.042	0.33	0.37	17.7	437	424	2,564	500
					0.23	0.50	0.062	0.33	0.39	15.2	417	544	2,554	500
		85	80	67	0.10	0.58	0.043	0.26	0.30	22.9	436	429	2,503	500
					0.24	0.64	0.064	0.26	0.32	23.9	428	557	2,521	500
			75	62	0.10	0.63	0.043	0.27	0.31	24.0	444	428	2,513	500
					0.23	0.53	0.062	0.27	0.34	19.0	416	544	2,532	500
		75	80	67	0.10	0.69	0.043	0.21	0.25	33.1	436	429	2,481	500
					0.24	0.67	0.064	0.20	0.26	30.3	426	556	2,513	500
			75	62	0.10	0.65	0.043	0.22	0.26	30.2	444	428	2,505	500
					0.23	0.59	0.062	0.22	0.28	25.0	415	545	2,521	500

Source: EPRI

Calculation of Seasonal Energy Efficiency Ratio

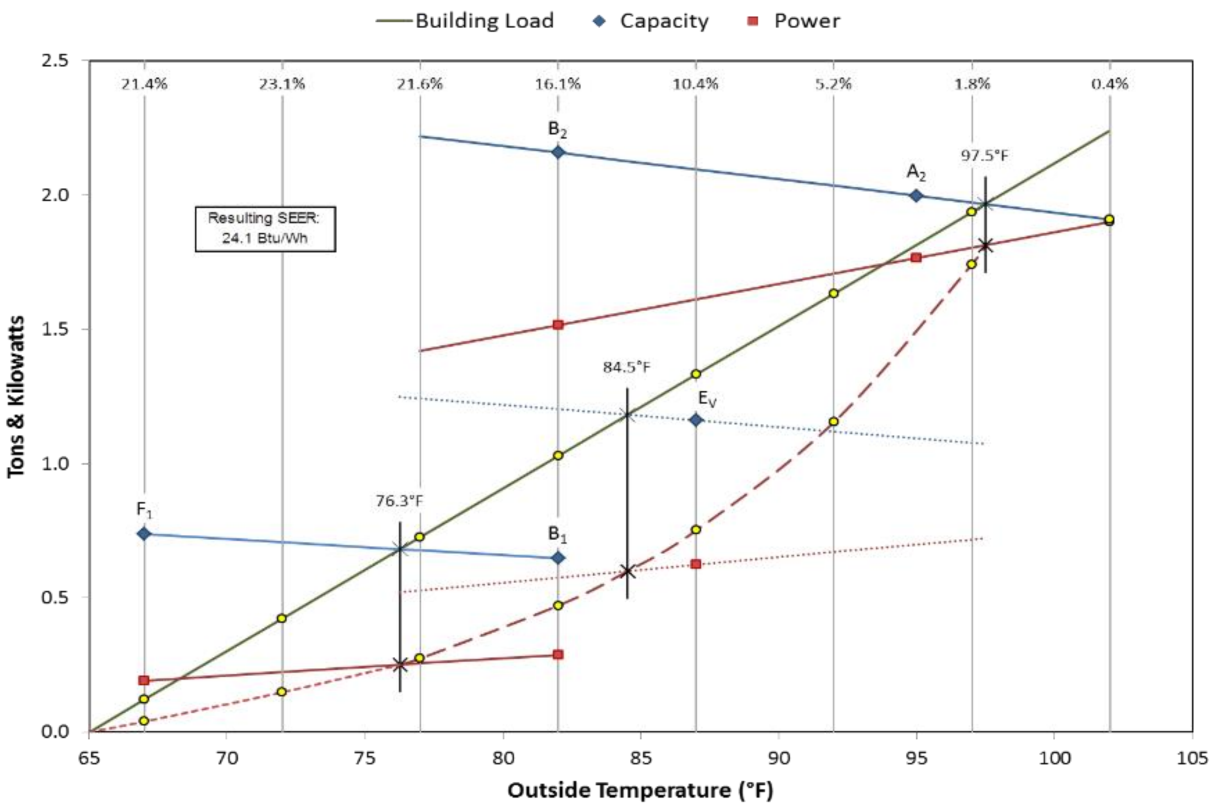
The AHRI calculation of seasonal energy efficiency ratio (SEER) for a variable speed system requires a bit of mathematical gymnastics to estimate what the average system efficiency would be through a cooling season. It is a bin method calculation using representative temperatures in 5°F (2.8°C) increments to calculate the cooling load and power consumed by the air conditioner at those temperatures, which are then weighted by the number of hours in a particular cooling season that the temperatures in the bins occur. The binned ton-hours and kWh are then summed, and the sums divided into each other to determine the SEER.

The building load is defined in the AHRI and U.S. DOE standards as:

$$\mathrm{BL} = \frac{(\mathrm{OAT}-65)}{95-65} \times \frac{\dot{Q}_c^{k=2}}{1.1} \qquad \textbf{Equation 2}$$

When graphed against outside temperature (see Figure 23), this produces a line that extends from zero at 65°F (18°C) through the standard rated capacity (A_2 test) divided by 1.1 at 95°F (35°C). The next step is to determine how much power the system will use to satisfy this building load, and that involves finding the temperatures where this building load line intersects the system capacity trends. At maximum speed, the system capacity is represented by a line drawn through the capacity values from the A_2 and B_2 tests. In Figure 23, that occurs at a temperature of 97.5°F (36.7°C).

Figure 23: Test Data Used to Derive SEER (at 0.10 IW)



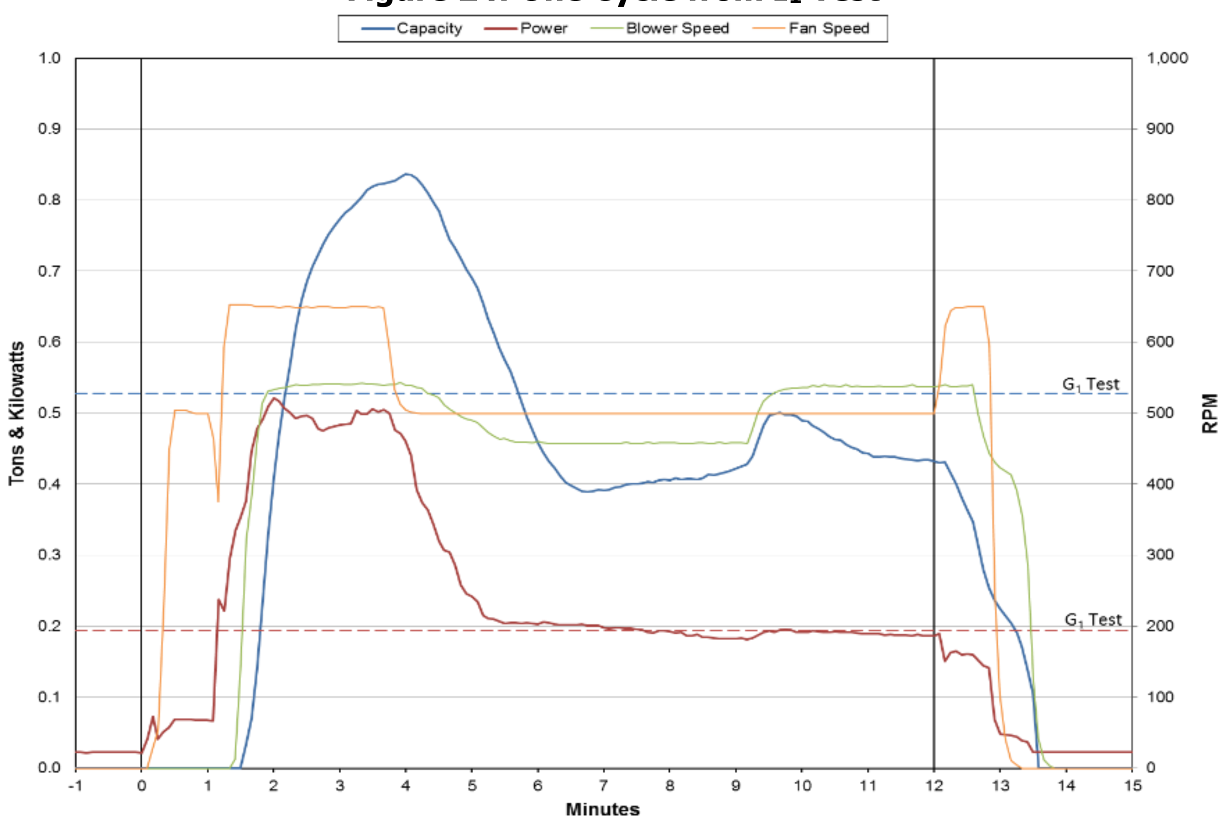
Source: EPRI

At outside temperatures above this, the system will be running constantly at maximum capacity and its power is that defined by the line drawn through the power values from the A_2 and B_2 tests. Under these conditions, the building load is greater than the system capacity, so it will not be able to maintain the indoor thermostat setpoint temperature.

The steady-state minimum speed capacity is represented by a line drawn through the capacity measurements from the B_1 and F_1 tests, which in this case intersects the building load line at 76.3°F (24.4°C). Below this temperature, the minimum speed capacity will be greater than the building load and the air conditioner is assumed to cycle between off and minimum speed. To determine the power for the temperature bin, the calculation involves finding the fraction of time that the system will need to operate to meet the load, and also applies a degradation coefficient to account for the non-steady-state operation.

The degradation coefficient may be derived from the optional G_1 and I_1 tests, which compare the integrated total cooling and power consumed during a 48-minute off / 12-minute on cycle (I_1) to what it would have been under steady-state operation (G_1). In a typical fixed speed compressor system, the capacity rises from zero at the start of an on cycle to asymptotically approach the steady-state capacity; but this system behaved differently. An example of the on cycle for this variable capacity unit is given in Figure 24 (the third of three cycles conducted following the G_1 test).

Figure 24: One Cycle from I_1 Test



Source: EPRI

The trends show that for the first minute after the unit was triggered to come on, only the outdoor unit fan was operating. Eventually, the compressor and indoor blower began to operate, but at a level higher than minimum to get the system moving. The capacity and power during this period actually exceeded their steady-state values. After a few minutes, the system began to slow down to the point where the total power was on par with that from the steady-state test. Capacity actually dipped below the steady-state value before rebounding when the blower speed was increased. When the system was triggered to turn off at the end of 12 minutes, the outdoor unit fan sped up for about one minute, after which the compressor and indoor blower finally turned off. Because of this unusual operational trend, the actual degradation coefficient calculated from these tests was 0.64; but the standard caps it at 0.25; making these two tests unnecessary. If the uncorrected value was used, it would have reduced the SEER result by about 4 percent.

Between the minimum and maximum speed intersection temperatures (76.3°F to 97.5°F [24.4°C to 36.7°C]), the system was assumed to be running at an intermediate speed. To determine the power consumption at these temperatures, pseudo-performance trend lines were drawn through the capacity and power values measured from the single intermediate speed test (E_V). These trends (shown as dotted lines in Figure 23) are weighted averages of the slopes of the maximum and minimum speed trend lines. Once again, the intersection of the capacity line and the building load line is determined, and in this case it was at 84.5°F (28.9°C).

The EER of the system was then determined at each of the three intersection temperatures, and the three points were used to create a second-order curve fit of EER as a function of outside temperature. When combined with the building load line, the power for the intermediate speed temperatures could be determined. The trend of power derived from the building load and EER curve fit is shown as the dashed parabolic curve in Figure 23, showing that it passes through the three points on the minimum, pseudo-intermediate, and maximum power trend lines at the intersection temperatures.

Finally, the load or capacity and power were calculated for each of the bin temperatures (indicated by the yellow circles) and multiplied by the weighting factors prescribed in the AHRI Standard, as shown at the top of the temperature lines. The end result of the bin calculation is a SEER of 24.1 when the external resistance was held to 0.10 IW; reducing to 21.9 when the external resistance at maximum speed was raised to 0.45 IW.

Cooling Mode Dynamic Test Plan

While standard steady-state tests under constant environmental conditions artificially hold the operation of the test unit fixed, dynamic system testing was intended to provide a more real-world representation of the system operation and involve the actual system controls (that is, the thermostat). This means that the temperature around the thermostat would be allowed to float in response to what the test unit was doing to the space.

There was no established method for conducting dynamic testing, and several options were considered. The original plan was to add a space heater to the room with a heating capacity proportionally less than the sensible cooling capacity of the test unit, and allow it to slowly heat up the test chamber until the thermostat reacts. This concept was dropped after deciding that the effect would not be very repeatable due to heat exchange to the environment around the test chamber. Also, the volume of the test chamber was much smaller than that of a house, so the rate of change in the space temperature would be accelerated, and it could not have been used to test the situation where the building load was greater than the capacity of the test unit. The power draw of the heater would also be considerable, given that it would take 7 kW of heat to counter the cooling effect of a 2-ton air conditioner. This method would also not translate easily to dynamic testing in heating mode.

Also considered was the fact that because the chamber's conditioning apparatus already had a resistance heater of sufficient capacity, it might be possible to measure and thus hold its capacity to a fixed level. To accomplish the measurement, an airflow sensing grid was added to the supply duct of the space conditioning apparatus along with humidity and temperature sensors in the supply and return ducts. During the steady-state tests, the measured capacity of the space conditioning system could be compared with that of the test unit when they should be about equal and opposite. That was the intent, but it did not work out very well. One problem was the aforementioned heat exchange with the chamber's exterior environment, as well as the environment around the space conditioning apparatus. But the larger problem was that the airflow through the space conditioning apparatus was three to four times as much as that through the test unit, which meant that for the same capacity level, the change in the temperature and humidity through the conditioning system would be significantly smaller. In most cases, the temperature change was less than 1°F (0.56°C). With the higher uncertainty in the instruments used, there was wide variability in the measured conditioning apparatus capacity. Graphing the space conditioning system capacity against the capacity measured for the test unit under steady-state conditions produced the very definition of a random scatter plot.

Algorithmic Temperature Setpoint Method

The eventual method applied was to algorithmically determine what the temperature of the room should be, based on the current outside room temperature and the measured capacity of the test unit, and use that to adjust the space conditioning apparatus setpoints. The algorithm developed was based on a linear approximation of the building load that is a function of temperature, similar to what is done in the SEER calculation (Figure 25). This simplification ignores solar gains, latent loads, and thermal mass effects. When this is done, the sensible building load reduces to an equation that is only a function of the temperature difference between the room and outside, as follows:

$$\mathrm{BLs} = \dot{\mathrm{Q}}\mathrm{ı} + \mathrm{UA} \times (\mathrm{OAT} - \mathrm{RAT})$$ **Equation 3**

where:

BL_s = Sensible Building Load (tons)

$\dot{Q}_ı$ = Internal heat gains (tons)

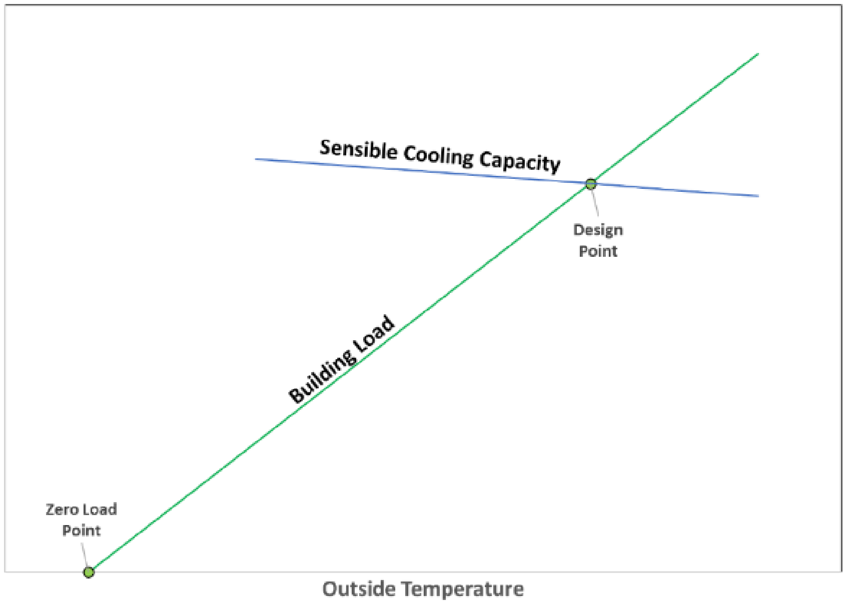
OAT = Outside dry bulb temperature (°F)

RAT = Room or return dry bulb temperature (°F)

UA = Heat transfer coefficient (tons/°F)
(This term is being used as a convenience as this temperature weighting can also contain the load from infiltration and ventilation, and not just from conduction.)

Outside dry bulb temperature (OAT) and return air temperature (RAT) are dynamic measured values, while $\dot{Q}_ı$ and UA are constants that can be derived based on assumed boundary conditions, specifically the range over which the air conditioner can maintain the design room temperature.

Figure 25: Linear Building Load Model



Source: EPRI

The first boundary condition was the outside temperature at which the building load was zero. In this case, the internal gains balanced the heat loss to the outside, as follows:

$$\dot{Q}\iota = \text{UA} \times (\text{RATD} - \text{OAT0}) \qquad \textbf{Equation 4}$$

where:

OAT_0 = Outside dry bulb temperature selected for zero BLs (for example, 65°F [18°C])

RAT_D = Cooling design interior dry bulb temperature (for example, 72°F [22°C])

The second boundary condition is the design point, or the outside temperature under which the air conditioning system when set to its maximum operating mode is just able to meet the sensible cooling load and hold the interior to the cooling design temperature. Above the design point, the system capacity will be less than the building load, and the interior temperature will rise above the design room temperature. Below the design point, the air conditioner will reduce speed or cycle in order to hold the thermostat setpoint. Thus, at the design point, the building load is equal to the air conditioner's sensible cooling capacity at the design conditions:

$$\dot{Q}s_D = \dot{Q}\iota + \text{UA} \times (\text{OATD} - \text{RATD}) \qquad \textbf{Equation 5}$$

where:

OAT_D = Design outside dry bulb temperature corresponding to the design cooling load (for example, 95°F [35°C])

$\dot{Q}s_D$ = Sensible cooling capacity of the subject air conditioning unit at maximum speed under design outside (OAT_D) and inside (RAT_D) conditions (tons)

This sensible cooling capacity can be measured from a specific test at the design conditions, which may vary depending on the climate zone, or the standard rated capacity could be used with an appropriate factor to adjust it to the chosen design conditions and sensible fraction like is done for SEER.

Substituting Equation 4 into Equation 5 produces:

$$\dot{Q}s_D = UA \times [RAT_D - OAT_0 + OAT_D - RAT_D] \qquad \textbf{Equation 6}$$

The RAT_D values cancel out, and the result can be rearranged for UA as:

$$UA = \frac{\dot{Q}s}{OAT_D - OAT_0} \qquad \textbf{Equation 7}$$

Substituting this result and the first boundary condition into Equation 3 produces:

$$BL_s = \frac{\dot{Q}s_D}{OAT_D - OAT_0} \times [RAT_D - OAT_0 + OAT - RAT] \qquad \textbf{Equation 8}$$

which can then be rearranged as:

$$BL_s = \frac{[(OAT - OAT_0) - (RAT - RAT_D)]}{OAT_D - OAT_0} \times \dot{Q}s_D \qquad \textbf{Equation 9}$$

This equation is comparable to the building load equation for calculating SEER given in Equation 2, which uses 95°F (35°C) for OAT_D, 65°F (18°C) for OAT_0, RAT always equal to RAT_D, and $\dot{Q}s_D$ equal to $\dot{Q}_c^{k=2}$ divided by 1.1.

In the laboratory environment, **Equation 9** is used with the current measured values of OAT and RAT to calculate the current building load, with the other factors as constants. The difference between this building load and the currently measured sensible cooling capacity of the test unit is then used to adjust the temperature setpoint for the indoor space. The rate of change for the temperature setpoint is also subject to the thermal capacitance of the air in the space, using the following energy balance:

$$BL_s - \dot{Q}_s = m \times c_p \times \frac{\Delta RAT}{t} \qquad \textbf{Equation 10}$$

where:

m = mass of air in the space

c_p = specific heat of air (0.24 Btu/lb-°F standard)

t = time (in appropriate units)

The equation can be simplified to a form that uses the interior space volume. Applying values of standard air density and specific heat, combining values and converting units gives:

$$\frac{[BL_s - \dot{Q}s](tons) \times 12{,}000\frac{Btu/hr}{ton}}{V(ft^3) \times 0.075\frac{lb}{ft^3} \times 0.24\frac{Btu}{lb{\cdot}°F} \times 3{,}600\frac{sec}{hr}} = \frac{185 \times [BL_s - \dot{Q}s]}{V} = \frac{\Delta RAT}{t}\left(\frac{°F}{sec}\right) \qquad \textbf{Equation 11}$$

The way these equations are used during testing is that on each iteration of the data scan (normally done at a rate of once per second), the values on the left side of the equation are calculated and then multiplied by the iteration period in seconds to produce the incremental change in the temperature setpoint for the space conditioning apparatus. This incremental change has to be slow enough so the space conditioning system can keep up with it and can be slowed further by increasing the value of the volume used. The external resistance on the test unit is also set to follow a constant duct coefficient as described in Equation **1** based on the measured supply airflow.

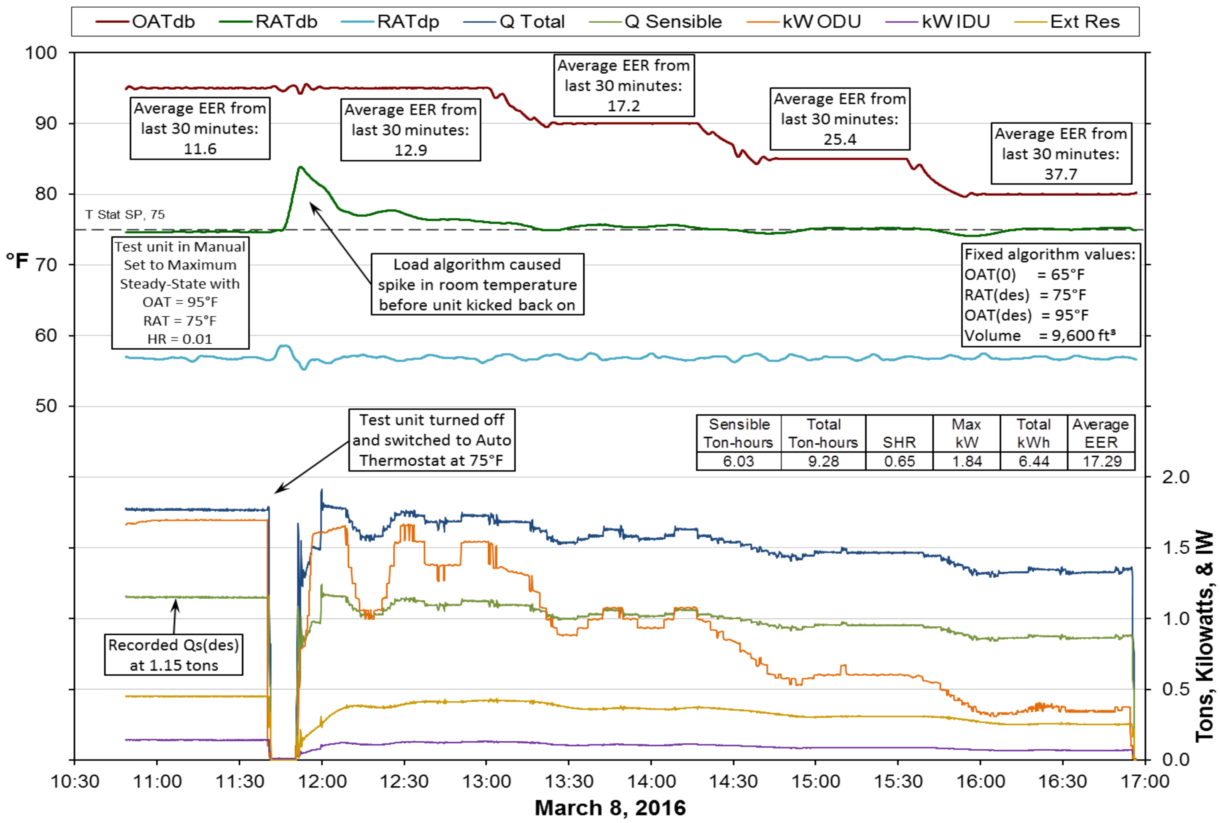
The addition of the difference between the actual RAT and the design RAT in the building load equation is important for the control algorithm simulation, particularly at outside temperatures above design. With it, the building load will decrease as the room temperature rises above the thermostat setpoint until it reaches equilibrium with the available cooling capacity of the air conditioner. Without it, the room temperature setpoint would continue to rise unabated. It should also be noted that the RAT used by the algorithm is measured from a single temperature sensor located near the thermostat and may not exactly match the temperature that the thermostat reads or respond at the same rate.

The question remains as to how to address the latent capacity of the air conditioner. While for its cyclic testing, the AHRI standard requires dry air to make the capacity only sensible, in real-world situations there will almost always be some latent load on the system. The method chosen for these dynamic tests was to set the space conditioning apparatus to maintain a constant humidity ratio or dew point temperature, as that is the easiest method of controlling the room humidity and will cause little variation in the return air dew point sensor. On a psychrometric chart, this would mean that the controlled indoor temperature should just move back and forth along a horizontal line as the temperature adjusts. At the AHRI standard rating condition of 80°Fdb/67°Fwb, the humidity ratio is 0.0112 (60°F [16°C] dew point); and at the 75°Fdb/62°Fwb condition used for the dry climate, the humidity ratio is 0.0089 (54°F [12°C] dew point). Using values such as these will keep the return air close to this design return air condition even as the temperature fluctuates around it. There remain questions about the accuracy of the supply air humidity measurement as it will still undergo considerable changes, and thus affect the measurement of total capacity. This is one of the main reasons why the control algorithm is only working with the measured sensible capacity, which can still be done with reasonable accuracy when fast-response temperature sensors are used. Further research into how the humidity in actual houses behaves in response to air conditioner operations and the outside condition may be used to refine the humidity control in the future.

Dynamic Testing Results in Cooling Mode

The first attempt using the control algorithm began with a steady state test to determine the design maximum sensible cooling load ($\dot{Q}s_D$) and the duct coefficient, with its result shown in Figure 26. The conditions used included an outside temperature of 95°F (35°C), an indoor temperature of 75°F (24°C) with a humidity ratio setpoint of 0.01 (57°Fdp). The test unit was initially controlled from the Ram Monitor program set to "High" Cooling_Test_Mode. After observing steady state operation and recording the base parameters, the test unit was turned off from that program, and its thermostat was then set to "Auto" with a setpoint of 75°F (24°C). Once the algorithm was triggered, the room temperature started to rise following the change in setpoint. Eventually, the thermostat warmed up enough to start the unit ramping up in capacity, finally reaching nearly the same operating condition as the earlier manual test. The room temperature dropped fairly rapidly at first, but didn't quite make it back down to 75°F (24°C), while the outside temperature was still at 95°F (35°C). The outside temperature was then reduced in steps and held constant at 5°F (2.8°C) intervals for one hour each to observe if the unit would reach a steady-state mode of operation. In general, it did reach a stable operating point and maintained the indoor room temperature, with improving efficiency as the outdoor temperature decreased. The compressor power trend also appears to show a number of step changes in power, indicating that it is not fully variable speed but rather has specific speed settings that it steps through.

Figure 26: Dynamic Cooling Test #1

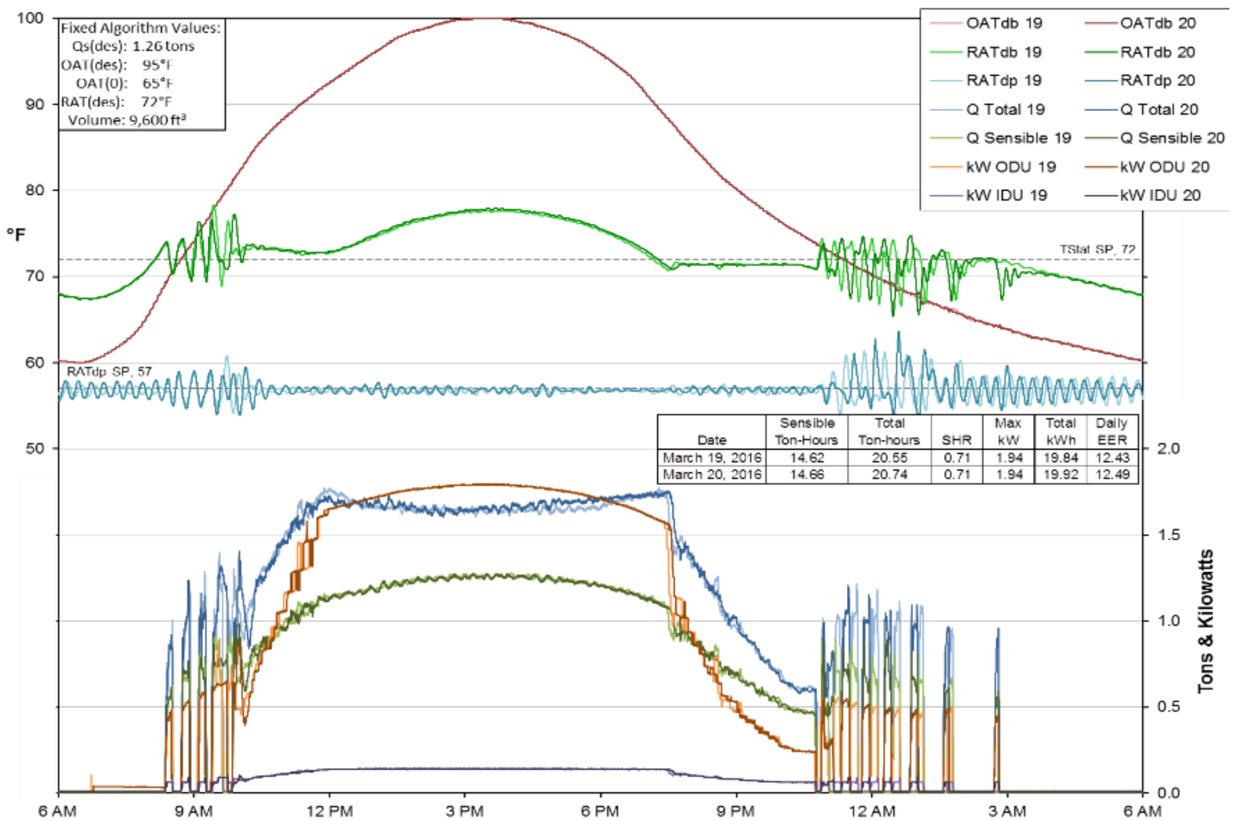


Source: EPRI

The second dynamic test using the algorithm was conducted over the weekend of March 19—20. Instead of step changes in the outdoor temperature like the first test, a script was developed to adjust the outdoor room temperature to follow a real-world 24-hour temperature profile with adjustments every five minutes, which repeated through each test day. The temperature trend was set to run between a low of 60°F (16°C) and a high of 100°F (38°C). For the indoor room temperature control algorithm, the sensible cooling capacity at design conditions was set at 1.26 tons as derived from the A_2 test at 0.45 IW, which was also used to establish the duct coefficient for the indoor unit external resistance. The outdoor temperature was kept at 95°F (35°C) for the design point and 65°F (18°C) for the zero load point, and the design room air temperature and thermostat setpoint were both set to 72°F (22°C). The room was set to maintain a humidity ratio of 0.01 (57°F [14°C] dew point), and the room volume for the algorithm was set to 9,600 cubic feet.

The result from this test was surprising in its repeatability. In Figure 27, the trends for several parameters are overlaid for the two test days. The trends of outside air temperature (OATdb) are identical for the two days since that is the way it was programmed to run. In the morning, as the outside temperature rises above the zero load point and the algorithm-controlled room temperature rises above the thermostat setpoint, the system cycles on at almost the same time on both days and continues to cycle off and on at about the same frequency. At an outside temperature of about 80°F (27°C), the system stops cycling and begins to modulate its output. As the outside temperature continues to rise, the unit maintains a room temperature slightly above the thermostat setpoint, up to about 95°F (35°C) when the building load begins to exceed the available system capacity, and the room temperature rises above the setpoint. Eventually the outside temperature dropped back down enough that the setpoint could again be maintained, although now it was holding to a temperature slightly below the setpoint. As the outside temperature dropped below about 74°F (23°C), the unit again began to cycle on and off, until the need for cooling in the space ceased. The table embedded in the middle of Figure 27 lists the daily summation of sensible and total ton-hours and kilowatt-hours, as well as an average sensible heat ratio (SHR – the ratio of sensible to total cooling), maximum power, and a daily EER, calculated from the integrated total cooling and total power. The values listed for the two days are nearly identical.

Figure 27: Dynamic Cooling Test #2 (with Realistic Temperature Profile)



Source: EPRI

Several similar tests were conducted on the weekends of April 9 – 10 (APPENDIX B: Phase 1 PG&E Additional Figures Figure B-1), May 14 – 15 (Figure B-2), and June 8 – 9 (Figure B-3), and for three days to demonstrate the automatic demand response (ADR) capability (discussed in the next section). Table 12 presents a summary of the daily results from all of the dynamic tests using real-world outdoor temperature profiles; the figures can be found in Appendix B.

Table 12: Summary of Dynamic Tests with 24-Hour Real-World Temperature Profiles

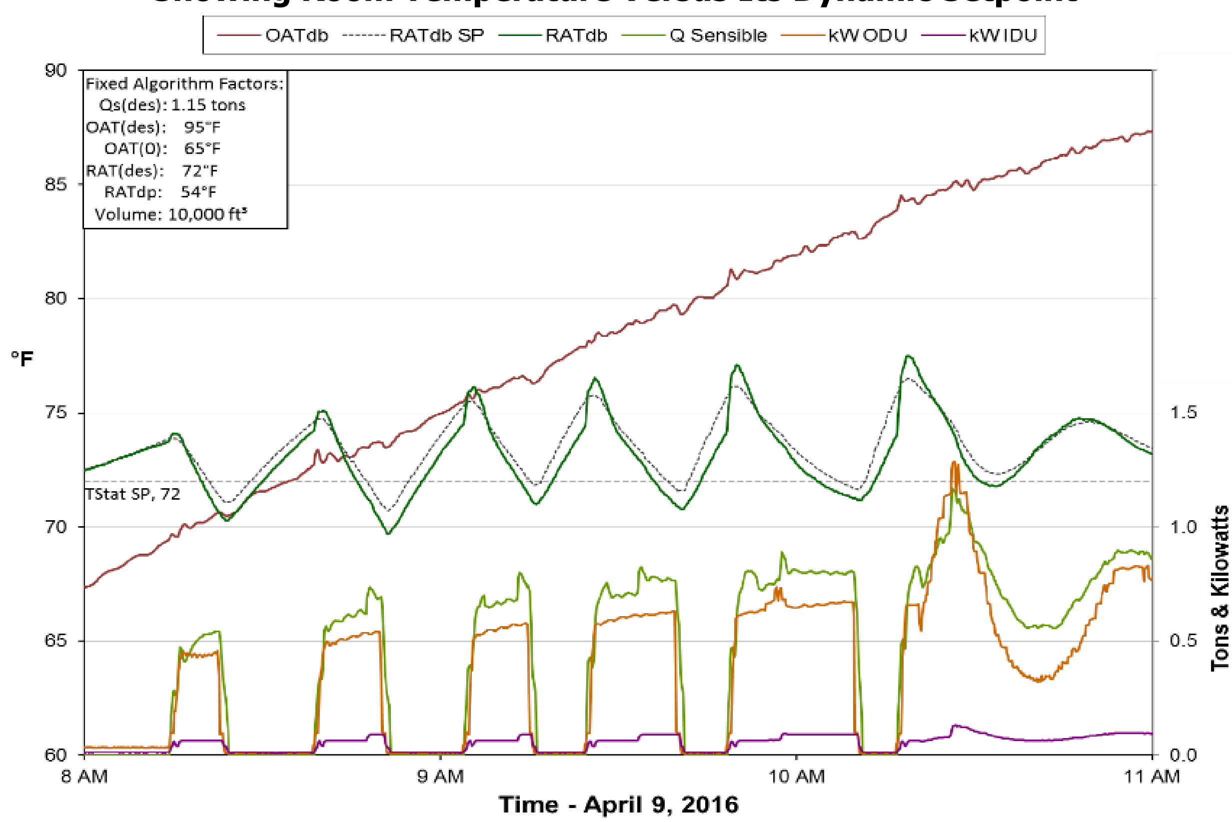
Date	Design Sensible Capacity	House Volume	Return Air Dew Point	Return Air Design & TStat	Outside Air Temperature Zero Load	Outside Air Temperature Design	Outside Air Temperature Daily Low	Outside Air Temperature Daily High	Sensible Ton-Hours	Total Ton-hours	SHR	Max kW	Total kWh	Daily EER
Test Unit Controlled by Thermostat														
March 19	1.26	9,600	57	72	65	95	60	100	14.62	20.55	0.71	1.94	19.84	12.43
March 20									14.66	20.74	0.71	1.94	19.92	12.49
April 9	1.15	10,000	54				62.5	97.5	12.92	17.72	0.73	1.86	16.10	13.21
April 10									12.98	16.94	0.77	1.86	15.50	13.11
May 14	1.20	12,000					65	90	10.94	13.19	0.83	1.09	10.07	15.72
May 15									10.97	13.22	0.83	1.09	10.14	15.65
Test Unit Controlled by Demand Response Computer														
June 8	1.15	10,000	54	72	65	95	62.5	97.5	13.12	16.18	0.81	1.85	17.05	11.38
June 9									11.92*	15.33*	0.78	1.85	16.31*	11.28*
July 21		12,000							14.51	20.11	0.72	1.84	20.20	11.95
July 23									14.55	19.65	0.74	1.84	20.19	11.68
July 24									14.47	19.58	0.74	1.84	20.41	11.51

*** Not a complete day – unit disabled at 9:00 p.m.**

Source: EPRI

Figure 28 shows a detail of three hours during the start of system cycling from the test on April 9 and is intended to show how well the measured room temperature tracked the setpoint as determined by the algorithm.

Figure 28: Detail of Dynamic Test #6
Showing Room Temperature Versus Its Dynamic Setpoint



Source: EPRI

The setpoint is shown as a dotted line, with the room temperature in green lagging slightly behind it. As mentioned previously, the actual temperature measured at the thermostat and used to control the operation of the test unit may lag further behind this temperature measurement. One curious effect in this plot is that when the test unit cycles on, there is a sudden rise in the room temperature, possibly as the result of changing air currents in the space. (The data logging interval in this test is every 10 seconds.)

The dynamic tests using the real-world outdoor temperature profiles conducted in June and July are different from the previous ones in that the operation of the test unit was controlled by the ADR computer rather than the thermostat, which apparently uses a different control algorithm. There were also a few anomalies in the tests. On June 8 at about 11 PM, it turned the indoor blower on alone for about two hours before activating the outdoor unit. During this period, there was some sensible cooling as the condensed water that had built up on the coil or retained in the condensate pan was evaporated back into the room, and the coil behaved in the same manner as an evaporative cooler. In the June 9 test at 9 PM, the computer stopped operating the system, possibly as the result of the computer being accessed remotely. Source: PG&E

Figure B-4 in Appendix B shows the operating trends from the tests on April 10 and July 21 demonstrating the difference in how the thermostat and the ADR computer respectively control the test unit. The trends from the tests using the ADR computer are discussed in more detail in the next section.

In addition to the several tests with real-world outside temperature profiles, there were also more tests conducted where the outdoor temperature was held constant for an extended period. The intent of these tests is to determine if the test unit will settle out at some fixed speed or continue to vary, what its average efficiency is as a function of the outside temperature, and whether it makes a difference if the temperature hold is approached from above or below. In Figure 29, the temperature was varied between 80ºF (27°C) and 100ºF (38°C) in 5°F (2.8°C) steps, with each held for two hours. For each step, the EER from the last half hour was calculated and added to the figure. There are certain periods where the test unit did continue to hunt for a stable operating point and oscillated between some of its speed settings.

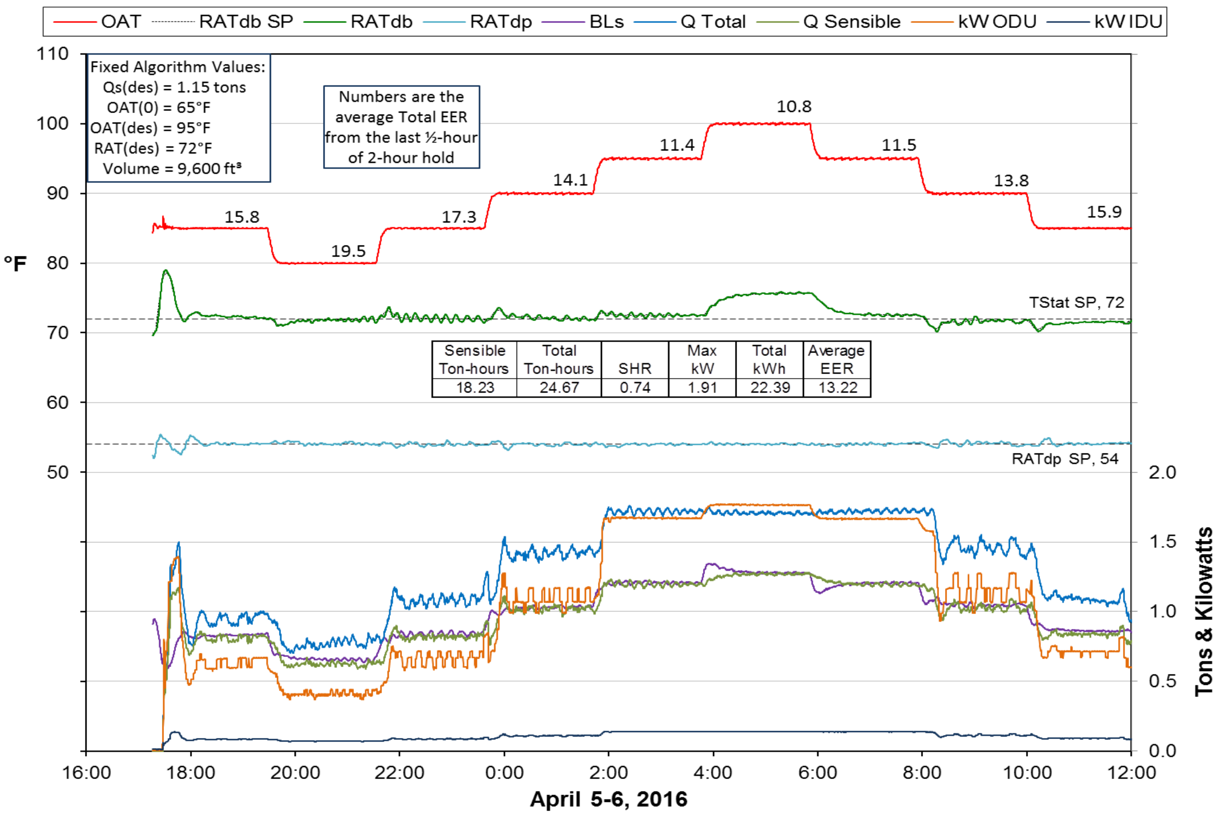
Additional tests using this technique were conducted on March 29-30 (Figure B-9), April 6-7 (Figure B-10), April 15-17 (Figure B-11), and May 27-28 (Figure B-12), with the figures located in Appendix B. The last two of these used outdoor temperature setpoints derived from a draft revision to a Canadian Standards Association (CSA) test standard, which is also considering the application of this dynamic algorithm. For the last of these, the temperature setpoint steps were fit to a 24-hour repeating cycle so that the test results from subsequent days could be overlaid. As with the test in Figure B-3, the test unit was controlled via the ADR computer rather than directly through the thermostat, so its operation was different and appears to favor cycling over speed control. The operational pattern was still very repeatable.

Automatic Demand Response (ADR) Testing

Automatic demand response, such as implemented through PG&E's SmartAC™ program, when applied to fixed speed systems, usually involves the cycling off of the compressor in the outdoor unit for a short period. Events are triggered such that the compressor will be off for a maximum of 15 minutes in any half hour period and totaling no more than 2½-hours on an event day between the hours of

11:30 AM and 6:00 PM. If the thermostat is still calling for cooling, the indoor blower will continue to run to circulate air through the house. This circulating air can provide some sensible cooling effect from just the air movement or by re-evaporating condensate collected on the coil or in the pan but can also act to heat up the space faster than normal due to the fan motor heat or if the duct work passes through a hot region like an attic. If the space temperature rise becomes noticeable or possibly unbearable to the customer, they may act to override the event and not provide the needed demand reduction.

Figure 29: Dynamic Cooling Test #4 (with Outside Temperature Steps)



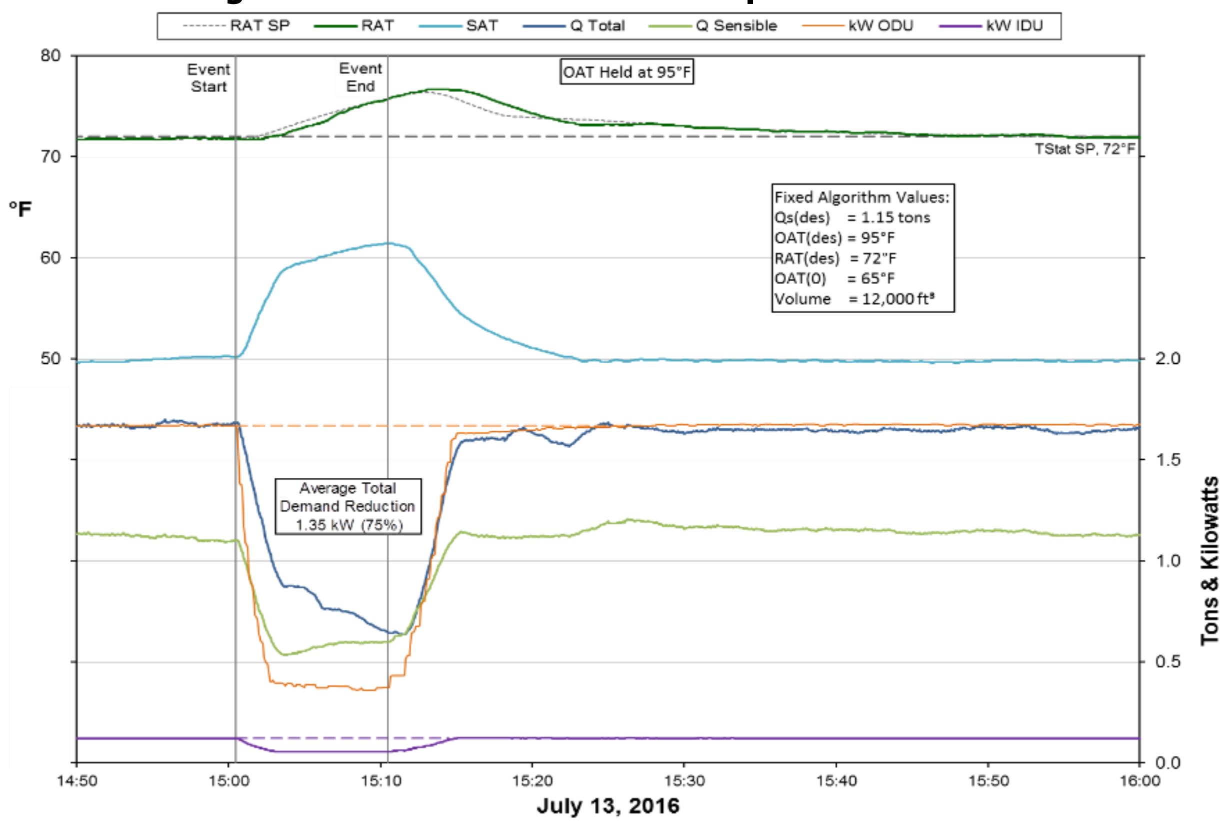
Sensible Ton-hours	Total Ton-hours	SHR	Max kW	Total kWh	Average EER
18.23	24.67	0.74	1.91	22.39	13.22

Source: EPRI

Variable capacity systems such as the test unit provide an alternative mode of operation in that the capacity of the compressor could be reduced but not necessarily turned off, and thus still provide some level of cooling to prevent the space temperature change from becoming noticeable. The implementation of the demand response capability of the test unit is under development and not currently incorporated into its thermostat. Instead, a computer was provided that would respond to simulated events sent through the Internet to change the operation of the test unit. The computer communicated with its own indoor and outdoor temperature sensors, although these seemed to have a slower response to room temperature changes than the thermostat. A toggle switch was provided to shift system control of the test unit from the thermostat to the computer. There are currently few options provided for the events, with a file on the computer containing settings for the speed that the compressor will reduce to during an event, and the upper and lower temperature limits that will cancel an event if they are exceeded. The event signal sent to the computer only contains a start time and duration, with nothing yet for the severity of the event or pricing signals.

For the initial test with an ADR event, the operation of the test unit was run with a fixed environmental condition in the outdoor room (95°F [35°C]), and the indoor room temperature was controlled by the algorithmic method using typical settings until a stable temperature condition was achieved with the system at maximum speed. An event of 10-minute duration was then triggered with the compressor speed ADR setpoint at its default 30 percent. The response from this event is shown in Figure 30. (The total cooling capacity – "Q Total" – is still shown even though it is likely inaccurate during the transition due to the slow response of the supply humidity measurement.) At the beginning of the event, the power draw of the both the outdoor and indoor units were reduced, although not immediately, taking about 2 minutes to ramp down to its new operating level. At the end of the event, the transition back to full power was also gradual, taking about four minutes. The average total power reduction was 1.35 kW, or 75 percent of the total power before the event. The indoor room control algorithm allowed the room temperature to rise by about 5°F (2.8°C) from the event with the reduced cooling capacity, but this was quickly recovered once the system returned to full capacity.

Figure 30: Automatic Demand Response Test Detail



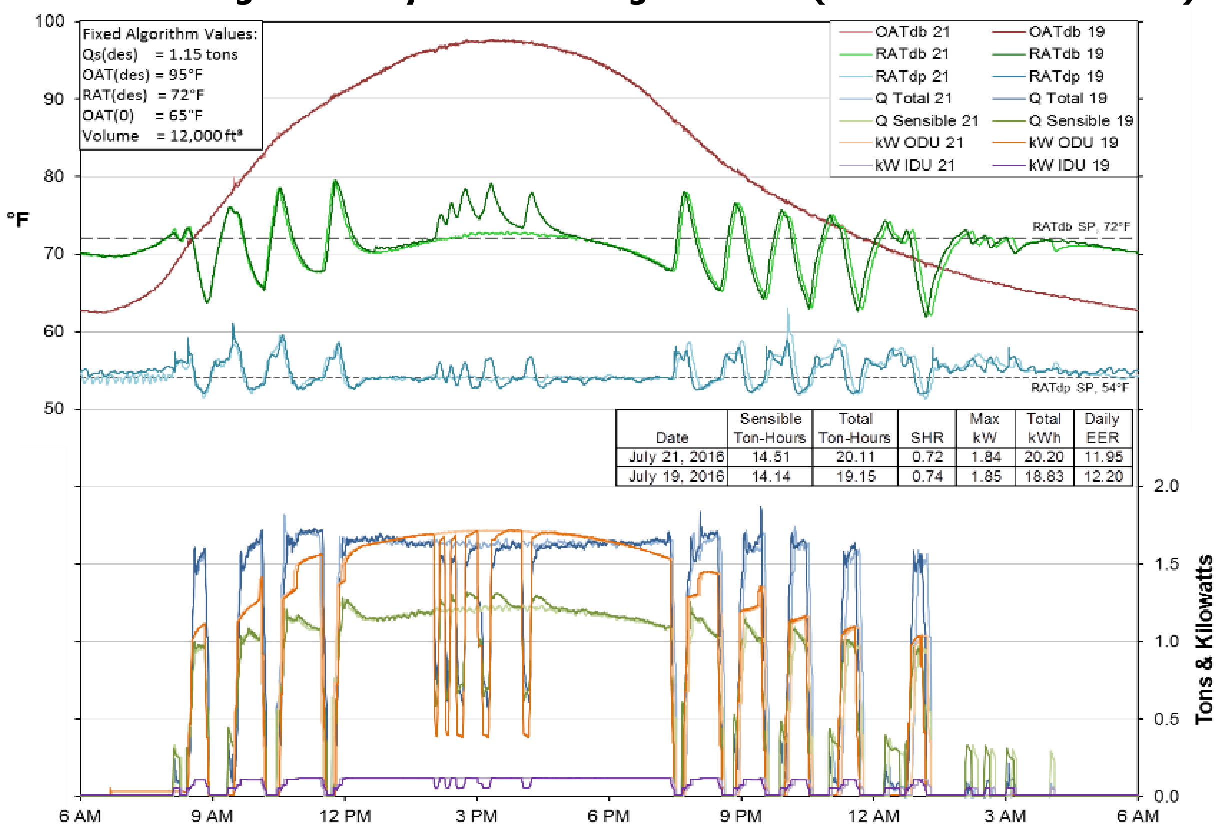
Source: EPRI

The subsequent evaluations of ADR events followed the same pattern as the previous 24-hour cooling mode dynamic tests with a real-world outdoor temperature profile. The difference being that one day would have no events while a second would have multiple events simulating an actual event day. The successful test runs were all done using an outdoor temperature profile having a daily low of 62.5°F (16.7°C) and a daily high of 97.5°F (36.7°C). For the event days, the events were scheduled as follows:

- 2:00 PM: 5 minutes
- 2:15 PM: 5 minutes
- 2:30 PM: 10 minutes
- 3:00 PM: 15 minutes
- 4:00 PM: 10 minutes

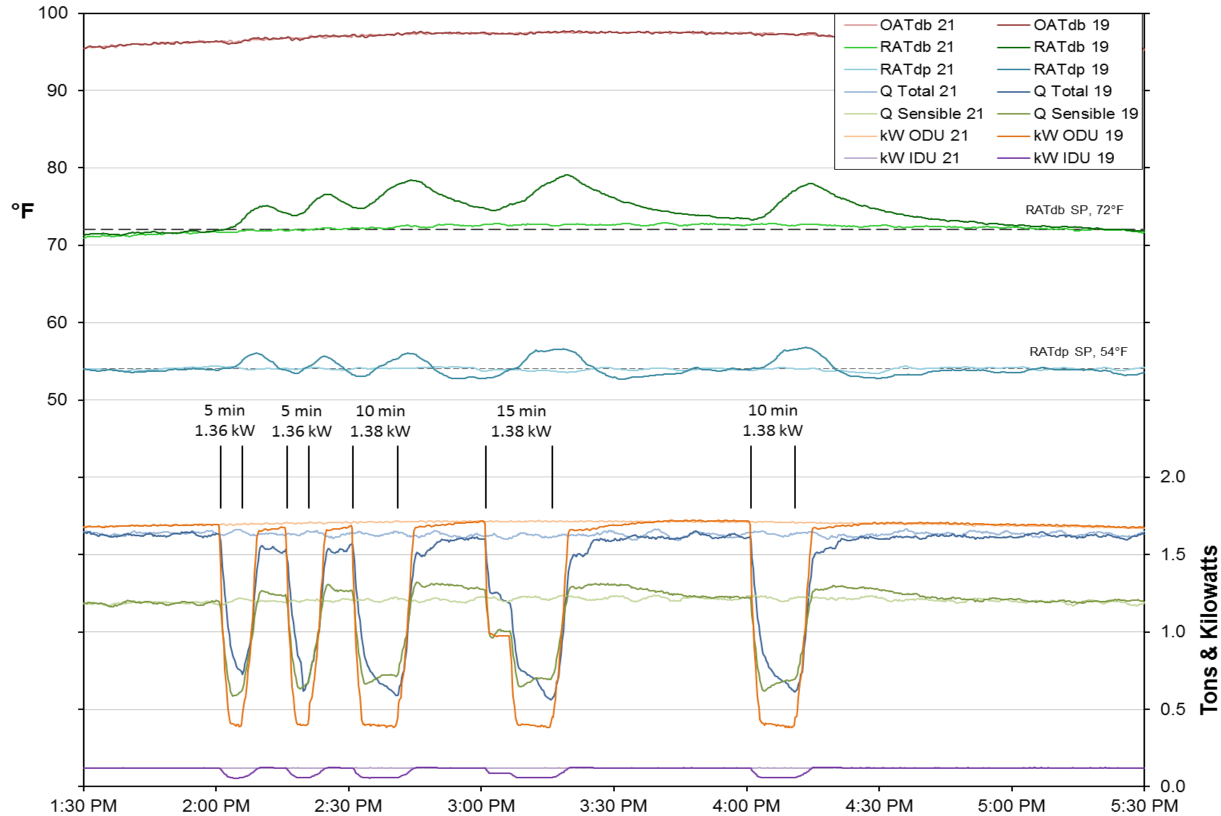
Three different speed reduction settings were applied for the event days: 30 percent (default), 0 percent (to approach the response of a fixed speed system), and 50 percent. The first of these event days is shown in Figure 31 and Figure 32; the second of which shows a detail of just the four-hour period around when the ADR events were occurring. In both of these, the trends from the event day (July 19) are overlaid on the trends from the non-event day (July 21). As before, the trends outside of the ADR events are almost exactly the same on the two days, which acts to highlight the effect produced by the events and gives a clearer picture of the differences.

Figure 31: Dynamic Cooling Test #11 (with 30% ADR Events)



Source: EPRI

Figure 32: Detail of Dynamic Cooling Test #11 (with 30% ADR Events)



Source: EPRI

The trends from the other two events with different capacity reduction setpoints are included in Appendix B. Figure B-5 and Figure B-6 show the test with a 0 percent load setpoint, and Figure B-7 and Figure B-8 show the test with the 50 percent load setpoint; again with the first figures showing the full 24-hour cycle and the second figures showing the four-hour window containing the ADR events. Table 13 provides some general statistics on the results listing how much of a demand reduction was achieved from each of the three ADR load setpoints, and how much the sensible capacity of the system was reduced, on average. (Total capacity was not included due to concerns about its accuracy.) The 0 percent ADR event did not achieve 100 percent demand reduction because it did not completely turn off the indoor blower, although the compressor capacity was reduced to zero.

Table 13: Summary of Demand Events from 24-hour Dynamic Tests

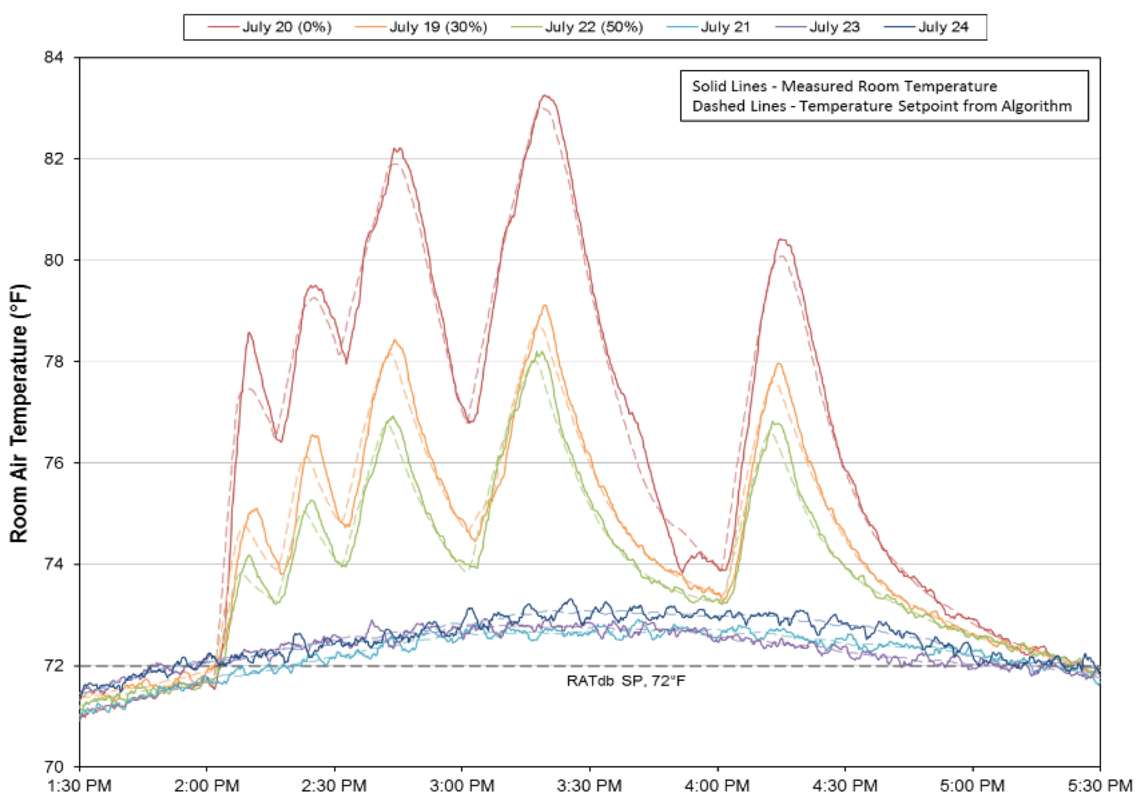
ADR Speed Setting	Achieved Demand Reduction	Sensible Capacity Reduction
50%	1.19 kW (65%)	38%
30%	1.37 kW (75%)	49%
0%	1.77 kW (95%)	58%

Source: EPRI

An early concern about conducting ADR tests in this manner was that it might show that the overall daily energy consumption of the system would be greater with the events than without, since it might need to run for a longer time in order to catch up on the loss of temperature control during the hottest part of the day. This concern proved to be unfounded, and all of the ADR event days actually saved energy in relation to the days without the events, in addition to providing needed demand reduction. One reason for this is that the outside temperature profile used put the system into steady-state operation at maximum capacity throughout the period covered by the events, both with and without them. In addition, as the return air temperature rises, it also acts to increase the capacity of the test unit due to the larger temperature difference between the return air and the evaporating refrigerant, until it drives the room temperature down to its previous equilibrium state.

The advantage of using the same indoor temperature control algorithm settings and the same outside temperature profile is that it can demonstrate the effect of the ADR speed setting on the indoor temperature. Figure 33 shows the trends of just the indoor room temperature measurement from all of the test days in the four-hour detail window (same as the RATdb trends shown in Figure 32, Figure B-6, and Figure B-8). Added to those trends are the dynamic room temperature setpoints derived from the control algorithm (shown as dashed lines) to show how well the temperature was actually able to track the setpoint, which was normally quite well except during the sudden transitions in the test unit's operating mode. The trends confirm that the larger the reduction in system capacity, the greater the room temperature rise from what it would have been able to hold. To reemphasize, the room temperature sensor for this measurement responds faster to temperature changes than either the thermostat or the remote sensor for the ADR computer.

Figure 33: Detail of Room Temperature Trends from ADR Events



Source: EPRI

Heating Mode Testing

Plan for Standard Tests in Heating Mode

As with cooling, the conditions applied for the heating mode tests included those used for rating purposes in AHRI Standard 210/240-2008, Table 14 (for variable-speed compressor systems), plus some additional outdoor conditions to produce a performance map. Once again, the Ram Monitor program was used to put the system into a fixed operating mode, except now using the Heating_Test_Mode parameter to operate the unit as a heat pump. The planned performance tests are listed in Table 14, including their AHRI designation. The $H2_2$ test and $H2_1$ tests are optional for a variable speed system as they can be estimated from other test results. ($H2_1$ and Maximum are not used in the calculation of HSPF.)

Table 14: Planned Heating Mode Standard Tests

Test Description	Air Entering Indoor Unit Temperature	Air Entering Outdoor Unit Temperature		Compressor Speed and Heating Air Volume Rate
	Dry-Bulb (°F)	Dry-Bulb (°F)	Wet-Bulb (°F)	
AHRI $H0_1$ Test	70	62	56.5	Minimum
AHRI $H0C_1$ Test (Cyclic)	70	62	56.5	Minimum
AHRI $H1_2$ Test	70	47	43	Maximum
AHRI $H1_1$ Test	70	47	43	Minimum
AHRI $H2_2$ Test (Optional)	70	35	33	Maximum
AHRI $H2_V$ Test	70	35	33	Intermediate
AHRI $H2_1$ Test (Optional)	70	35	33	Minimum
AHRI $H3_2$ Test	70	17	15	Maximum
AHRI Maximum Conditions	80	75	65	Maximum
Performance Mapping	70	65, 55, 45, 35*, 25, 15		Maximum & Minimum

*** Mapping test condition already included in standard tests**

Source: EPRI

For the performance mapping tests, the decision was made to try to achieve a constant 70 percent relative humidity in the outdoor room for each of the dry bulb temperature steps. This is consistent with the H0, H1, and H3 test conditions, but not the H2 or Maximum tests.

The outdoor space conditioning apparatus was actually able to achieve temperatures into the low 40s through continuous recirculation of the chamber air, and the secondary refrigeration system was not needed. The difficulty with performing any of the heating mode tests at outside temperatures below about 45°F (7°C) is the tendency for ice to form on the evaporator coils. This problem is not just experienced by the sub-chamber refrigeration coil but can also happen on the coil of the outdoor chamber space conditioning system. As ice forms, it cuts off airflow through the coil, and will make it impossible to properly control the space temperature. It becomes necessary then to periodically turn off the cooling systems and defrost their coils. For the outdoor room space conditioning system, this can be done relatively quickly by toggling its reversing valve to put it into heat pump mode just long enough to melt the formed ice. For the sub-chamber refrigeration coil, defrosting required spraying the coil with hot water.

The test unit itself also has a built-in defrost cycle when operating as a heat pump. The frequency of these defrosts can be set in the thermostat to a maximum interval of two hours but could not be disabled. Operating the heat pump via the Ram Monitor program does not override these periodic defrosts, and this leads to extended testing in order to capture a 30-minute stable period. Compounding this is that the system can take a long time to return to steady-state operation following a defrost cycle, and as much as an hour is needed for recovery when operating at minimum speed.

Heating Mode Standard Test Results

The start of the heating mode testing was delayed because of a defective part in the outdoor unit that prevented its operation in heating mode. The actuator that controlled the electronic expansion valve (EXV) had become loose and was not actually moving the valve. A factory-certified technician diagnosed and repaired the problem on March 30. Most of the standard tests were completed between April 5 and April 22, with some repeats happening later interspersed with the dynamic tests.

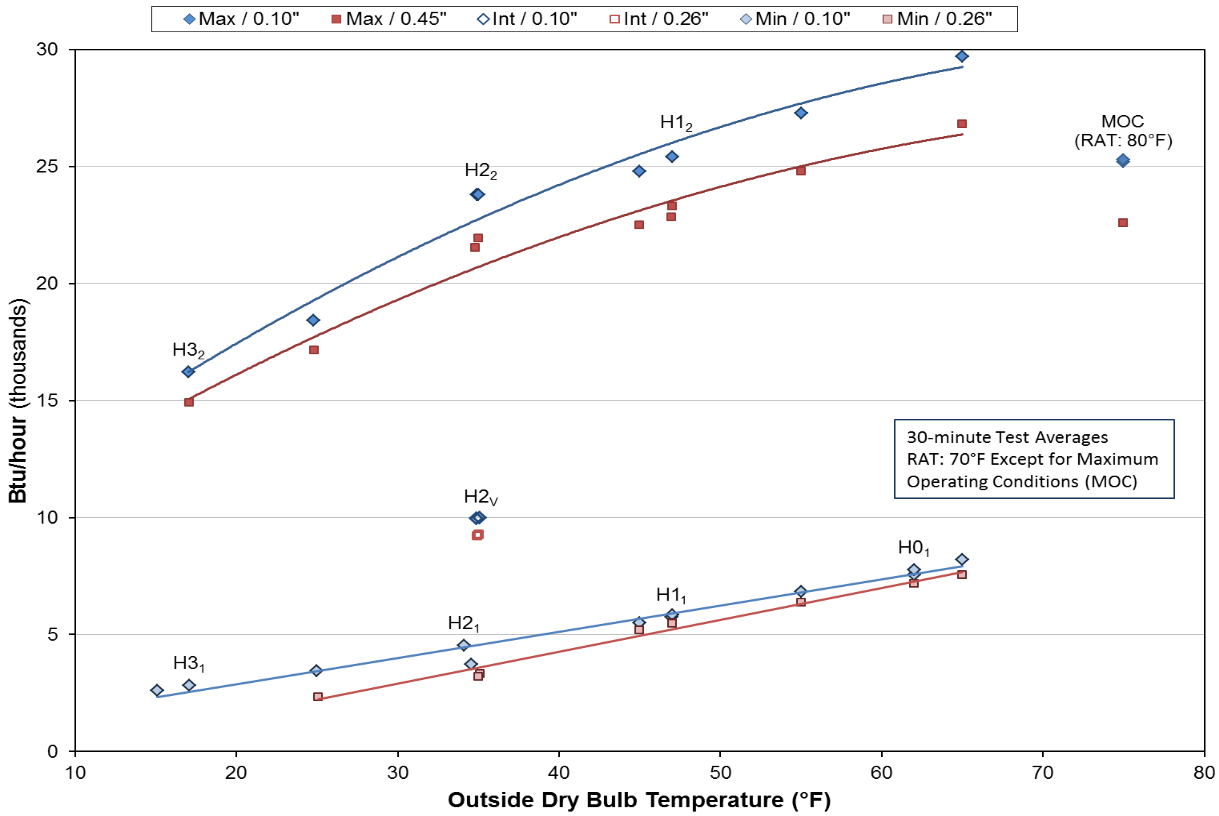
As was done with the cooling mode tests, the results from the heating mode tests are presented in three figures and a table. By convention, heating mode capacity is given in thousand Btu per hour (kBtu/hr) and the efficiency is given as a dimensionless coefficient of performance (COP). Table 15 contains the summary table listing the results from all of the standard heating mode tests. As with the cooling mode testing, the measured supply airflow was lower than expected, but higher than in cooling. The ducting on the outdoor unit was not connected during these tests, so its airflow was not measured.

Figure 34 shows the trends of heating mode capacity as a function of the outside temperature. While the trends for cooling mode included four different cases, here there are only two corresponding to the different supply external resistance levels. One feature that stands out is the result for the optional $H2_2$ test at maximum speed, which is slightly above the trend drawn through the other test points. This test was actually repeated to confirm this finding, and it is thought that the offset has to do with the test being conducted at a higher outdoor room relative humidity (82 percent) versus being near 70 percent for the other tests. This also affects the Maximum Operating Condition test, which is conducted at a relative humidity of about 59 percent as well as a return air temperature of 80°F (27°C) versus 70°F (21°C), and is why it is not included in the trend line. Higher humidity levels in the outdoor air provides more potential latent heat to be captured through condensation and freezing.

Compared with the cooling mode tests in Figure 20, the trends of total power in heating mode shown in Figure 35 are nearly flat as a function of the outside temperature. Surprisingly, there actually appears to be a slight downward trend in power at maximum speed with decreasing outside temperature, even with the compression ratio increasing. This is primarily the effect of the refrigerant at the compressor suction having a very low density, and this reduces the mass flow through the compressor and its power requirement.

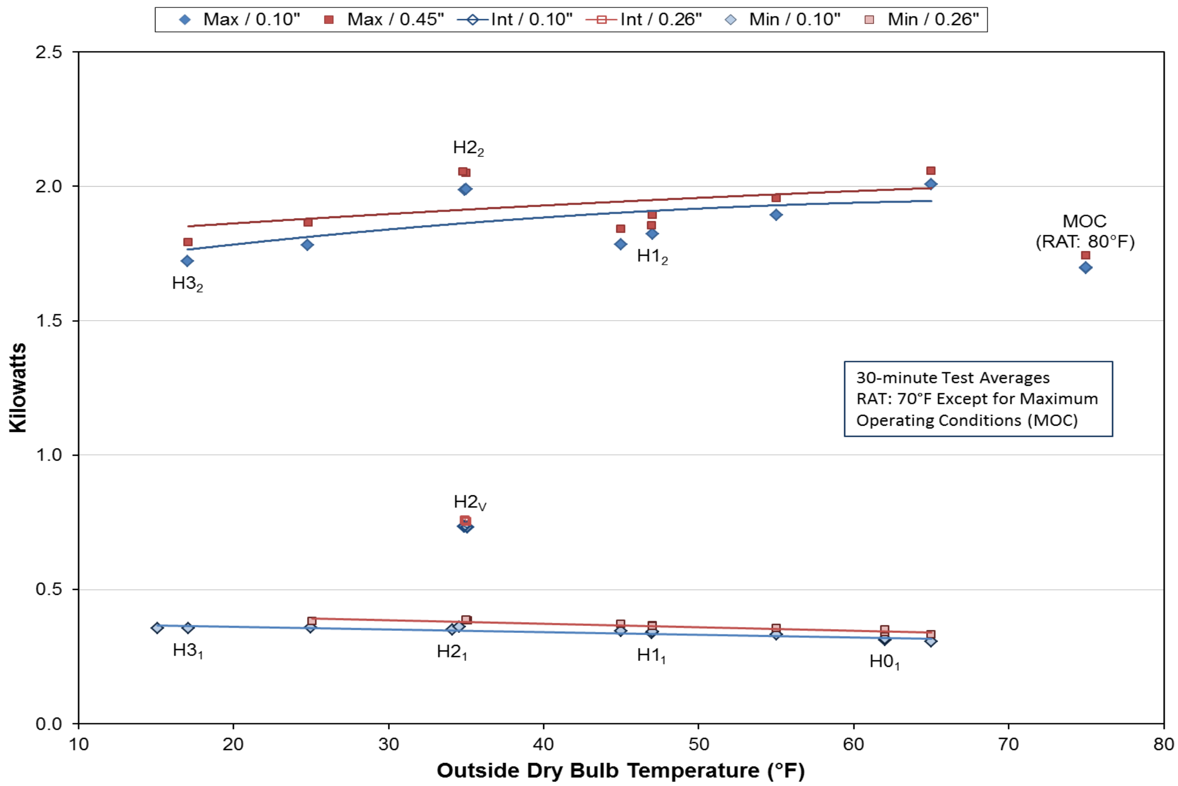
The trends of COP in Figure 36 show a crossover point between maximum and minimum speed, where the maximum speed efficiency is higher at colder temperatures, and the minimum speed efficiency is higher at warmer temperatures. This is important since those are the speeds that the system will normally be operating at under those conditions to meet the building load.

Figure 34: Heating Mode Mapping – Capacity



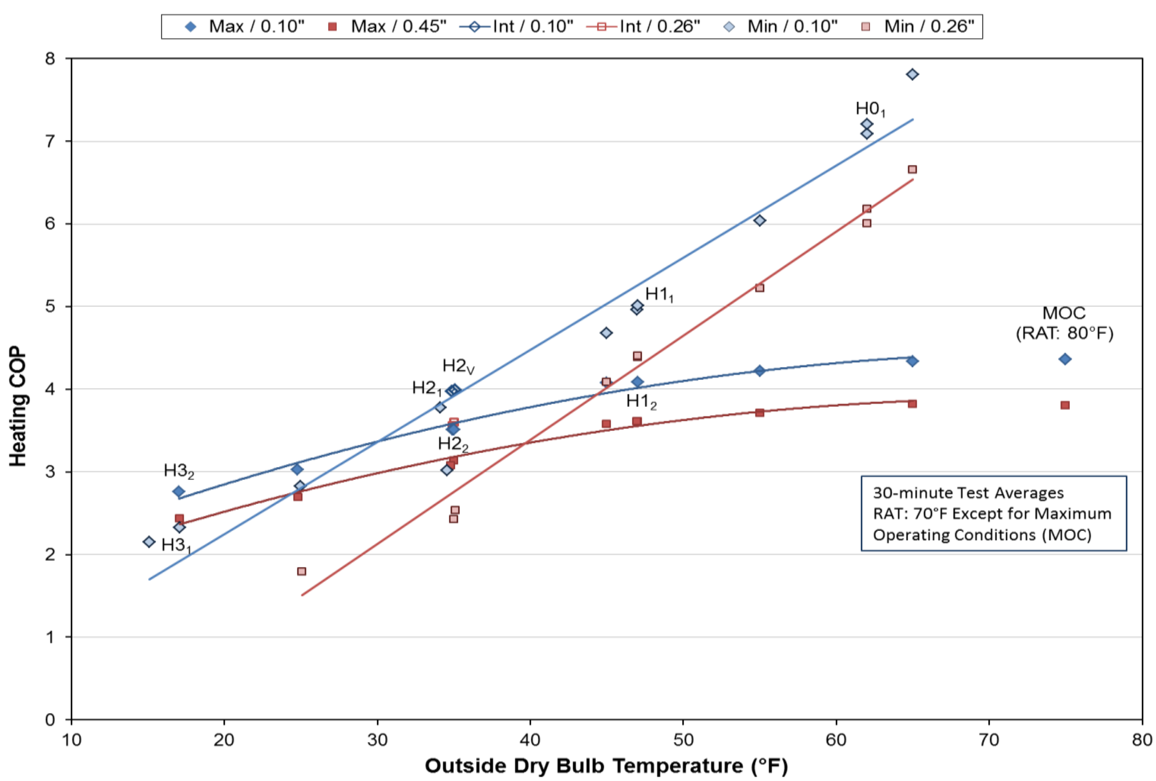
Source: EPRI

Figure 35: Heating Mode Mapping – Total Power



Source: EPRI

Figure 36: Heating Mode Mapping – Efficiency (COP)



Source: EPRI

Table 15: Heating Mode Standard Test Results

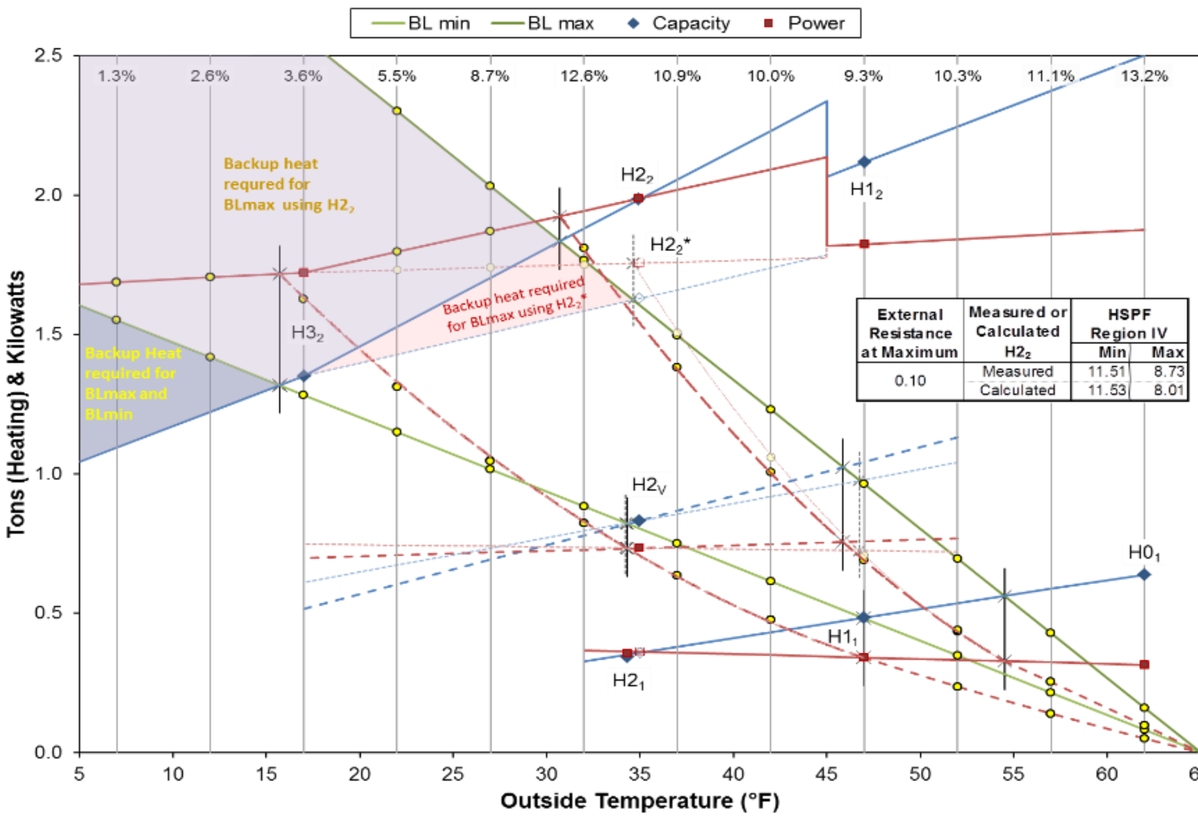
Test Description	Compressor Speed and Cooling Air Volume Rate	Outside Air Tdb (°F)	Outside Air Twb (°F)	Return Air Tdb (°F)	Ext. Res. (IW)	Heating Capacity (kBtu/hr)	Power (kW) IDU	Power (kW) ODU	Power (kW) Total	COP	Indoor Blower Airflow (CFM)	Indoor Blower Speed (RPM)	Outdoor Fan Speed (RPM)
$H0_1$	Minimum	62	56.5	70	0.10	7.66	0.050	0.26	0.31	7.15	528	447	600
					0.28	7.17	0.080	0.27	0.35	6.09	491	597	600
$H0C_1$	Minimum (Cyclic)	*62*	*56.5*	*70*	0.10	1.22			0.113	3.16	Cyclic		
					0.28	1.01			0.118	2.52			
$H1_2$	Maximum	47	43	70	0.10	25.41	0.074	1.75	1.82	4.08	706	516	720
					0.45	23.07	0.149	1.73	1.87	3.61	660	751	720
$H1_1$	Minimum	47	43	70	0.10	5.79	0.050	0.29	0.34	4.99	528	449	600
					0.26	5.48	0.076	0.29	0.37	4.40	494	583	600
$H2_2$	Maximum	35	34	70	0.10	23.81	0.075	1.91	1.99	3.51	713	519	720
					0.45	21.73	0.149	1.90	2.05	3.11	657	753	720
$H2_V$	Intermediate	35	34	70	0.10	9.98	0.050	0.68	0.73	3.99	536	451	600
					0.26	9.25	0.077	0.68	0.76	3.58	497	586	600
$H2_1$	Minimum	35	33	70	0.10	4.12	0.050	0.31	0.36	3.39	527	450	600
					0.26	3.26	0.076	0.31	0.38	2.48	488	583	600
$H3_2$	Maximum	17	16	70	0.10	16.21	0.075	1.65	1.72	2.76	701	518	720
					0.45	14.90	0.149	1.64	1.79	2.44	650	753	720
Max	Maximum	75	65	80	0.10	24.74	0.062	1.59	1.65	4.40	640	488	720
					0.45	22.59	0.133	1.61	1.74	3.80	599	740	720
Performance Mapping	Maximum	65	59	70	0.10	29.72	0.075	1.93	2.01	4.34	723	520	720
					0.45	26.82	0.150	1.91	2.06	3.82	667	754	720
		55	50	70	0.10	27.30	0.076	1.82	1.89	4.22	722	521	720
					0.45	24.79	0.151	1.81	1.96	3.71	665	755	720
		45	42	70	0.10	24.81	0.075	1.71	1.78	4.08	716	520	720
					0.45	22.50	0.149	1.69	1.84	3.58	659	754	720
		35	34	70	0.10	23.81	0.075	1.91	1.99	3.51	713	519	720
					0.45	21.73	0.149	1.90	2.05	3.11	657	753	720
		25	24	70	0.10	18.43	0.075	1.71	1.78	3.03	707	519	720
					0.45	17.14	0.149	1.72	1.86	2.69	651	752	720
		17	16	70	0.10	16.21	0.075	1.65	1.72	2.76	701	518	720
					0.45	14.90	0.149	1.64	1.79	2.44	650	753	720
	Minimum	65	60	70	0.10	8.19	0.050	0.26	0.31	7.80	535	451	600
					0.26	7.54	0.077	0.25	0.33	6.66	495	586	600
		55	50	70	0.10	6.84	0.050	0.28	0.33	6.04	533	452	600
					0.26	6.36	0.077	0.28	0.36	5.22	494	587	600
		45	42	70	0.10	5.50	0.050	0.29	0.34	4.68	530	451	600
					0.26	5.20	0.077	0.30	0.37	4.09	492	586	600
		35	33	70	0.10	4.12	0.050	0.31	0.36	3.39	527	450	600
					0.26	3.26	0.076	0.31	0.38	2.48	488	583	600
		25	24	70	0.10	3.45	0.050	0.31	0.36	2.83	528	450	600
					0.26	2.33	0.076	0.30	0.38	1.79	486	582	600
		15	14	70	0.10	2.60	0.050	0.30	0.35	2.15	528	451	600

Source: EPRI

Calculation of Heating Season Performance Factor

The calculation of heating season performance factor (HSPF) is done in much the same manner as for seasonal energy efficiency ratio (SEER), but there are a number of multipliers that expand the number of calculations and the results. As before, there is a multiplier of two for the two different external resistance values at which the tests were conducted. There is another factor of two in that the optional $H2_2$ test results could be used, or a mathematical approximation of this test can be obtained from a calculation based on the values from the $H1_2$ and $H3_2$ tests. Both of these situations were looked at, with the calculated approximation indicated by an asterisk (*) in the results. The main reason for the proliferation of HSPF calculations is that the standards call for it to be calculated for six "Regions" with different temperature bin weighting factors; and in each region, two building load lines referenced to region-specific outdoor design temperatures. Thus, instead of two SEER calculations, there are forty-eight HSPF calculations. Reporting of HSPF is normally just done for Region IV at a single external resistance, so there would be just the range between the minimum and maximum load lines provided. Figure 37 is a very complicated chart to describe how four values of HSPF are derived for one particular region (Region IV) using this very complicated calculation method. Another wrinkle in this calculation from what is done for SEER is that the slope of the capacity and power at maximum speed within the range of 17°F (–8°C) to 45°F (7°C) is to be derived from the $H2_2$ (or $H2_2$*) and $H3_2$ tests, and outside this range the slopes are derived from the $H1_2$ and $H3_2$ tests. (The trends determined using the calculated $H2_2$* test values are shown as thin dotted lines.) The minimum building load line is based on the capacity of the system from the $H1_2$ test rounded off to a "standard" heating value and referenced to a region-specific design outdoor temperature (5°F [–15°C] for Region IV), and BLmax is about twice the slope of Blmin, also subject to rounding.

Figure 37: Test Data Used to Derive HSPF for Region IV (at 0.10 IW)



Source: EPRI

Once again, the intersections between the load lines and the available capacity trends for the minimum and maximum speeds, and the pseudo-trend for the intermediate speed are used to determine a second order curve fit for the power demand at intermediate speeds. (Since there are two alternative slopes for the maximum speed capacity and power trends, there are also two alternative slopes for the pseudo-intermediate speed capacity and power trends, and two intermediate speed power curves.)

In the SEER calculation, when the building load exceeded the available cooling at maximum speed, the capacity and power followed just what the system could provide. For HSPF, when the heating load exceeds the available heat pump capacity at maximum speed, electric resistance heat is assumed to pick up the difference.[2] The regions where the system would be using some form of backup heat are shaded in the upper left corner of the figure.

Four HSPF values are calculated, from this demonstration analysis chart, based off of the two load lines and the two values of $H2_2$ for one Region. Table 16 and Figure 38 present the results from all 48 combinations of variables. Note that all of the results for each of the two external resistance levels are derived from the same set of measurments, but just with different weighting in the analysis.

Table 16: Multiple HSPF Values Derived from the Same Two Test Data Sets (in Btu/Wh)

External Resistance at Maximum	Measured or Calculated $H2_2$	Region I		Region II		Region III		Region IV		Region V		Region VI	
		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max
0.10	Measured	16.14	16.08	15.14	14.18	13.89	11.80	11.51	8.73	10.09	6.72	15.94	15.27
	Calculated	16.18	14.97	15.18	13.13	13.91	10.65	11.53	8.01	9.25	6.41	15.94	14.12
0.45	Measured	14.29	14.50	13.46	12.38	12.33	10.91	10.20	7.78	8.29	6.64	14.13	13.83
	Calculated	14.31	13.90	13.47	11.20	12.30	10.01	10.18	7.10	8.32	6.38	14.10	13.12

Source: EPRI

Plan for Dynamic Testing in Heating Mode

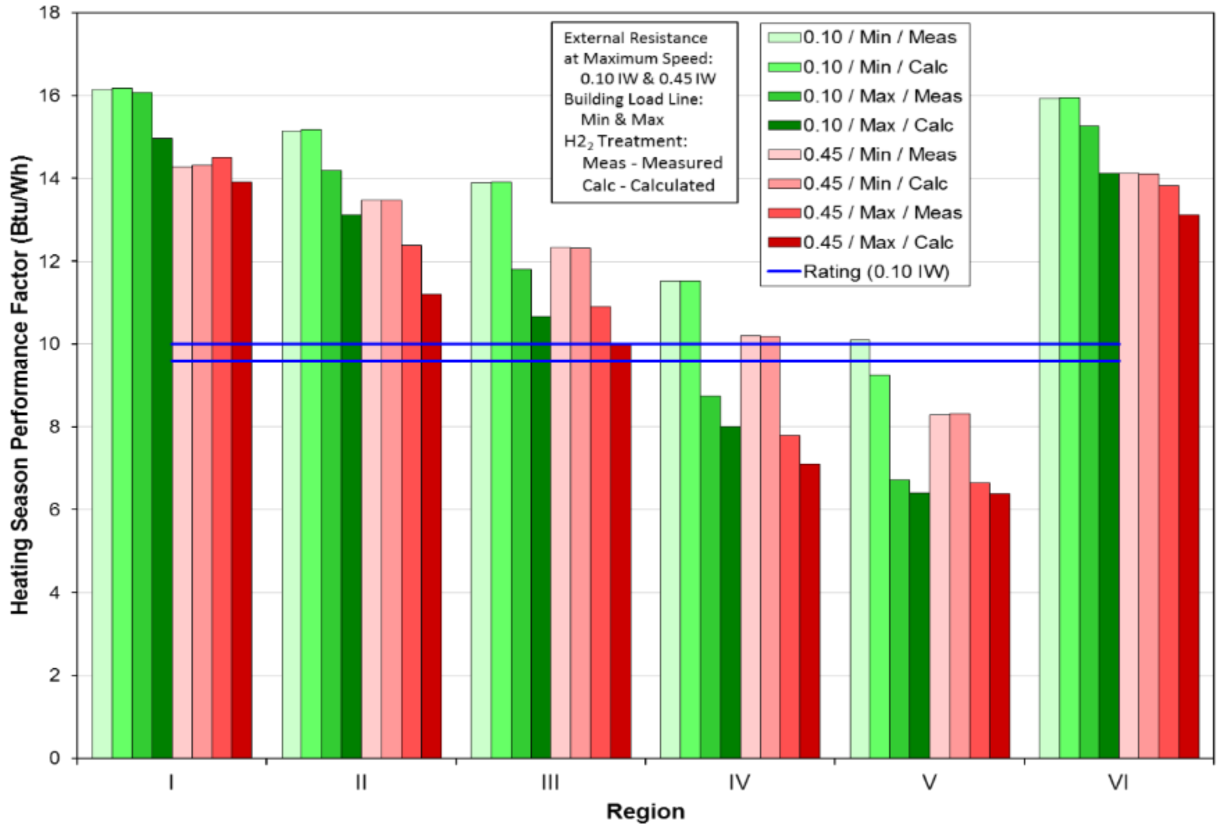
The heating dynamic tests used the same algorithmic procedure developed for the cooling mode tests, but with the added variable of dealing with icing. Another issue is the settings within the thermostat that control when the unit will switch between heat pump operation and the gas furnace for heating, and what values are important in this changeover. These settings include:

- "Backup Heat Differential" – the temperature difference between the air temperature and the setpoint at which the backup heat is triggered.
- "Compressor Lockout / Balance Point" – the temperature below which the heat pump will not be allowed to operate, and the unit will only run on its backup heat source.
- "Backup Heat Lockout" – the temperature above which the backup heat source will not be allowed to operate, and the unit will only run as a heat pump.

Most of these settings are not visible to the average user (requiring a passcode), and may only be set once by the installer, and thus may not be set optimally. Due to the wide range of possible settings, only two sets of them were examined.

[2] Although the test unit uses a natural gas furnace for its backup heat source, the HSPF calculation still assumes electric resistance heat since it is applied to an outdoor unit that could be combined with either. The furnace combination is also an either/or function where the system will operate as a heat pump or using the furnace, but not together, while the electric resistance option is providing supplemental heat with both systems running.

Figure 38: Multiple HSPF Values Derived from the Same Test Data Sets



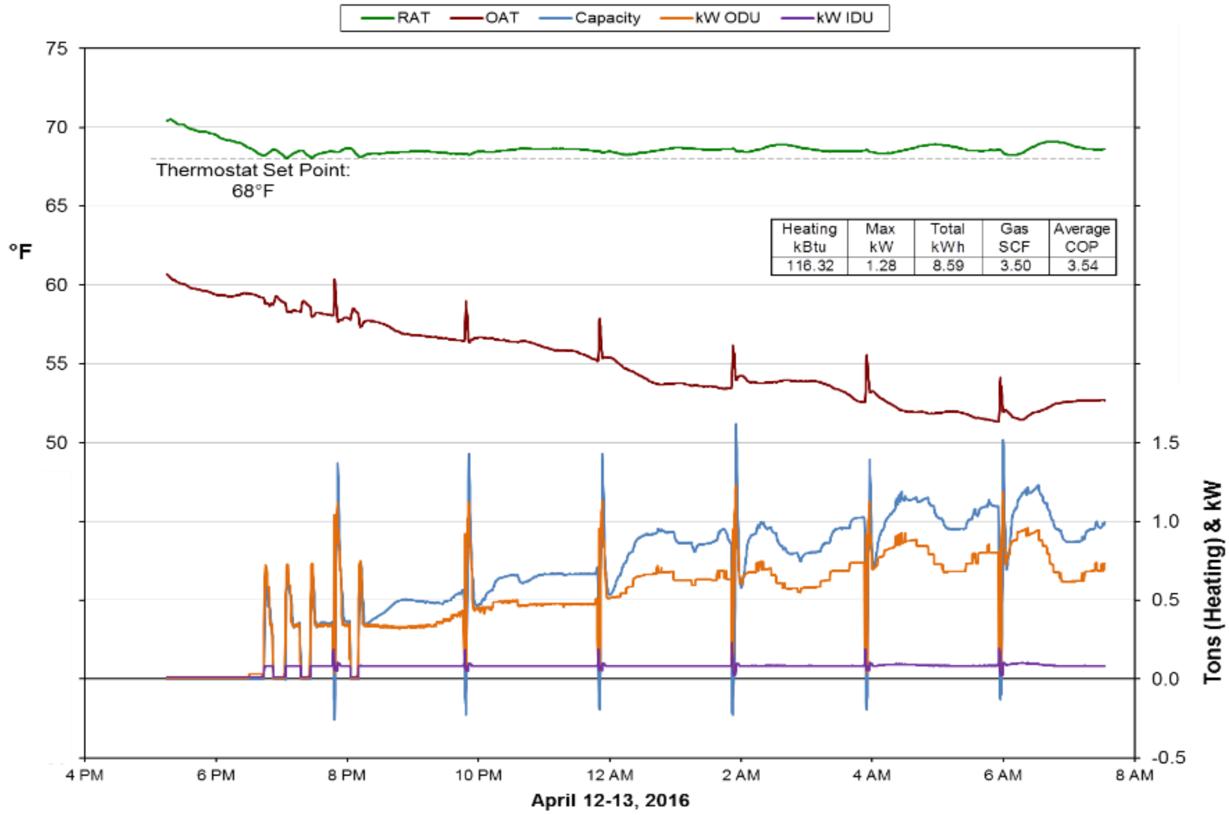
Source: EPRI

Heating Mode Dynamic Test Results

The first attempt at dynamic heating mode testing did not actually use the algorithm or control temperatures in any way. It was just to check that the system would be able to control the temperature in the indoor room via its thermostat. The test was conducted overnight by running the space conditioning apparatus on the outdoor room in full ventilation mode (economizer wide open), and the indoor room with its conditioning system doing mainly recirculation but with the economizer cracked open to create some load from the cool outside air. The thermostat was set for heating mode with a setpoint of 68°F (20°C).

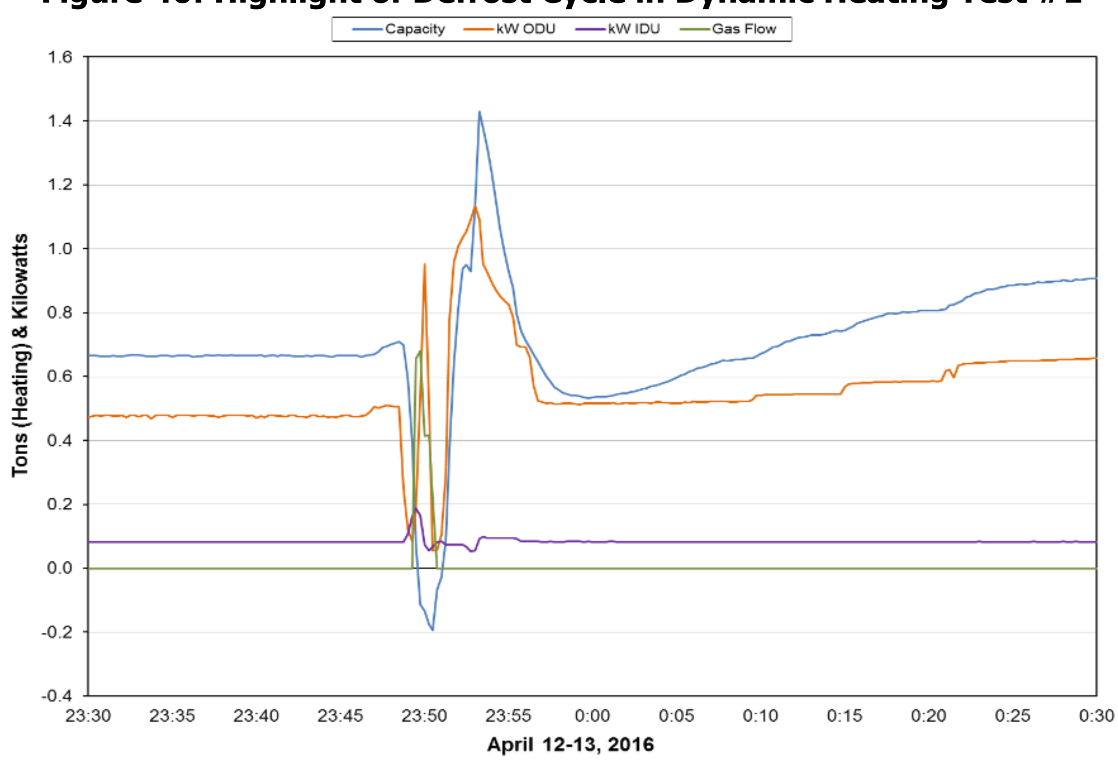
Figure 39 shows the trend of performance indicators from this test, which shows that as the outdoor room temperature dropped, the heat pump first started operation by cycling on and off until eventually running continuously at a slowly increasing capacity and power. This trend also highlights the defrost cycles that occur every two hours. For defrosting, the system switches to cooling mode, causing the outdoor unit to reject heat rather than absorb it from the air. With the outdoor unit enclosed in a 10-foot cube, the sudden change from absorbing to rejecting causes a spike in the measured outside air temperature that would not normally occur. A close examination of the defrost cycle (Figure 40) shows that the system also activates the gas furnace for a short period in order to offset the cooling done by the test unit when it switches to cooling mode, and also to provide energy that will be absorbed and transferred to defrosting the outdoor coil. However, since the outside temperature never dropped below 50°F (10°C), it is highly unlikely that there would have been any frost accumulated on the outdoor unit.

Figure 39: Dynamic Heating Test #1



Source: EPRI

Figure 40: Highlight of Defrost Cycle in Dynamic Heating Test #1



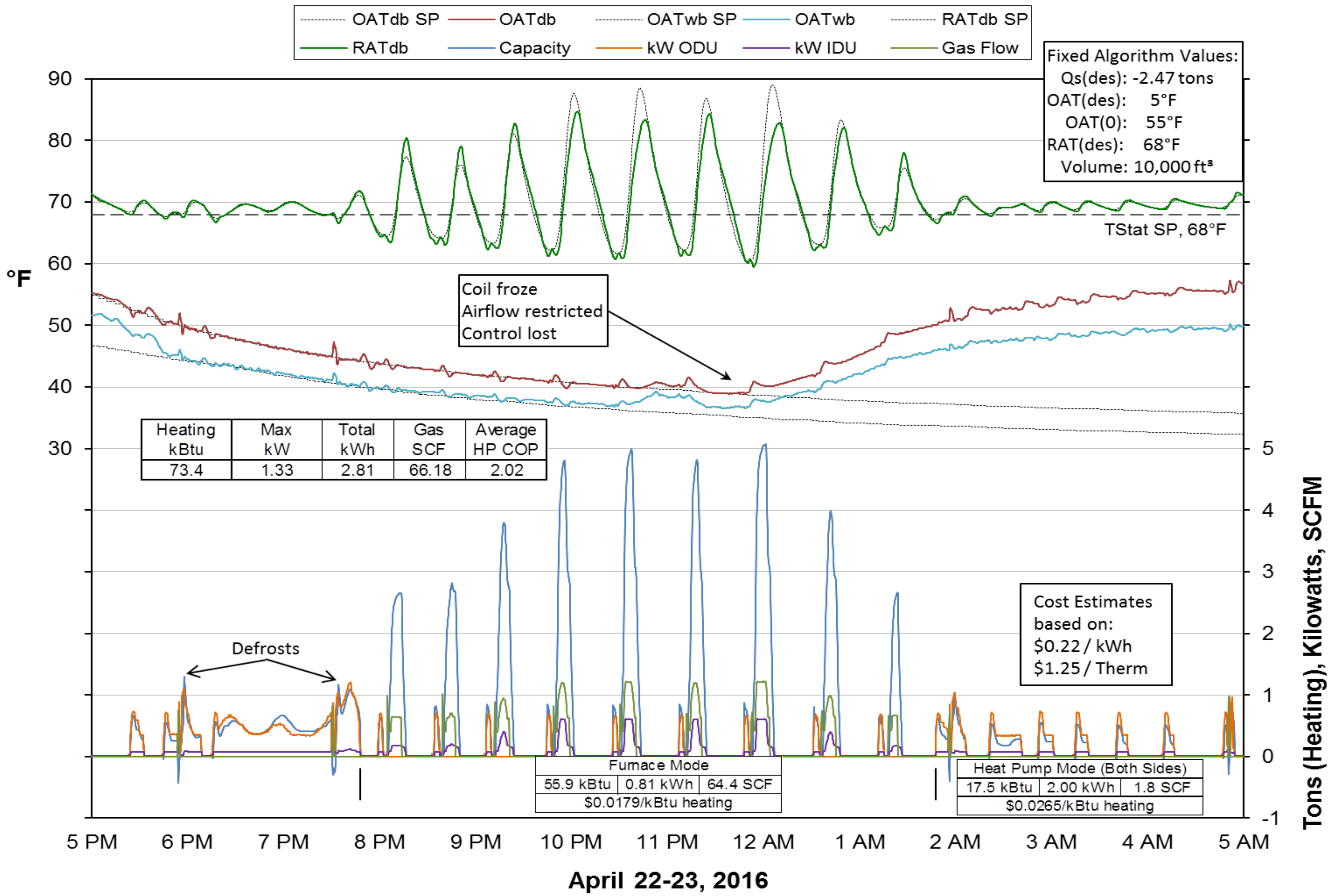
Source: EPRI

Other than briefly during the defrost cycles, the unit did not operate in furnace-priority mode and was able to keep the space at the thermostat setpoint using just the heat pump.

However, this was not the case from the first attempt using the room control algorithm and lower outdoor temperatures. The formulation of the algorithm did not change from the cooling mode dynamic tests, but the constants in the formula have been changed and the design capacity is now a negative number to represent heating rather than cooling. The building load line was based on the assumption that at an outside temperature of 5°F (–15°C) the load would be 1.5 times the rated capacity obtained from the $H1_2$ test. This was derived from the same reference temperature as Region IV, and a mid-approximation of the minimum and maximum building load lines used in the HSPF calculation. The outside temperature assumed for zero load was set to 55°F (13°C) versus the 65°F (18°C) used for HSPF, and the design room temperature was set to 68°F (20°C); the same as the setting in the thermostat. For the first test, the outside temperature was programmed to follow a realistic nighttime temperature profile ramping down from 60°F (16°C) aiming for a low of 35°F (2°C).

The result of this test is shown in Figure 41. The test was begun with a steady-state test at the $H1_2$ test condition (manually controlled using the manufacturer's program) in order to capture the capacity and duct coefficient to be applied during the remainder of the test. The system was then switched to control via its thermostat, and the room temperature was purposely dropped in order to trigger a heating cycle to confirm that the thermostat would control the test unit properly. The room temperature setpoint was then switched to being controlled by the algorithm. Up until about 8 PM, the system was able to meet the heating load with just the heat pump, either cycling or operating at a low speed capacity. After 8 PM, the operation of the system changed to cyclic with just a brief operation as a heat pump before switching over to the gas furnace. The output of the gas furnace is considerably more than that of the heat pump, and the measured room temperature went through some large temperature swings, although this does not necessarily reflect the temperature actually measured by the thermostat.

Figure 41: Dynamic Heating Test #2 (with Realistic Temperature Profile)



Source: EPRI

At around midnight, the coil of the chamber conditioning apparatus froze, and control of the outdoor room temperature was lost. The temperature in the outdoor room rose to approach that of the outside air, and once it reached about 50°F (10°C), the system switched back to just cycling the heat pump to maintain the room temperature. This same scenario was repeated on another night except with a minimum temperature setting of 40°F (4°C) with a similar result (shown in Figure B-13 in Appendix B).

The result from this test can also be used to examine the difference in the cost of operation between heat pump mode and furnace mode. In this example, heat pump mode is assumed to be the shoulder periods when the furnace was not activated between 5 PM to 8 PM and just before 2 AM to 5 AM, and does not include the brief starts before activating the furnace. The representative energy costs chosen for comparison were $0.22/kWh (based on the current "Average" cost for PG&E's E-1 residential rate) and $1.25/Therm (based on a historical residential average over the past two years). This test is not a particularly good example, since only 24 percent of the total heating was done with the system in heat pump mode. The operating cost numbers produce $0.0265 per thousand Btu of heating for the heat pump, and $0.0179 per thousand Btu for the gas furnace heating, making the furnace operation more economical than the heat pump. Factors that can affect the actual cost of power such as on-site solar generation or net metering credits will help to improve the economics of operating in heat pump mode.

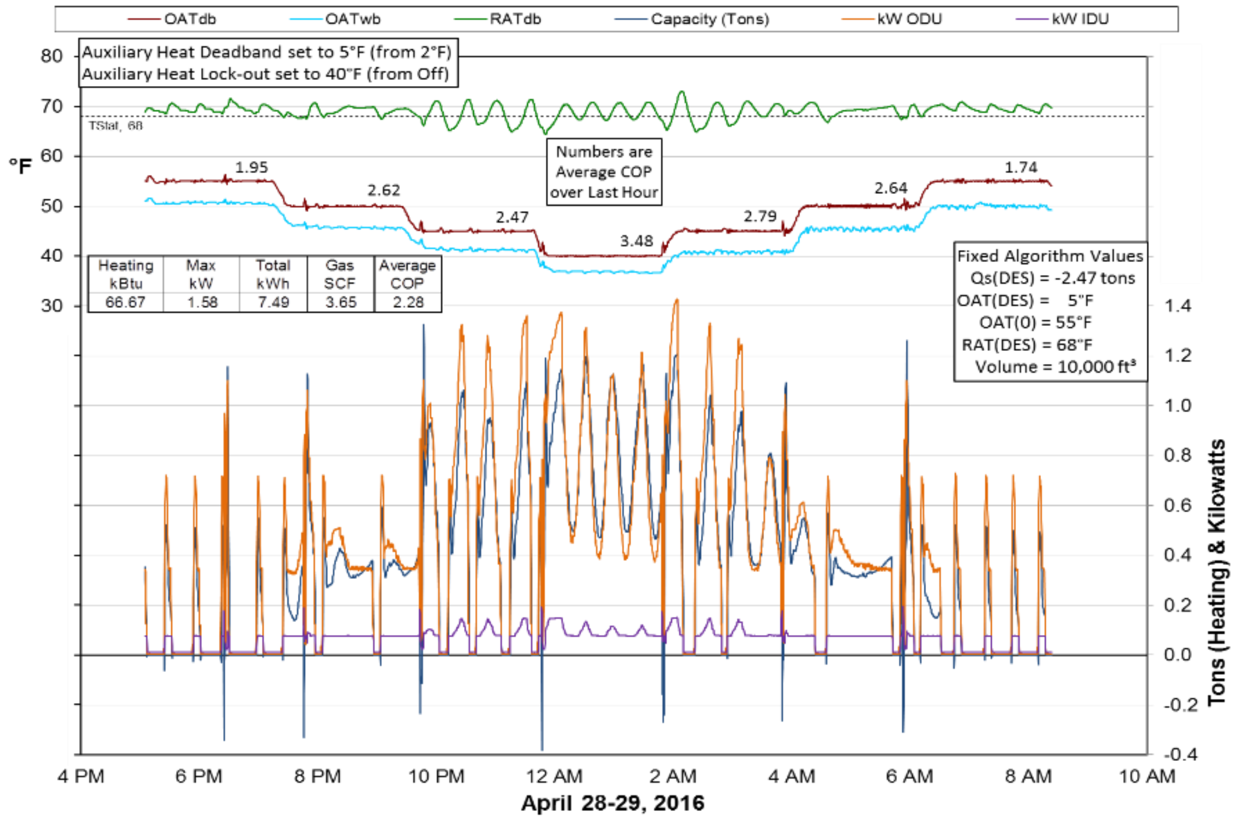
Following these two tests and another attempt at running a real-world profile (Figure B-13), the settings of the thermostat were checked to try to determine why it was switching to gas heat before it even reached maximum capacity of the heat pump. The variables that control the interchange were set at their default, such that the backup heat would be triggered when there was only a 2°F (1.1°C) temperature differential between the reading and the setpoint, and no lockouts were set for either the heat pump or backup heat. As the temperature difference between the RAT and the OAT increases, the rate of change for RAT also increases, and this may have led to the thermostat detecting the 2°F (1.1°C) differential before it had a chance to ramp up the heat pump to a higher capacity.

The default response of the system may also be related to another thermostat setting that defines the type of backup heat, which for this unit is "High Efficiency Gas Forced Air." With this backup source being usually more economical under PG&E rates, the default settings may have it using that heat source preferentially or whenever the outdoor temperature is below a certain level. This setting also puts the system into an either/or mode of operation, so that only the furnace or the heat pump can be on at any time and never together (except during defrost events). To explore the sensitivity of the system behavior to the thermostat setpoints, the thermostat was changed to have a 5°F (2.8°C) differential before triggering the backup heat, and the backup heat lockout was set to 40°F (4°C). The low temperature lockout for the heat pump was left as "disabled."

The next run of a dynamic heating test was done using two-hour holds of outside temperature ranging between 55°F (13°C) and 40°F (4°C) in 5°F (2.8°C) steps. The inputs to the room temperature setpoint algorithm were left the same. The response of the system to this scenario is shown in Figure 42. With the adjusted settings, the gas furnace was never triggered (other than briefly during the defrost events). The capacity of the heat pump only appeared to reach a near-steady level when the outside temperature was around 50°F (10°C).

At warmer temperatures, the unit cycles on and off, and at lower temperatures it either cycled or fluctuated over a fairly broad range of capacity. The unit still did not reach full speed operation at any time during this test.

Figure 42: Dynamic Heating Test #4 (with Outside Temperature Steps)

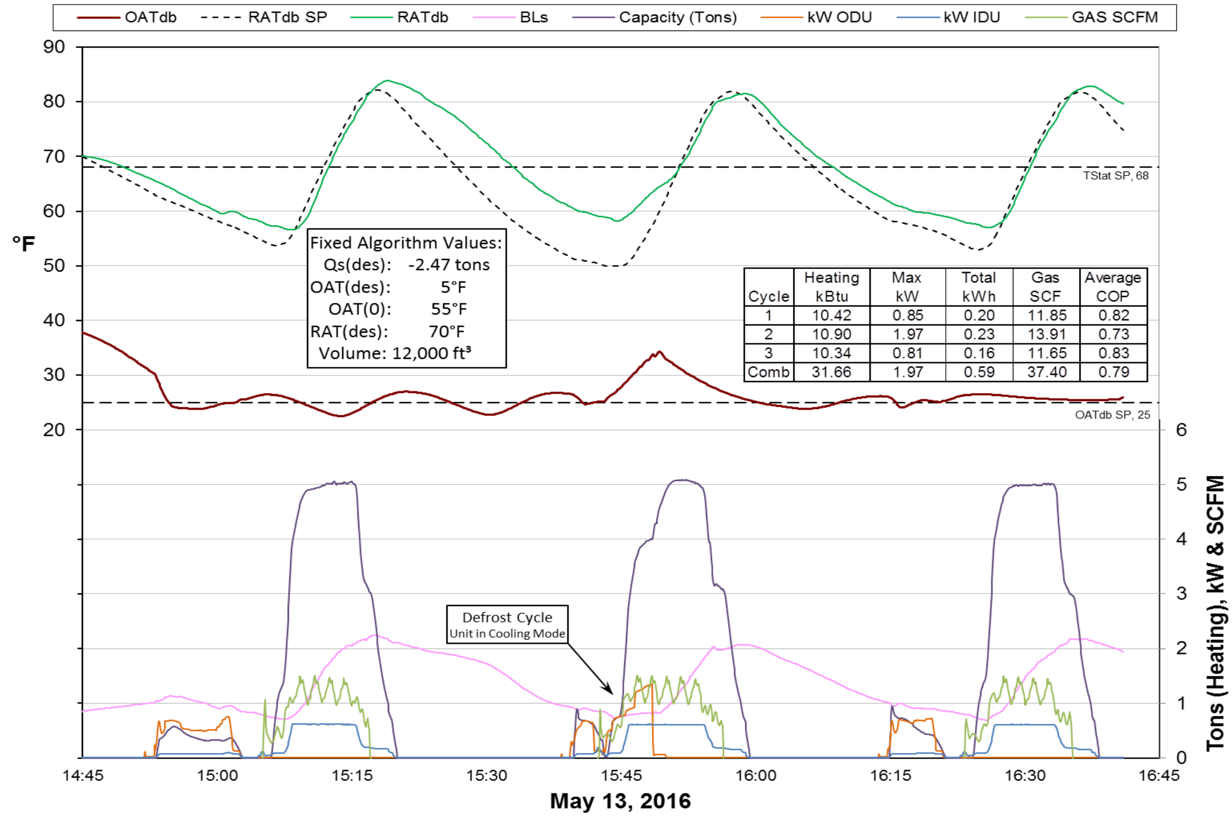


Source: EPRI

For the next test, the outdoor room space conditioning system set to operate with full ventilation air and a fixed level of cooling. Even though this would not be a repeatable pattern, it was thought that this mode of operation would not lead to the space conditioning system freezing. The indoor room temperature was still controlled according to the algorithm. The response from the test unit was similar to the previous test, except without the large fluctuations. This test is shown in Figure B-14 in Appendix B.

Figure 43 provides a detail of three "on" cycles at a very low outside temperature (25°F [–4°C]) with the adjusted settings. With the backup heat lockout set to 40°F (4°C), the system returned to switching over to gas heat before ramping up the heat pump. During the second cycle, a defrost sequence was triggered, putting the refrigeration circuit into cooling mode. The output heating capacity is visibly reduced during this event and represents the only time that the compressor and the gas furnace operate simultaneously.

Figure 43: Dynamic Heating Test #5 (Holding 25°F OAT)



Source: EPRI

Several more attempts at running dynamic heating tests were conducted in an attempt to try to achieve a repeatable outdoor temperature pattern, with limited success. These tests were conducted on the night of April 27-28 (Figure B-13), the weekend of May 20–22 (Figure B-15), the night of May 24-25 (Figure B-16), and the night of May 26-27 (Figure B-17), with the figures located in Appendix B. For the weekend test, the outdoor chamber space conditioning system was manually switched to heat pump mode in order to defrost its coil after operating for about 12 hours in a frozen state. The next two tests attempted to include a periodic defrost into the code used to control the space conditioning system, by toggling its reversing valve on a regular basis if its supply temperature is below 45°F (7°C). In the first attempt, the defrost cycle period, was too long, which caused the room temperature to rise back above 50°F (10°C). The unit operated to keep the coil from frosting even while attempting to bring the room temperature down to 35°F (2°C). For the second attempt, the defrost cycle period was shortened, but by too much and the coil eventually did freeze up more than what the subsequent defrost cycle could handle. In all three of these scenarios, the furnace was effectively locked out at temperatures above 40°F (4°C) but switched to gas-priority below 40°F (4°C), cycling the furnace on as needed.

Table 17 presents a summary of performance metrics from the heating mode dynamic tests. Following the column with the test start date are the minimum and maximum temperatures achieved for the outdoor room. The next column is the total amount of heating energy supplied to the indoor room from the test unit in thousands of BTUs, followed by what fraction of this was provided while the system was in heat pump mode rather than furnace mode. (Note that there is gas usage in heat pump mode during defrosts, and in furnace mode prior to each burner event is often a brief operation of the compressor in addition to the electric consumption by the blower, so neither mode of operation is energy source exclusive.) In the next columns are the maximum system electric demand during the test (always from while operating as a heat pump), the total electric energy consumed during the test period, and the average COP when operating in heat pump mode. (COP is the heating energy delivered in Btu divided by the sum of the electrical energy consumed multiplied by 3412 Btu/kWh, and the standard cubic feet of gas consumed multiplied by 1020 Btu/SCF.) The next two columns give the total gas consumption for the entire test period and the average COP when operating in furnace mode. The last two columns give the average operating cost from operating in either heat pump or furnace mode based on $0.22/kWh and $1.25/Therm for gas. In all of the tests where the system switched between heat pump and furnace modes, the furnace was more economical to operate. However, in the very first dynamic test on April 12, the economics for the heat pump were better than any of the later furnace mode tests as the result of a much higher average heat pump COP. This likely derived from the system operating at a steady operating condition most of the time with very little cycling.

Table 17: Summary of Heating Mode Dynamic Tests

Test Start Date	Duration (Hours)	OAT Low	OAT High	Total Heating kBtu	HP Mode Heating Fraction	Max kW	Total kWh	Avg HP COP	Total SCF	Avg Fur COP	Operating Cost* $ per kBtu of Heat HP	Operating Cost* $ per kBtu of Heat Furnace
Thermostat set to default 2°F differential to trigger backup heat and no lockouts												
April 12	14.4	46	61	116.3	100%	1.28	8.59	3.54	3.5	-	$0.0166	-
April 22	12.0	39	57	73.4	24%	1.33	2.81	2.02	66.2	0.82	$0.0265	$0.0179
April 27	16.3	41	56	98.4	45%	1.22	5.34	2.60	63.1	0.83	$0.0225	$0.0182
Thermostat adjusted to 5°F differential and 40°F backup heat lockout												
April 28	15.3	40	56	66.7	100%	1.58	7.49	2.28	3.6	-	$0.0247	-
May 18	46.0	44	60	91.3	100%	1.20	12.01	1.75	11.0	-	$0.0305	-
May 20	42.7	38	67	125.5	10%	1.26	12.35	2.33	21.0	0.80	$0.0244	$0.0175
May 24	16.7	36	52	135.8	19%	1.67	6.24	2.16	141.1	0.72	$0.0279	$0.0223
May 26	16.9	37	53	120.8	62%	1.52	7.73	2.59	61.3	0.81	$0.0220	$0.0171

*** Operating cost based on $0.22/kWh and $1.25/Therm for gas.**
RATdb Control Algorithm Settings: QS_{DES} = -2.47 tons, OAT_{DES} = 5°F, OAT_0 = 55°F
RAT_{DES} = 68°F, V = 12,000 ft^3

Source: EPRI

Dual Fuel Heat Pump Analysis and Test Results

The dual fuel heat pump function of the Next-Generation Residential Space Conditioning System (Next-Gen RSCS) for California uses a variable speed electric heat pump as the primary heating means and a gas furnace for supplemental heat and when it is more economical to run the furnace.

The following paragraphs summarize the operation and energy cost analysis of the Next-Gen RSCS for a range of operating conditions and utility prices.

An Excel spreadsheet program was developed and utilized to perform the required analyses, determining the balance point temperature and the breakeven temperature, calculating heat pump and gas furnace heating costs and comparing these costs.

The building heating load was 44,867 Btu/hr @ 12°F (–11^{o}C) and zero at 62°F (17^{o}C). The furnace capacity was 80,000 Btu/hr Next Gen heat pump capacity varied with speed with a capacity of 21,167 Btu/hr at 37°F (3^{o}C) and 13,773 Btu/hr at 17°F (–8^{o}C) at maximum speed.

Above the balance point the Next-Gen RSCS varies the speed of the unit to provide the capacity to match the building heating load.

The balance point temperature, the temperature at which the heating load exceeds the heat pump capacity, was approximately 37.8°F (3.3°C). Above the balance point the heat pump can handle the entire load and no backup is required. Below the balance point backup is required and the furnace must supply the shortfall in capacity.

Calculations were performed to determine the breakeven temperature based on heat pump and furnace efficiencies and the relative prices of electricity and natural gas. Above the breakeven temperature, it is more economical to run the heat pump and below the breakeven temperature it is more economical to run the furnace. For the Next-Gen RSCS heat pump efficiencies and an AFUE of .97, the prices used in this analysis resulted in breakeven temperatures ranging from 22°F (–6°C) to 47°F (8°C). Appendix C describes the dual fuel heat pump spread sheet analysis and results.

Appendix C also describes the functional testing performed to verify dual fuel control system operation in switching between the heat pump and furnace above and below the breakeven temperature and in supplying backup heat from the furnace below the heat pump balance point.

Western Cooling Efficiency Center Laboratory Activities and Results

The laboratory experiments for this project were planned in two phases. The objective of Phase 1 experiments was to study the effect of evaporator airflow and compressor speed on the overall system efficiency of a single-zone residential air-conditioning system that includes equipment and ductwork running through an attic. Phase 1 testing was completed between December 2015 and May 2016 at the UC Davis Western Cooling Efficiency Center (WCEC) Laboratory. Phase 2 testing was performed between September 2016 and March 2017 and assessed the impact of multi-zone capabilities combined with variable speed controls. The results of Phase 1 testing are described below.

Laboratory Setup

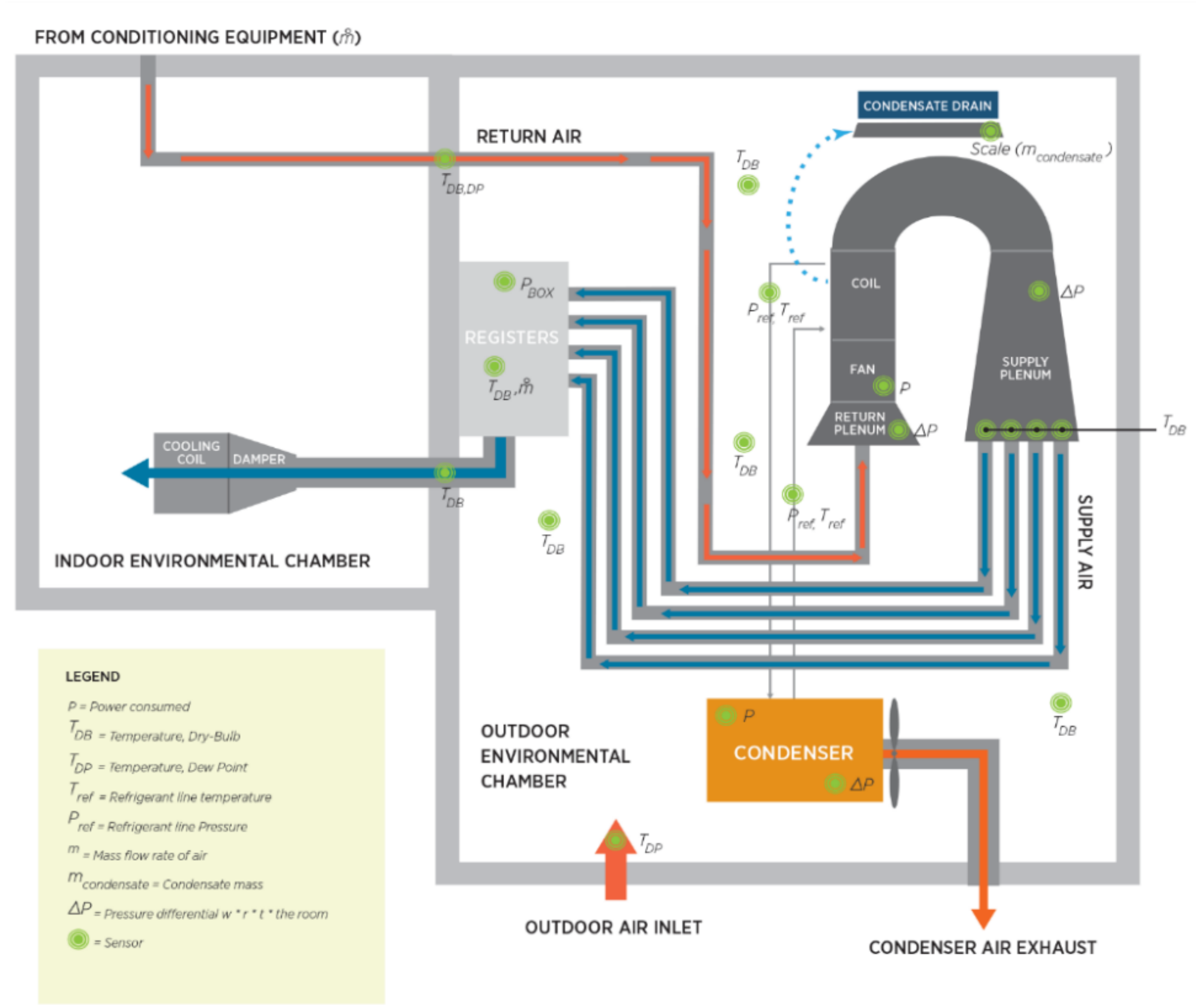
Psychrometric Rooms

The air-conditioning system was tested in the UC Davis Western Cooling Efficiency Center Laboratory in Davis, California. The laboratory includes two environmental chambers designated to simulate various outdoor and indoor climate conditions. Dry-bulb and wet-bulb temperatures in these rooms were maintained within ± 0.5°F (0.4°C) during steady-state operation. Conditioning equipment located on the roof of these chambers is capable of heating, cooling, dehumidification and humidification of air, so that the air being returned to the equipment is at the desired temperature and humidity set-points. Each chamber includes a fan to provide the required air-flow through the rooms while maintaining the desired pressure difference across the evaporator fan-coil unit and the condenser coil.

Assembly of Experimental Apparatus

The schematic diagram of the experimental set-up is shown in Figure 44. Conditioned air from the indoor psychrometric room passes through a 16" diameter x 15' long return duct into the return plenum of the air handler. The supply plenum is designed to split the airflow into four paths, each path representing the airflow to a particular zone. Thus, there are four zones and eight supply air registers in total. (For Phase 1 testing the system was set up as a single zone with no zonal control equipment.) Supply ducts are connected to each outlet of this four-way splitter.

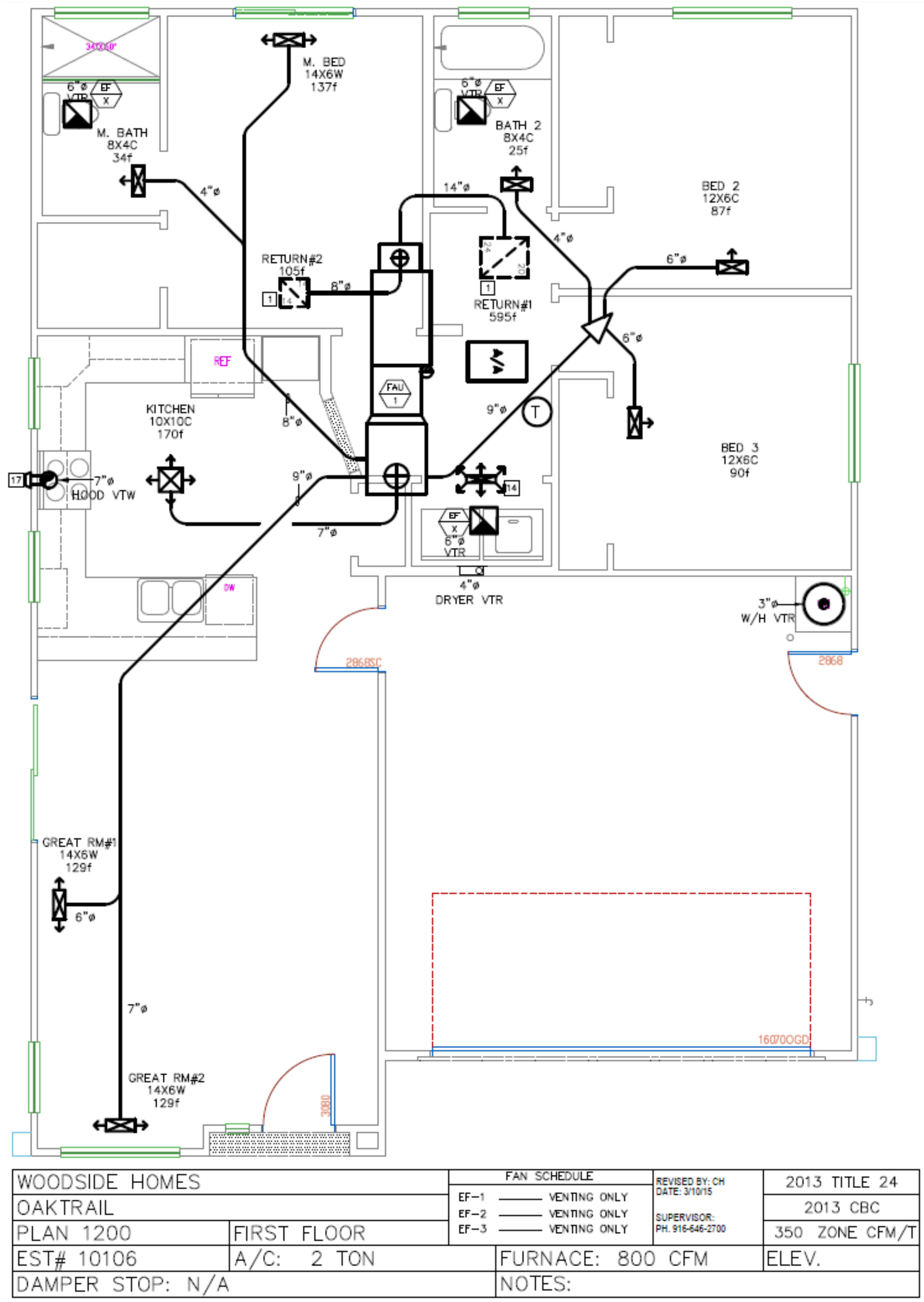
Figure 44: Experimental Apparatus and Instrumentation



Source: EPRI

Sizing of the supply (and return) ducts was guided by 2013 Title 24 Residential Compliance Standards for a single-story home that utilizes a 2-ton A/C unit (Figure 45). The trunk sections carrying air to zones 1 and 3 were split into two branches each and each branch delivered the air at a grille downstream. The trunk section carrying air to Zone 4 split into three branches, with each branch delivering the air at a grille downstream. The trunk section for Zone 2 delivers the air at a grille downstream directly without splitting into branches.

Figure 45: Duct Layout for a Single-Story Home



Source: EPRI

The grilles for all the zones were installed on a 72″ x 40″ x 20″ wooden plenum box (Figure 45). A 10′ rubber flex-duct was attached to one side of the wooden box and connected to the indoor environmental chamber to deliver the cold air coming out of all the grilles to the room. All duct sections were arranged on shelves to prevent direct thermal contact between ducts and an effort was made to reduce air leakage of the duct sections. The outdoor unit of the A/C equipment was ducted out of the outdoor chamber. In order to ensure that the air in the attic was well-mixed, roughly five times the manufacturer-specified amount of airflow through the condenser unit was supplied to the outdoor chamber. Roughly one-fifth of the air exits through the duct to the condenser unit, and the remaining air exits through the chamber exhaust. The additional airflow through the chamber allowed the duct system to see a uniform temperature which was as close as possible to the desired attic/outdoor temperature.

Figure 46: Duct and Grille Layout in WCEC Outdoor Environmental Chamber

Source: EPRI

Instrumentation Plan

The instrumentation plan for Phase 1 tests were divided into chamber conditions measurements, evaporator measurements, grille measurements, and refrigerant-side measurements (Table 18).

- Chamber conditions measurements
    - The supply and return air conditions of the indoor chamber were monitored with two General Electric (GE) Optisonde chilled mirror hygrometers. The chilled mirrors 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 temperature. For the outdoor chamber, room temperature was measured using 16 Type-T thermocouples strategically

placed at different points in the room such that their average represents the average room temperature seen by the ducts.

- Evaporator measurements
    - Average dry bulb temperature of the supply plenum was measured using RTDs installed at each outlet of the 4-way splitter. Return plenum dry bulb temperature was measured using a single RTD midway through the return plenum of the air handler. The dew point temperature of the supply air was measured just after the supply grilles and assumed to be constant throughout the supply side of the system since the supply ducts operated at positive pressure $T_{dp,grilles} = T_{dp,sp}$. The dew point temperature of the return was measured at the entrance to the return duct, and since leakage was demonstrated to be minimal on the return side of the system (~2 percent of flow), this temperature was also assumed to be constant throughout the return side of the system ($T_{dp,room} = T_{dp,rp}$). The static pressures at the supply and return plenums were measured using an Energy Conservatory DG-700 differential pressure transducer. Condensate generation was measured and recorded using a high accuracy bench scale.
- Grille measurements
    - The measurements recorded at each supply register included the dry-bulb temperature of the air and air flowrate leaving the grille. The latter was measured using pitot tubes, whose differential pressure measurements were recorded using Energy Conservatory APT-8 differential pressure transducer. The Pitot tube measurements were calibrated using a powered flow-hood.
- Refrigerant measurements
    - Properties of the refrigerant were determined by measuring the temperature and pressure of the refrigerant before and after the compressor. 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.

Table 18: Summary of Instrumentation Used

Location	Measurement(s)	Symbol	Instrumentation	Accuracy
Indoor Unit	Power	P	Dent Powerscout 3+	± 1%
	Static Pressure	▪ P_{AH}	Energy Conservatory DG-700	± 1%
Outdoor Unit	Power	P	Dent Powerscout 3+	± 1%
Return duct	Dry bulb Temperature at the end closer to conditioned space	T_{DB}	GE Optisonde 2-1-11-1-0-0-0	± 0.36°F (0.2°C)
	Dew Point Temperature at the end closer to conditioned space	T_{DP}	GE Optisonde 2-1-1-1-1-0-0-0	± 0.36°F (0.2°C)
	Dry bulb Temperature at the end closer to the air handler	T_{DB}	Omega HSRTD-3-100-A-120-E	± 0.15°F to 0.17°F (0.08°C to 0.09°C)
	Static Pressure	▪ P_{RA}	Energy Conservatory DG-700	± 1%
	Air Flow Rate	$\dot{m}_{air}$	Tracer Gas System	±2%
Air entering the supply duct (main)	Dry bulb Temperature	T_{DB}	Omega HSRTD-3-100-A-120-E	± 0.15°F to 0.17°F (0.08°C to 0.09°C)
Air leaving each supply duct through grilles	Dry bulb Temperature	T_{DB}	Omega HSRTD-3-100-A-120-E	± 0.15°F to 0.17°F (0.08°C to 0.09°C)
	Differential pressure	▪ ▪ ▪ P_{grille}	Pitot Tubes calibrated using flowhood capture	±2%
	Average Static Pressure of all grilles	▪ $P_{grille,avg}$	TSI VelociCalc 9565-P	± 1%
Outdoor Environmental Chamber	Dry bulb Temperature	T_{DB}	Type T thermocouples	± 0.15°F to 0.17°F (0.08°C to 0.09°C)
	Dew Point Temperature	T_{DP}	GE Optisonde 2-1-1-1-1-0-0-0	± 0.36°F (0.2°C)

Location	Measurement(s)	Symbol	Instrumentation	Accuracy
Indoor Environmental Chamber	Dry bulb Temperature	T_{DB}	GE Optisonde 2-1-1-1-1-0-0-0	± 0.36°F (0.2°C)
	Dew Point Temperature	T_{DP}	GE Optisonde 2-1-1-1-1-0-0-0	± 0.36°F (0.2°C)
Suction Line	Refrigerant Pressure	P_{ref}	Climacheck 200100	± 1%
	Refrigerant Temperature	T_{ref}	Omega SA1- RTD	± 0.15°F to 0.17°F (0.08°C to 0.09°C)
Discharge Line	Refrigerant Temperature	T_{ref}	Omega SA1- RTD	± 0.15°F to 0.17°F (0.08°C to 0.09°C)
	Refrigerant Line Pressure	P_{ref}	Climacheck 200100	± 1%

Source: EPRI

Experimental Tests

A total of 32 steady-state experiments were conducted for Phase 1. The different combinations of tests involved: (a) varying the compressor speed and the blower speed together between 40 and 100 percent speeds in 20 percent increments, and (b) maintaining the compressor speed constant at 25, 40, 60, 80 or 100 percent while varying the blower speed from 60 to 100 percent in 20 percent increments. During the tests, the average outdoor chamber temperature was varied from 85°F (29°C) to 115°F (46°C) DB in 10°F (5.6°C) increments while the indoor chamber was maintained at 75°F (24°C) DB/62.5°F (16.7°C) WB. One series of tests was also performed with the indoor chamber at 80°F (27°C) DB/67°F (19°C) WB when the outdoor chamber was maintained at 115°F (46°C) in order to analyze the effect of indoor temperature setpoint. Data was recorded at 30 second intervals using LabView. All tests were conducted for a minimum of half hour, with final results calculated based upon the last 30 minutes of operation. Steady state conditions were ensured by adhering to the temperature tolerances set forth in AHRI 210/240.

The external static pressure (defined as the pressure drop across the indoor unit) was controlled to 125Pa when the flow rate through the unit was at 100 percent of the rated flow (for example, 800 cfm for a 2-ton unit). In order to simulate the conditions of a fixed ducting network, as the blower speed was reduced to obtain lower airflow rates, an external circulator fan on the indoor chamber conditioning loop was adjusted to maintain the appropriate pressure vs. flow relationship between for all tests.

Table D-1 and Table D-2 (Appendix D) present a summary of results of steady-state experiments conducted at 95°F (35°C) DB outdoor and 75°F (24°C) DB/62.5°F (16.7°C) WB indoor with varying airflow rates at a fixed compressor speed show the summary of steady-state tests conducted during Phase 1. These tables report the chamber temperatures during testing, the indoor airflow rate and compressor speed, power consumed by the units, and the external static pressure. The calculated parameters being reported are the total delivered capacity, the equipment efficiency, delivery effectiveness of the ducts and the overall system efficiency.

System efficiency was calculated based on the overall cooling/heating supplied to the home, which includes the impact of a duct system placed in an attic ($\dot{Q}_{cool}$). The delivered cooling/heating and the system COP were calculated based on the enthalpy difference between the indoor chamber and the air supplied at the grilles (Equation 12 and Equation 13).

$$\dot{Q}_{cool} = \dot{m}\left(h_{room} - h_{grille}\right)$$ **Equation 12**

$$COP_{sys} = \frac{\dot{Q}_{cool}}{P_{indoor}+P_{outdoor}}$$ **Equation 13**

The equipment efficiency is based on the conditions of the air at the return and supply plenums of the system. This efficiency does not account for losses in the return or supply ducts and is calculated using Equation 14.

$$COP_{equip} = \frac{\dot{m}(h_{rp}-h_{sp})}{P_{indoor}+P_{outdoor}}$$ **Equation 14**

With minimal air leakage throughout the entire system, latent heat transfer between the attic and ducts was ignored. Thus, the overall delivery effectiveness is assumed to only affect the sensible capacity of the air in the ducts (Equation 15).

$$H_{duct} = \frac{T_{room}-T_{grille}}{T_{rp}-T_{sp}}$$ **Equation 15**

For analyzing the effect of the blower power consumption (P_{indoor}) at higher flow rates, a compressor only COP was defined in order to split out the blower power from the rest of the system (Equation 16).

$$COP_{comp} = \frac{\dot{m}\left(h_{rp}-h_{sp}\right)+ P_{blower}}{P_{cond}}$$ **Equation 16**

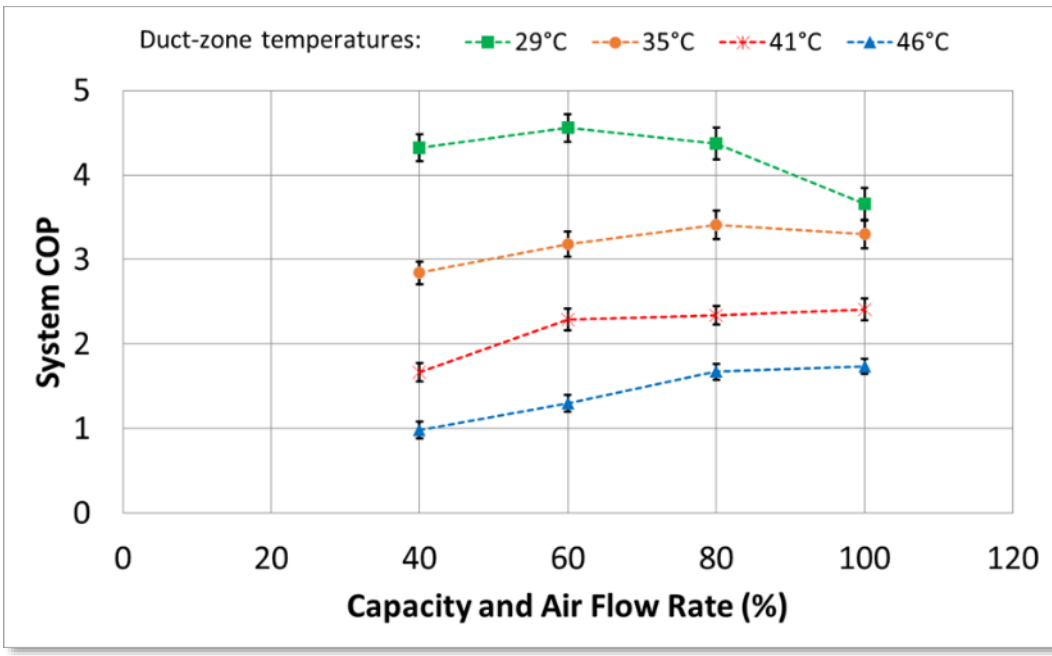
In addition to the steady-state tests, experiments were also performed while the system operated in transient mode in order to compare the system efficiency when cycling versus part load. Four tests were conducted in transient operation at 95°F (35°C) outdoor temperature setting and 62.5°F (16.7°C) DB/75°F (24°C) WB indoor temperature setting. The duty cycle for each of the cycling tests was chosen such that the ratio between the on-time in minutes and the total testing time represented the percentage capacity. For example, the 80 percent capacity cycling test meant that the compressor was on for 48 minutes and off for 12 minutes, such that 48/(48+12) = 80 percent. The 80 percent and 20 percent capacity settings were conducted with two on-off cycles during the one hour of testing, whereas the 40 percent and 60 percent capacity settings had three on-off cycles. During the off-time, the blower was left on until the average temperature of all grilles approximately equaled the room temperature.

WCEC Experimental Results and Discussion

Testing with Compressor and Fan Speeds in Sync

Figure 47 shows the variation of the system COP plotted against the compressor and fan speed percentages, which were kept in sync while the indoor was maintained at Western Indoor Performance Mapping conditions (75°F DB/62.5°F WB or 24°C DB/17°C WB). At an outdoor dry bulb temperature of 85°F (29°C), the compressor/blower speed that delivered the highest system efficiency of the speeds tested was 60 percent of maximum speed. The optimum speed increased to 80 percent as the simulated attic temperature increased to 95°F (35°C). At the hottest temperature tested of 115°F (46°C), running the compressor and blower at speed lower than 100 percent caused the system COP to decrease due to heat gain through the attic. These results clearly indicate that the optimal speed for variable capacity and variable-speed fan air-conditioning equipment depends on the attic temperature in which the ducts are located.

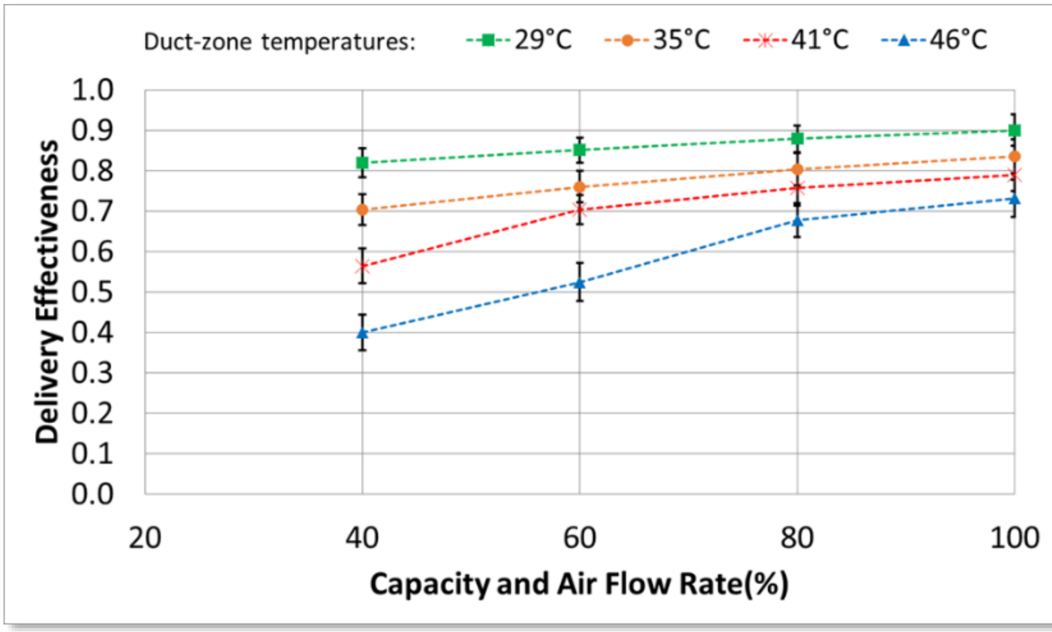
Figure 47: System COP vs. Compressor/Fan Speed Variations



Source: EPRI

The air-conditioning system is composed of the equipment and ductwork, and it has been shown previously that the efficiency of the equipment increases as the speed of the equipment is reduced. However, as illustrated in Figure 48, the effectiveness of the duct system invariably decreases as the air flow rate through them is reduced.

Figure 48: Delivery Effectiveness vs. Compressor/Fan Speed



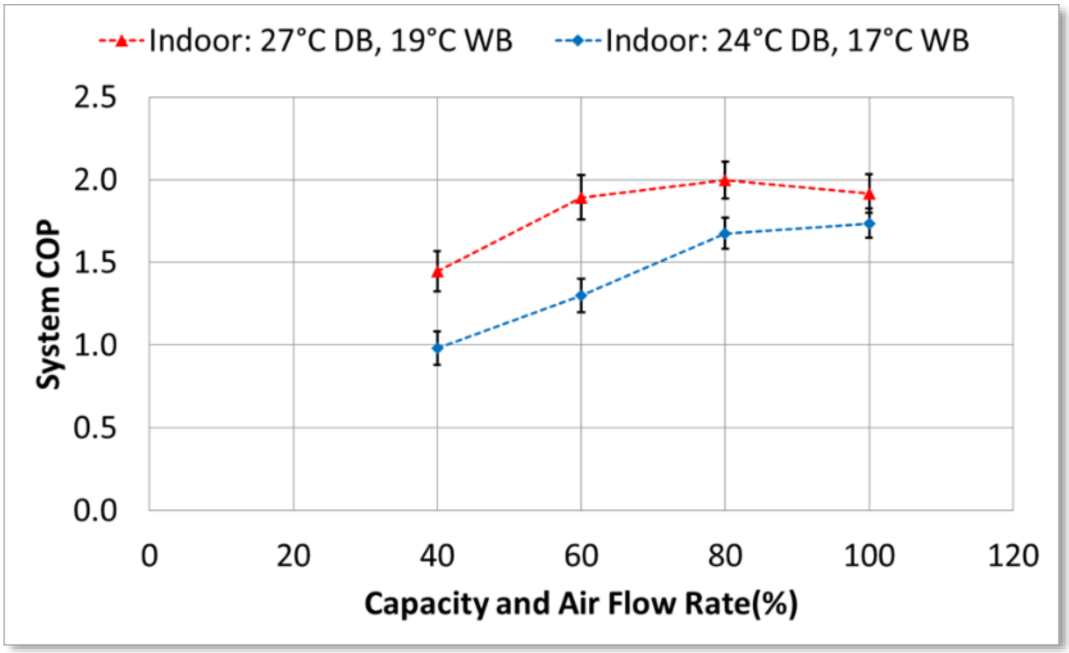
Source: EPRI

This is due to the fact that reducing the airflow leads to longer residence time of air in the ducts prior to reaching the grilles, and the rate of heat transfer is dominated by conduction through the duct walls which is not affected by changing air velocity in the duct. The effect is most pronounced at 115°F (46°C), when the delivery effectiveness reduces from 0.73 at 100 percent blower/compressor speed to only 0.40 when the blower operates at 40 percent of maximum speed. These results demonstrate that placement of ducts in the attic can largely impact the efficiency of variable-capacity equipment. Increasing the R-value requirement for ducts routed through an attic would significantly improve the delivery effectiveness and should be considered as a way to improve the efficiency of the system at lower speeds. Alternatively, this issue could be eliminated by locating the ducts in conditioned space since there would be no losses to unconditioned spaces.

Much of the testing for this project was performed using a relatively dry return air condition to simulate the hot-dry California climate. This resulted in a higher fraction of sensible cooling compared to latent cooling which tends to result in lower delivery effectiveness. Some tests, however, were conducted at the standard AHRI return air condition of 80°F (27°C) dry-bulb temperature and 67°F (19.4°C) wet-bulb temperature to capture a more humid climate condition. To understand the quantitative effect of the air-conditioner being operated at a dry vs. a humid climate, testing was performed by maintaining the indoor condition at 67°F WB (80°F DB) (19°C WB [27°C DB]) and the compressor and fan speeds again varying from 40 to 100 percent in increments of 20%. At this condition, the equipment delivered between roughly 10 and 20 percent latent cooling at an outdoor temperature of 115°F (46°C). The reduction in the ratio of delivered sensible cooling to the delivered latent cooling translates into an increase in both the delivery effectiveness as well as the overall system COP (Figure 49). The delivery effectiveness at the more humid room condition varies between 0.75 and 0.51, as compared to between 0.73 and 0.40 in the dry climate condition. The optimum operating speed for the compressor and fan for maximum system efficiency also reduces from

100 percent of maximum speed for the dry condition to 80 percent speed for the humid condition at an outdoor/attic temperature of 115°F (46°C).

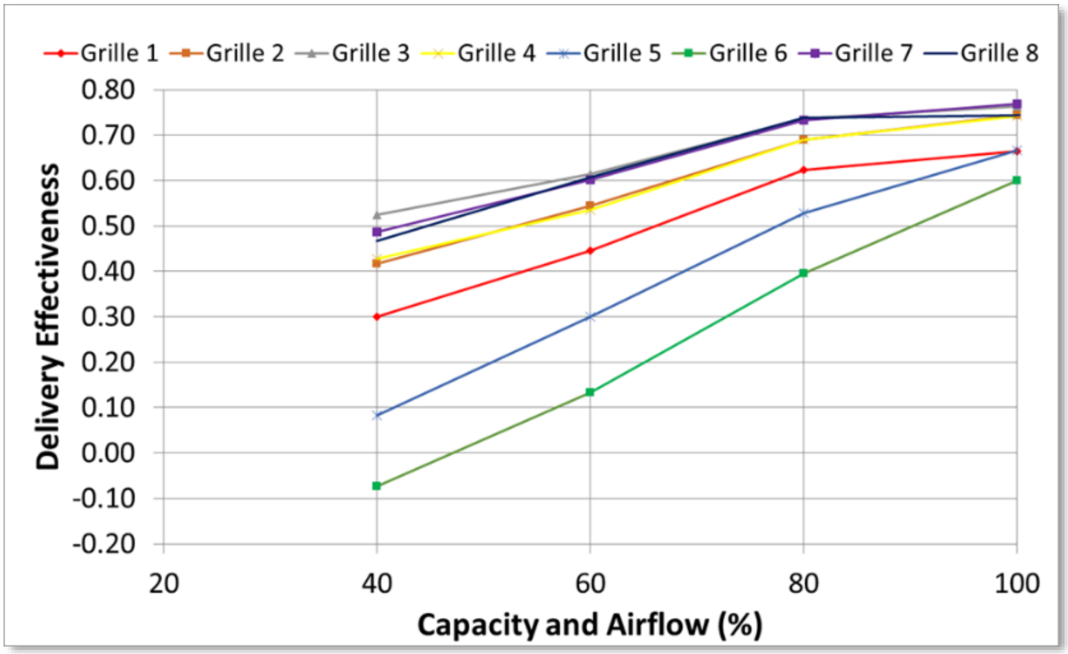
Figure 49: System COP vs. Compressor/Fan Speed (%) at Different Indoor Conditions



Source: EPRI

It is also worth noting that not all the supply ducts in a home will have the same size and length resulting in varying residence times for air in each duct. Figure 50 shows the variation in delivery effectiveness for each individual duct when the compressor and fan speed were varied from 40 to 100 for the case of 75°F DB/62.5°F WB (24°C DB/16.7°C WB) temperature indoor condition and 115°F (46°C) DB temperature outdoor condition. Clearly the efficiency of Grille6, which has the longest residence times among all the grilles, decrease more rapidly when the total airflow rate through the equipment decreases. At a fan speed of 40 percent, or a system airflow rate of 392 cfm, the temperature of air coming out through this grille was warmer than the indoor return air condition, resulting in negative delivery effectiveness. This implies that as the airflow rate through the system is changed, the delivery effectiveness is impacted differently from one grille to another. This result is extremely significant when extending variable-speed fan technology to multi-zone systems.

Figure 50: Duct Efficiencies at Individual Grilles vs. Capacity/Airflow (%)

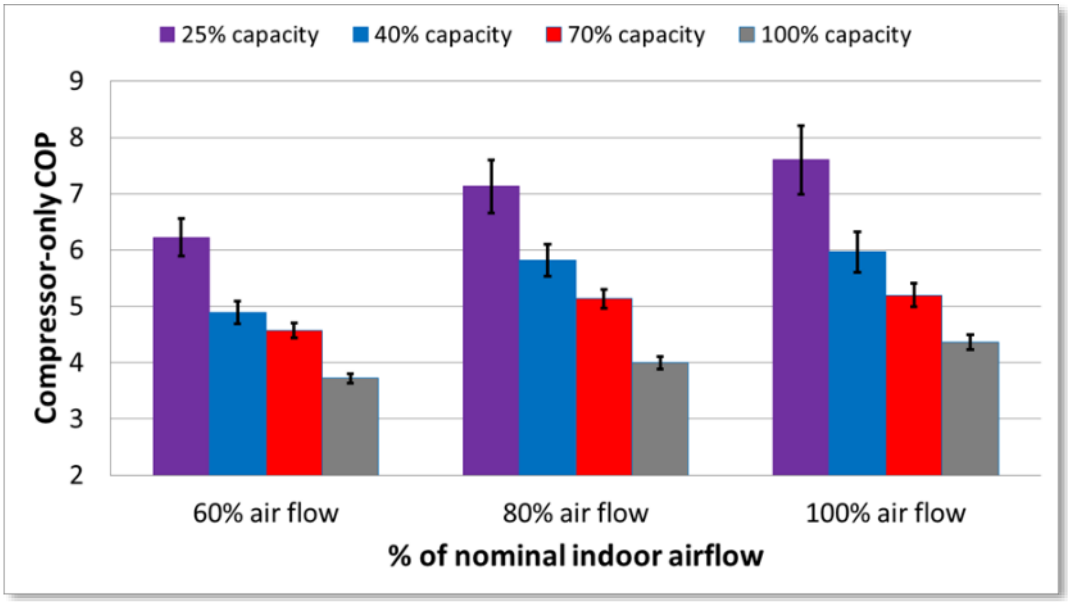


Source: EPRI

Testing with a Fixed Compressor Speed and Varying Fan Speed

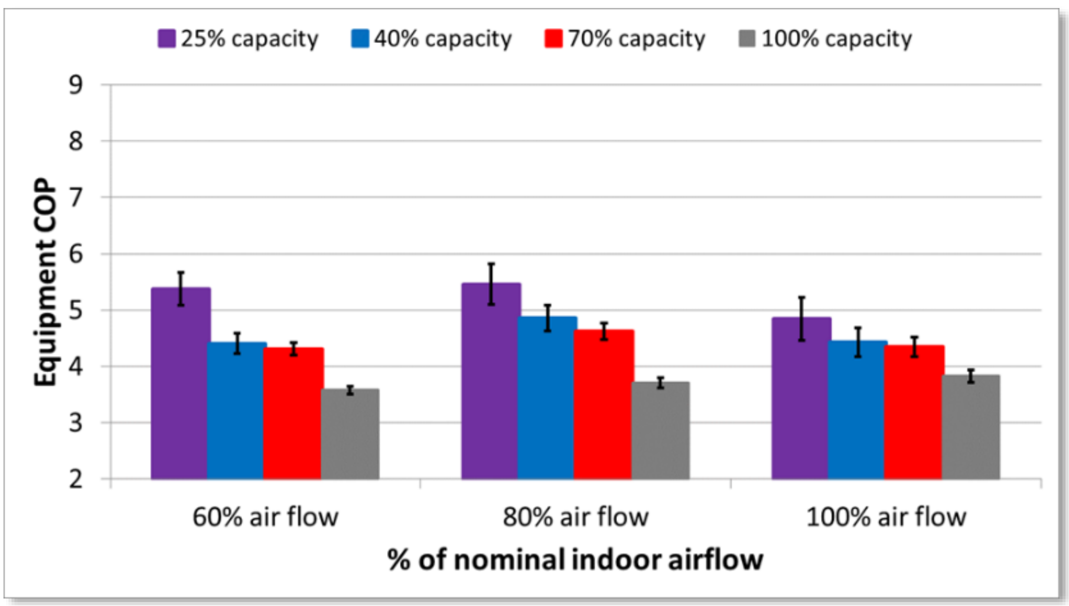
Upon the determination that delivery effectiveness decreases at lower airflow rates, the next step was to test the system at higher flow rates while keeping the compressor at a fixed speed. This experiment also revealed the tradeoff between the aerodynamic, heat transfer, and thermodynamic efficiencies of the system such that the overall system efficiency is maximized. Figure 51 to Figure 53 show the variation of delivery effectiveness, compressor-only COP, equipment COP and system COP for fixed capacities of 25, 40, 70, and 100 percent while varying fan speeds to 60, 80, and 100 percent of full flow. As expected and observed in Figure 47 to Figure 50, it can be seen that the delivery effectiveness improves with increasing air flow rate for all compressor speed settings. In addition, the compressor-only COP, which represents the thermodynamic efficiency of the equipment, also increases with increasing air flow rate. As an example, for the case of 25 percent capacity, when the fan speed increases from 60 to 100 percent the compressor-only COP increased from 6.23 to 7.60 (22 percent improvement). There are two primary explanations for this observation: (1) for the same cooling capacity, the increased air flow results in a higher average air temperature through the evaporator thus reducing the work required to cool that air, and (2) the increased airflow increases the heat transfer coefficient between the air and the evaporator coil improving the efficiency of the refrigerant cycle.

Figure 51: Compressor Efficiency vs. Compressor Speed (%) at Different Fan Speeds - Outdoor condition: 95°F, Indoor: 75°F DB/62.5°F WB



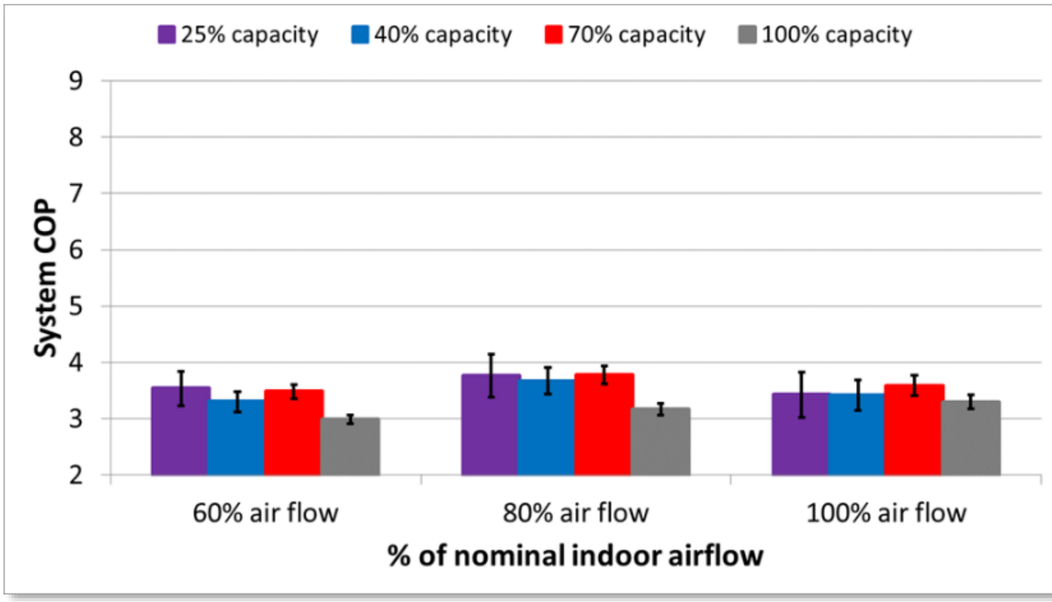
Source: EPRI

Figure 52: Equipment COP vs. Compressor Speed (%) at Different Fan Speeds - Outdoor condition: 95°F, Indoor: 75°F DB/62.5°F WB



Source: EPRI

Figure 53: System COP vs. Compressor Speed Percentages at Different Fan Speeds - Outdoor condition: 95°F, Indoor: 75°F DB/62.5°F WB



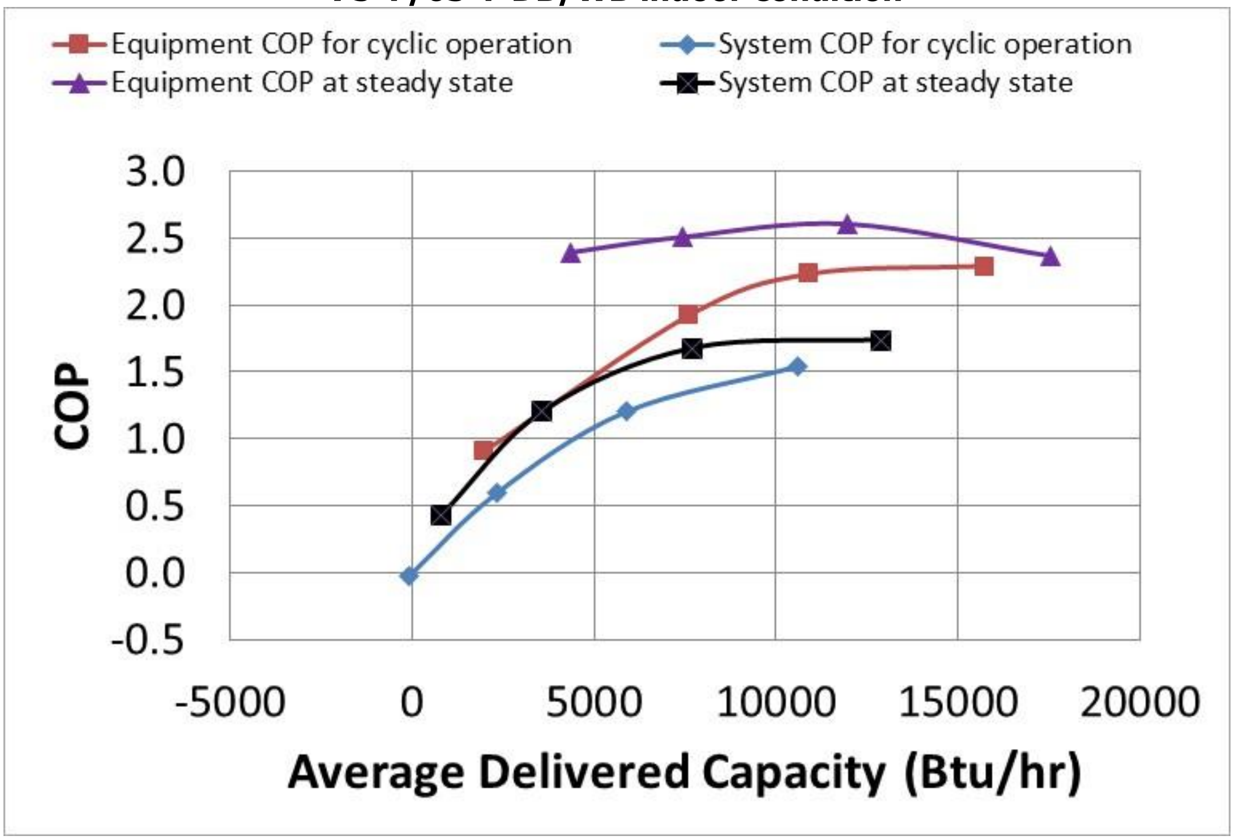
Source: EPRI

Figure 51 clearly illustrates that the thermodynamic efficiency of the air conditioner improves as the compressor speed is reduced. In addition, there seems to be a significant improvement in COP when operating at 80 and 100 percent airflow as compared to operating at 60 percent airflow rate. When the blower fan power is included (Figure 52) the COP is obviously reduced, but what is also evident is that including fan power changes the optimal fan speed in most circumstances. This can be visualized by comparing Figure 51 and Figure 52. In Figure 51, it is seen that 100 percent fan speed setting has the highest COP for all compressor speed settings, but the optimum fan speed when including the impact of fan power usage drops to 80 percent for most compressor speeds (Figure 52). The additional fan power required for increased airflow tends to cancel out the thermodynamic improvement when going to from 80 to 100 percent fan flow. Finally, the system COP trends (Figure 53) which include the impact of delivery effectiveness also indicate the same optimal fan speed setting of 80 percent for compressor speeds at or below 80 percent. In addition, Figure 53 shows that the efficiency gains when reducing compressor speed is somewhat minimal when temperatures are hot and the system is connected to a duct system routed through an attic.

System Testing with Transient Operation

In order to determine whether the variable-capacity/variable-speed equipment has benefits in cyclic operation, testing was performed by cycling the units on and off. Figure 54 shows a comparison of equipment COP and system COP between steady-state and cyclic operation, both COPs plotted against the capacity provided. As expected, the equipment COP is always higher while running the system at steady state than when cycling the unit on and off. This result is consistent with previous studies conducted on transient operation of residential air-conditioning equipment [Tassou et al., Adhikari et al.]. The noteworthy fact however is that delivery effectiveness is higher in case of cyclic operation for the same percentage demand. This can be understood as follows: Due to the equipment being off periodically, the air provided to the ducts during cyclic operation will be warmer on average than in a steady cooling operation. As a result, the driving force for heat loss through conduction reduces, thus increasing the delivery effectiveness. The result of higher delivery effectiveness means that the relative difference in system COPs between the two operating modes is reduced; although for all tests, the steady-state testing still yielded higher system COP.

Figure 54: COP Variations vs. Percentage of Full Demand
Steady-state and transient operating modes at 95°F outdoor temperature and 75°F/63°F DB/WB indoor condition



Source: EPRI

Mathematical Model of Delivery Effectiveness

A mathematical model was developed with the purpose of providing simulated analyses of the detailed interactions between the variable-capacity/variable-speed air conditioner and the duct system, and the combined impact of these on the efficiency of the system. The objective of the model was to capture the energy losses due to conduction heat transfer and leakage in the ducts during steady-state operation, and then to develop functional relationships to characterize the system according to the outdoor air temperature, humidity level, varying compressor speed and varying airflow rate. The basic assumptions of the model as well as their validity with respect to the experiments conducted are listed below.

Model Assumptions

1. Duct thermal resistance is dominated by conduction through the insulated duct wall so the impact of convection resistances on the inside and outside of the duct was ignored. The overall heat transfer resistance of 6 British thermal units per hour per square feet per degree Fahrenheit (Btu/h ft^2 °F) was used for validation with the experiments. For

comparison, the convection resistance on the inside of the duct is less than 1 percent of the conduction resistance through the wall. The model can be modified to include other thermal resistances if deemed appropriate for other situations.

2. The temperature throughout the attic is assumed to be uniform.
3. Thermal regain, or the phenomenon by which heat transfer through the duct walls is split among the various other heat transfer pathways (such as, through roof, ceiling, or ventilation) was ignored.

Under such assumptions, the delivery effectiveness of a single duct, defined as the fraction of the energy imparted to the duct system by the cooling equipment that is delivered to the space at the registers [Modera and Treidler, 1995, Francisco et al., 1998] can be written as:

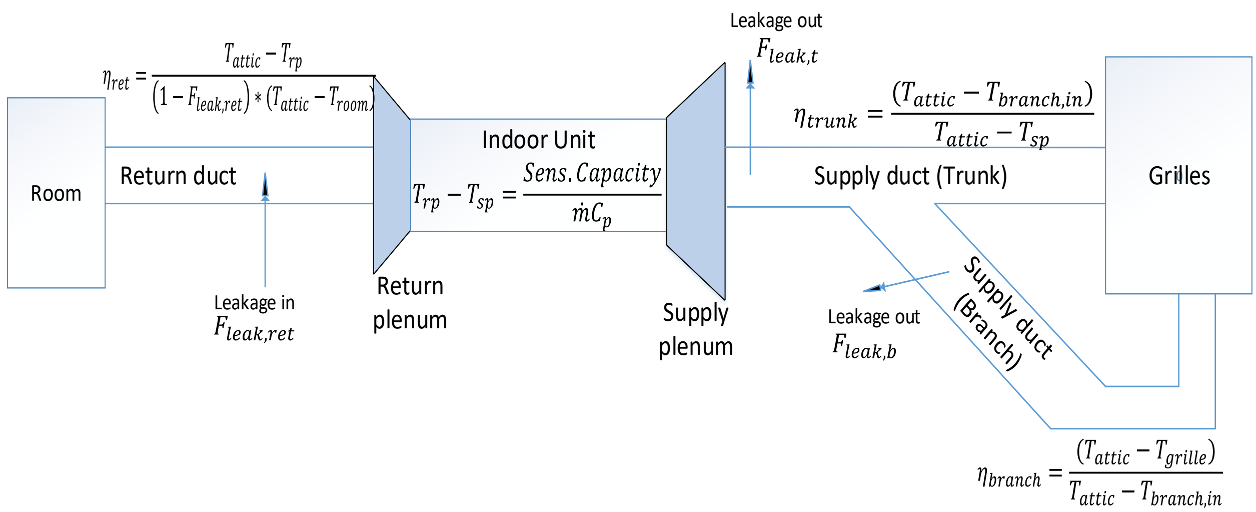
$$\eta_{del} = \frac{\dot{m}_{grille} C_p (T_{room} - T_{grille})}{\dot{m} C_p (T_{rp} - T_{sp})}$$ **Equation 17**

Model Development

Consider the duct system shown in Figure 55. The mass-flow rate of the air passing through the indoor unit is represented by the variable $\dot{m}$. This represents the flow taken from the room, $\dot{m}(1 - F_{leak,ret})$, and combined with the air entering the return duct through leaks, $\dot{m}F_{leak,ret}$, where F_{leak} is the mass fraction of air leaking into the return duct. The supply system is split into the supply trunk and the supply branches where the outlet condition of the supply trunk is the inlet condition for the supply branches. Air leakage from the supply trunk, $\dot{m}(F_{leak,t})$, reduces the overall flow entering the supply branches where additional air leakage in the branch system is then considered, $\dot{m}(F_{leak,b})$. Using heat exchanger theory, the efficiency of each part of the duct system can be separately evaluated using Equation 18.

$$H = e^{\frac{-UA}{\dot{m}C_p}}$$ **Equation 18**

Figure 55: Efficiencies of Different Parts of a Duct



Source: EPRI

By drawing a control volume around the trunk of the supply duct shown in Figure 55, the supply trunk delivery effectiveness is calculated using Equation 19.

$$\mathrm{H_{trunk}} = \frac{\mathrm{T_{attic}} - \mathrm{T_{b,inlet}}}{\mathrm{T_{attic}} - \mathrm{T_{sp}}}$$ **Equation 19**

Similarly, the efficiencies of other components of the duct system can be written as:

$$\eta_{\mathrm{b}} = \frac{\mathrm{T_{attic}} - \mathrm{T_{grille}}}{\mathrm{T_{attic}} - \mathrm{T_{b,inlet}}}$$ **Equation 20**

and

$$\eta_{\mathrm{ret}} = \frac{(\mathrm{T_{attic}} - \mathrm{T_{rp}})(1 - \mathrm{F_{leak,ret}})}{(\mathrm{T_{attic}} - \mathrm{T_{room}})}$$ **Equation 21**

where the term $(1 - F_{leak,ret})$ has been included in the numerator to capture the effect of air leaking into the return duct. Note that there is no leakage term in Equation 19 and Equation 20 since the air through the supply duct leaks out rather than in (due to positive pressures), and as a result does not change the temperature of the air in the ducts.

Finally, by drawing a control volume around the indoor unit:

$$\mathrm{T_{rp}} - \mathrm{T_{sp}} = \frac{\text{Sensible Capacity}}{\dot{m}\mathrm{C_p}}$$ **Equation 22**

Equation 19 through Equation 22 allow the delivery effectiveness to be re-written as:

$$\eta_{\mathrm{del}} = \left(\eta_{\mathrm{branch}}\eta_{\mathrm{trunk}} - \frac{(\mathrm{T_{attic}} - \mathrm{T_{room}})(1 - \eta_{\mathrm{ret}}\eta_{\mathrm{branch}}\eta_{\mathrm{trunk}})(1 - \mathrm{F_{leak,ret}})}{\frac{\text{Sensible Capacity}}{\dot{m}\mathrm{C_p}}}\right)(1 - \mathrm{F_{leak,t}})(1 - \mathrm{F_{leak,b}})$$

Equation 23

Equation 23 defines the delivery effectiveness of a duct system in terms of parameters that are generally known about a particular system; namely attic temperature, room temperature, air flowrate through the equipment, capacity, and the parameters of the duct (length, area, leakage, and duct insulation). The key implications of this equation are as follows:

Lowering the cooling capacity decreases delivery effectiveness. This is physically understandable since the duct losses are nearly constant due to the overall heat transfer coefficient of the ducts being fixed, and as a result, at lower capacities, the relative amount of delivery losses through the ducts is higher.

Higher airflow rate for the same cooling capacity increases delivery effectiveness. This observation has already been validated by the results shown in Figure 47 to Figure 54 and the accompanying text discussing synced operation and operation with fixed compressor speed and varying fan speed.

1. The maximum delivery effectiveness is the product of conduction and leakage efficiencies in the supply duct, regardless of the temperature difference between the outdoors and the indoor setpoint.
2. Longer and larger size ducts have greater conduction losses. The increased length and diameter of the ducts increase the UA value of the duct which reduces the efficiency as

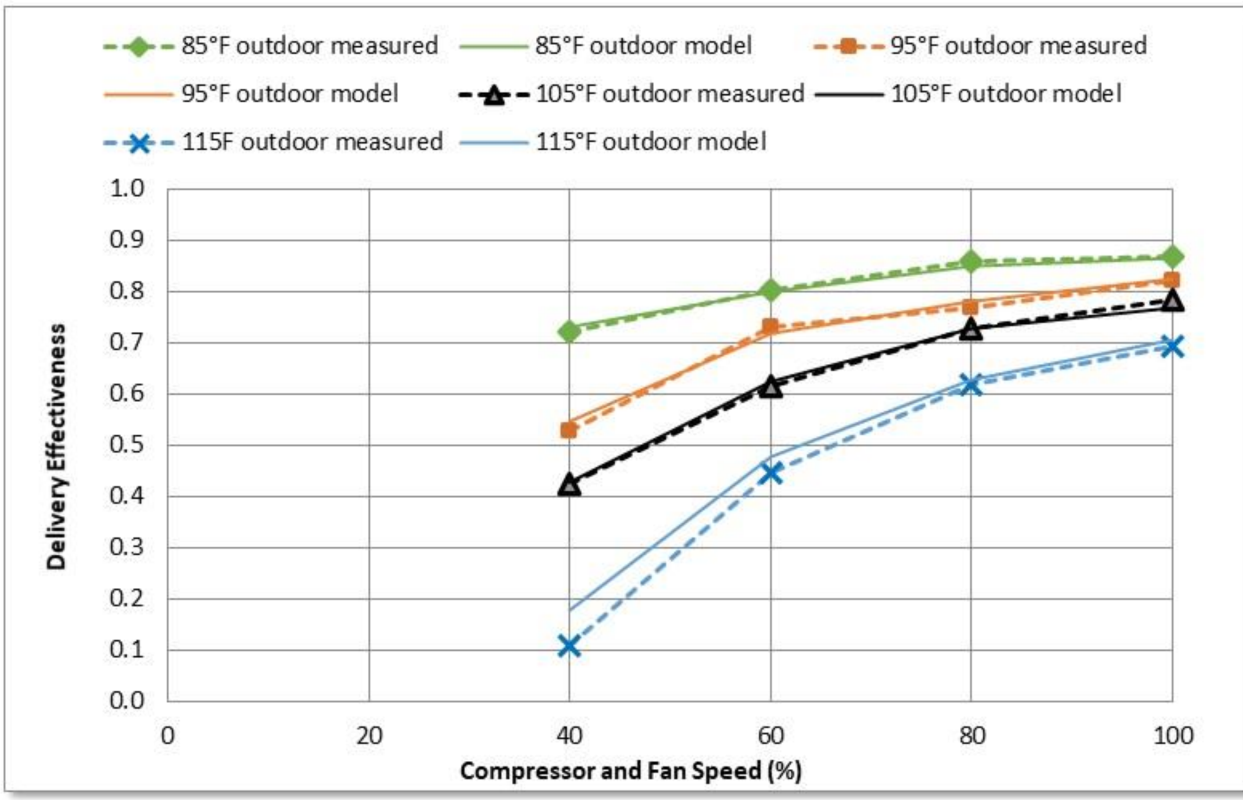
per the heat exchanger model ($\eta = e^{\frac{-UA}{\dot{m}C_p}}$). This partially explains why all grilles do not see the same delivery effectiveness.

The overall delivery effectiveness can be obtained by calculating the delivery effectiveness at each grille using Equation 23 and then obtaining the flow-weighted average of those quantities.

Comparison of Model Results with Experimental Data

The modeled results for delivery effectiveness were compared to laboratory measurements where the compressor and fan speeds were synchronized at the Western Climate Performance Mapping indoor conditions (Figure 48) and is plotted in Figure 56. The model agrees with the measured values within ±3 percent except for the lowest speed setting at 115°F (46°C) outdoor temperature.

Figure 56: Comparison of Delivery Effectiveness vs Compressor/Fan Speed (%) - 75°F DB/62.5°F indoor condition and 115°F outdoor condition



Source: EPRI

Summary of Phase 1 Results

Phase 1 laboratory evaluation involved the assessment of the operation and performance of six technology features through a variety of steady-state and dynamic mode tests at three independent laboratories: EPRI, PG&E, WCEC. The key results are summarized as follows:

1. Variable capacity heat pumps (VCHP): variable capacity compressor and variable speed blower (EPRI, PG&E, WCEC):

a. Enables 22-32 percent cooling energy savings across California climate zones compared to a 14 SEER single speed system as a baseline.

b. The Next-Gen RSCS can provide over 90 percent of the annual heating load without requiring backup heating for most of the of the 16 California climate zones modeled.

c. Cooling and heating part-load efficiencies are better than full-load efficiencies at mild temperatures (between 35°F (2°C) and 90°F (32°C)).

d. Higher part-load efficiency was found with the Next-Gen RSCS which corresponds to higher SEER/HSPF values.

e. The Next-Gen RSCS is able to modulate and match well with imposed dynamic load.

2. Integrated ventilation control (EPRI):

a. The use of a heat recovery ventilator with the VCHP provides an additional 1-4 percent cooling savings compared to a baseline SEER 14 fixed-speed HVAC system for California Climate Zone 10 (cooling design condition of 101°F (38°C) and a heating design condition of 35°F (2°C)).

b. Modeling results for the heating season showed that the capacity of the Next-Gen RSCS system (without backup) could be increased by around 1 percent in cooler California Climate Zones (Zones 1,2, 11, 12, 13, 14, 15 &16) when using an HRV.

3. Automated demand response (ADR) (EPRI, PG&E):

c. Demonstrated VCHP's capability as a flexible demand response resource

d. During ADR events, VCHP's capacity reduced in a non-linear relation with reduced load.

Table 19: Summary of ADR Test Results

ADR Speed Setting	Achieved Demand Savings	Sensible Capacity Reduction
50%	1.19 kW (65%)	36%
30%	1.37 kW (75%)	49%
0%	1.77 kW (95%)	58%

Source: EPRI

4. Dual fuel (intelligent heating) (PG&E):

a. The economics of the dual fuel heat pump depend on the outdoor temperature and the prevailing gas and electric rates. Substantial heating season cost savings are possible using a dual fuel heat pump compared to a high efficiency gas furnace when using favorable electricity rates. Calculations were performed to assess the economics of dual fuel heat pumps in selected locations in California.

California energy prices tend to be high relative to National averages and natural gas prices have not kept pace with electric prices. Despite these factors there are still situations, as illustrated in Appendix C, where the Next Gen Dual Fuel Heat Pump (DFHP) can provide attractive savings.

b. The fact that operation of the DFHP can be adjusted as utility prices vary permits the homeowner the option to benefit from future changes in utility prices that might reduce the ratio of electricity to gas prices in the future. The assurance that the homeowner will be able to experience the lowest future heating costs possible, is an important attribute of dual fuel heat pump capability and increases the value of this feature to potential purchasers of the Next Gen Residential Space Conditioning System.

c. Laboratory testing confirmed the functionality of the dual fuel heat pump concept in all possible modes of operation.

5. Duct-loss assessment for single-zone configuration (WCEC):

The single-zone operation of a variable-capacity/variable-fan residential air-conditioner utilizing ductwork routed through an attic was studied experimentally at the UC Davis Western Cooling Efficiency Center laboratory. The objective was to determine the optimum operating speeds for both the compressor and indoor fan for achieving maximum system efficiency in hot and dry California climates. The data collected describes the performance characteristics of the system operating under three different modes—a) varying compressor speed and indoor fan speed together, b) varying indoor fan speed while holding compressor speed fixed and c) transient operation of the equipment. The results highlight the potential for variable-capacity/variable-speed cooling systems to reduce residential energy use in California and corroborates the proposal to add these technologies to the portfolio of advanced efficiency solutions to be used for maximum energy efficiency. The major results of the laboratory tests are briefly summarized below:

a. VCHP connected to a duct system shows significant part-load energy-saving potential. For example, when the compressor and fan speeds are synced with each other, the variable-capacity/variable-speed blower system performs at maximum system efficiency at a speed setting of 60 percent of the rated speed when the outdoor dry-bulb temperature is 85°F (29^{o}C). This speed setting would need to increase to 80 percent as the outdoor temperature gets warmer than 85°F (29^{o}C) and further increase to 100 percent speed setting at outdoor temperatures of 115°F (46^{o}C) or higher.

b. System COP (including the ducts) is less sensitive to variations in the compressor and fan settings.

c. For compressor speeds at or below 80 percent of the rated speed, operating the system at 80 percent of the rated airflow yields maximum system efficiency when temperatures are hot. The countering effects between the increased delivery effectiveness and compressor efficiencies at higher airflow rates and the increased fan power use is what determines this balanced setting of evaporator airflow rate.

d. For the same outdoor temperature, the compressor and fan can be operated at lower speed settings when the indoor wet-bulb temperature is greater because the equipment is doing more dehumidification. Since the duct losses occur through sensible heat gains, a lower percentage of total cooling produced is lost through the ducts. The result also implies that duct losses play a more significant role in the hot and dry California climates compared to hot and humid climates. Adding more insulation to ducts, or keeping at least a portion of them in conditioned.

e. The efficiencies of the system measured at different grilles of a home change disproportionately as the compressor speeds and airflow rates through the equipment are varied. The grilles with lower airflow rates suffer the greatest efficiency losses as the total airflow rate through the system is reduced. Depending on the flow rate, these grilles may even have negative efficiencies. This result will be extremely useful when extending the current technology being studied to multi-zone applications.

f. Ducting impacts system efficiency at hotter temperatures. For the same outdoor temperature, the compressor and fan can be operated at lower speed settings when the indoor wet-bulb temperature is greater because the equipment is doing more dehumidification. Since the duct losses occur through sensible heat gains, a lower percentage of total cooling produced is lost through the ducts. The result also implies that duct losses play a more significant role in the hot and dry California climates compared to hot and humid climates. Adding more insulation to ducts, or keeping at least a portion of them in conditioned spaces will render the variable-capacity/variable-speed cooling technologies more beneficial*.*

g. Mathematical model of the system with ducts agrees with ±3 error between predicted and observed delivery effectiveness. The largest error occurs when ducting delivery effectiveness is very low, and the Next-Gen RSCS should therefore never be allowed to run under those conditions.

Preliminary Project Benefits Based on Phase 1 Results

Per California's Energy Efficiency Strategic Plan, HVAC is the single largest contributor to peak power demand in the state, comprising up to 30 percent of total demand in the hot summer months. The next-generation space conditioning system's combined technologies could appreciably reduce peak demand. Variable-capacity systems have the unique attribute of going to a state of higher operating efficiency when the compressor speed is reduced. For a Demand Response event, a reduction in compressor speed provides a reduction in power draw, but with a correspondingly smaller reduction in cooling capacity. According to the Strategic Plan, the CEC estimates that a peak demand reduction of 1,096 MW could be achieved through high-quality HVAC installations by 2020. If next-generation air conditioners, or similar technology, were adopted by California energy codes, the potential energy savings is on the order of 5 times greater, or 3.62 GWh per year with a rough energy cost value of greater than $5 Billion over the equipment lifetime.[3] This will benefit the ratepayers through avoided electric capacity and energy costs, providing greater reliability, lower costs, and increased reliability for California Investor-Owned Utility (IOU) electricity ratepayers. Highlights of the preliminary benefits of this project based on the Phase 1 laboratory evaluation are:

- Variable capacity heat pump (VCHP) performs at higher system efficiency when operating at lower speed settings instead of rated levels.
- VCHP with demand response capability enables utilities to reduce peak demand.
- More efficient control strategies are needed for heat pumps connected to ductwork located in an attic for hot and dry California climate zones. The system balance is affected by the duct-zone temperatures, which invites the need for revising ducting standards.
- The next-generation VCHP has demonstrated its versatility with intelligent heating capability and integrated ventilation configuration.

[3] 1 PPEU = 100MW peak generation. Generation = load + 15% T&D losses + 15% reserve margins.

CHAPTER 3: Phase 2 Laboratory Evaluation (Task 3)

Phase 2 of this project consisted of evaluation of alternative refrigerants at PG&E and EPRI facilities, testing of fault detection and diagnostics at PG&E, investigation of zonal control at EPRI and assessment of duct losses in multi-zone systems at WCEC. The following sections describe the features tested and the results obtained. Each of the three labs used the same model 2-ton ducted split-system air-source heat pump unit from Daikin/Goodman described in Chapter 2.

Phase 2 Technology Features Evaluation

In the Phase 2 evaluation, some of the tests were conducted with R-410A and some of the tests were conducted with R-32 as the refrigerant as shown in Table 20.

Table 20: Phase 2 Technology Testing

Technology	Laboratory/Refrigerant		
	EPRI	**PG&E**	**WCEC**
Alternative Refrigerants	R-32	R-32	
Zonal Control	R-32		
Multizone with Duct Losses			R-410A
Fault Detection & Diagnostics		R-410A	

Source: EPRI

Both EPRI and PG&E evaluated the use of R-32, conducting tests similar to those run with R-410A in Phase 1, evaluating the steady state and dynamic features of the variable capacity features of the heat pump system (variable-capacity compressor and variable-speed blower) and comparing results to those obtained with R-410A. Additionally, EPRI performed analysis and testing of zonal control with a variable capacity heat pump, by evaluating a two-zone residential configuration using R-32 as the refrigerant. PG&E performed a laboratory evaluation of the Fault Detection and Diagnostic capability of the variable capacity heat pump control system using R-410A refrigerant. Extensive adjustments were made in system settings and test conditions in order to trigger a wide range of faults and corresponding alerts. WCEC performed experimental and analytical studies of duct loss assessments for a multizone system using the variable capacity heat pump system in a residential building, comparing results to those obtained with a single-zone configuration in Phase 1.

The following sections describe the characteristics, operation and testing of the Phase 2 features evaluated for the Next-Generation variable speed space conditioning system under dynamic and steady state conditions: Alternative Refrigerants, Zonal Control, Multizone with Duct Losses, and Fault Detection & Diagnostics.

Alternative Refrigerants in Residential Space Conditioning

Traditional refrigerants utilized in HVAC equipment exhibit properties that can cause environmental concerns. Two common metrics used to characterize the environmental impact of substances are ozone depletion potential (ODP) and global warming potential (GWP). The Montreal Protocol in effect since 1989 called for the elimination of ozone depleting refrigerants from commercial use. These included substances that contain chlorine or bromine, leaving mainly compounds that contain fluorine for use as refrigerants since these do not affect the ozone layer.

In recent years, domestic and international legislators have required the phase-out of traditional refrigerants with non-zero ODP and high GWP. In California, the California Air Resources Board has established low GWP targets for new refrigeration and HVAC equipment and GWP limits on the sale and distribution of refrigerant.

In response to environmental concerns and legislation, the HVAC community has been steadily investigating alternative Low GWP refrigerants for various HVAC applications and equipment types. The clear majority of alternative refrigerants identified to date have shortcomings in HVAC performance, safety, or applicability. HVAC performance refers to the nominal capacity or efficiency of the refrigerant, safety refers to the refrigerants toxify and flammability, and applicability refers to the usage of the refrigerant for differing HVAC applications.

Currently in residential HVAC equipment, the most common refrigerants used are R-22 and R-410A. R-22 exhibits an ODP of 0.04 and GWP of 1,800 while R-410A has zero ODP and a GWP of 2,100. R-410A refrigerant, which is a zeotropic blend of 50/50 percent by weight of refrigerants R-32 (difluoromethane) and R-125 (pentafluoroethane). It has a GWP rating of 2,100, mainly because of the R-125, which by itself carries a GWP rating of 3400. The GWP of R-32 in contrast is only around 675; one of the lowest of all HFCs (ASHRAE, 2013). The main reason for blending the two is that by itself, R-32 is classified as an A2L refrigerant, which means it is slightly flammable. The addition of the R-125 into the blend effectively eliminates the flammability of the blend and makes it safer to handle.

Characteristics of R-32 in Heating, Ventilation, and Air Conditioning Equipment

The refrigerant typically used in the past for heat pumps, R-22, is being phased out because of its ozone depletion potential (ODP). The most common alternative to R-22 in residential space conditioning applications is R-410A, a blend of R-32 and R-125. While R-410A has zero ODP, it has a higher global warming potential (GWP) than R-22. To reduce GWP, replacements for R-410A are being sought, where R-32 is one of the most promising replacement options with a GWP of 650, albeit with mild flammability (ASHRAE Fundamentals, 2013). The ASHRAE safety classification for R-32 is an A2L refrigerant. "A" refers to the toxicity of R-32 which is low toxicity, while "2L" refers to the flammability of R-32 as mildly flammable.

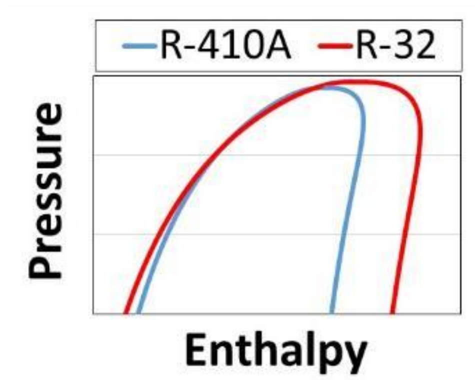
R-32 also offers potential performance improvements compared to R-410A. Phase 2 testing investigated the use of R-32 for improving performance and environmental impact of the Next-Gen RSCS. Several investigators have found that R-32 can provide energy performance advantages over R-410A. Work at Daikin showed that R-32 improved the System COP by 5 percent at 95°F (35^{o}C) and 8 percent at 131°F (55^{o}C).

Research work organized by Oak Ridge National Laboratory, under the AHRI AREP (Air-Conditioning, Heating, and Refrigeration Institute Alternative Refrigerants Evaluation Program), tested R-410A alternatives. R-32 appeared to be the most promising one. One AREP study, conducted at Lennox (Crawford and Uselton, 2013), tested a unit optimized for operation with R-410A. Results showed comparable SEERs for R-32 and R-410A with higher HSPF, similar heating capacity, slightly lower cooling efficiency and higher cooling capacity for R-32. Another AREP study, conducted at Goodman (Hao and By, 2012), following AHRI 210-240 and ASHRAE 37, showed a 1.3 percent improvement for R-32 compared to R-410A for Cooling Test A, a 1.2 percent improvement for Cooling Test B and a 2.2 percent improvement for Heating Test H1.

Earlier testing work at the University of Maryland (Xu, Hwang, Radermacher, 2012) found that R-32 increased cooling capacity between 3.4 and 9.7 percent, and COP increased between 2.0 and 9.0 percent, compared to an identical conventional vapor compression cycle using R-410A. R-32 was deemed an excellent alternative to R-410A, and the system with R-32 was thought to be capable of further enhancement via component optimization.

Figure 57 provides a comparison of R-410A and R-32 for their pressure – enthalpy diagrams, which illustrates that R-32 has an expanded two-phase region. Industry observations comparing R-32 and R-410A include: 20 to 30 percent charge reduction with R-32, ~10 pounds per square inch (PSI) increase in operating pressure with R-32, and +10°F (+5.6°C) increase in compressor discharge temperature.

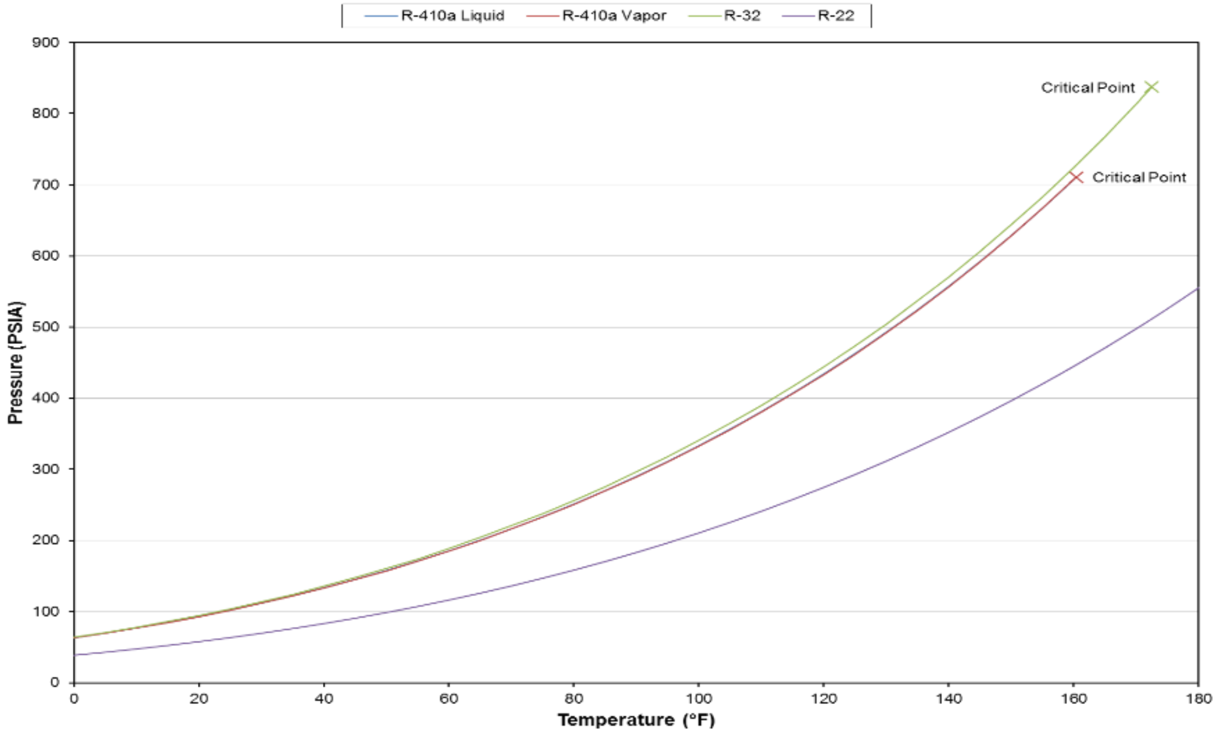
Figure 57: Pressure–Enthalpy Diagram of R-410A and R-32



Source: EPRI

Compared with the R-410A blend, R-32 has only a 1-2 percent higher saturation pressure as a function of temperature, as shown in Figure 58 (which are both much higher than R-22—included for reference—which was the primary refrigerant for air conditioning before the Montreal Protocol).

Figure 58: Saturation Properties of R-410A vs R-32



Source: EPRI

The performance of R-32 has been investigated in split HVAC equipment in both heating and cooling operation for certain residential equipment types. Table 21 provides the details of three previous laboratory studies evaluating the performance of R-32 in comparison to R-410A, which were identified in the AHRI Low-GWP Alternative Refrigerants Evaluation Program. Each study was unique with differences in system type, nominal size, and optimization method. The two possibilities for optimization of the HVAC unit were charge quantity and the setting of the expansion device. The indoor condition for these identified studies was maintained at approximately 80°F (27°C) and 50 percent relative humidity.

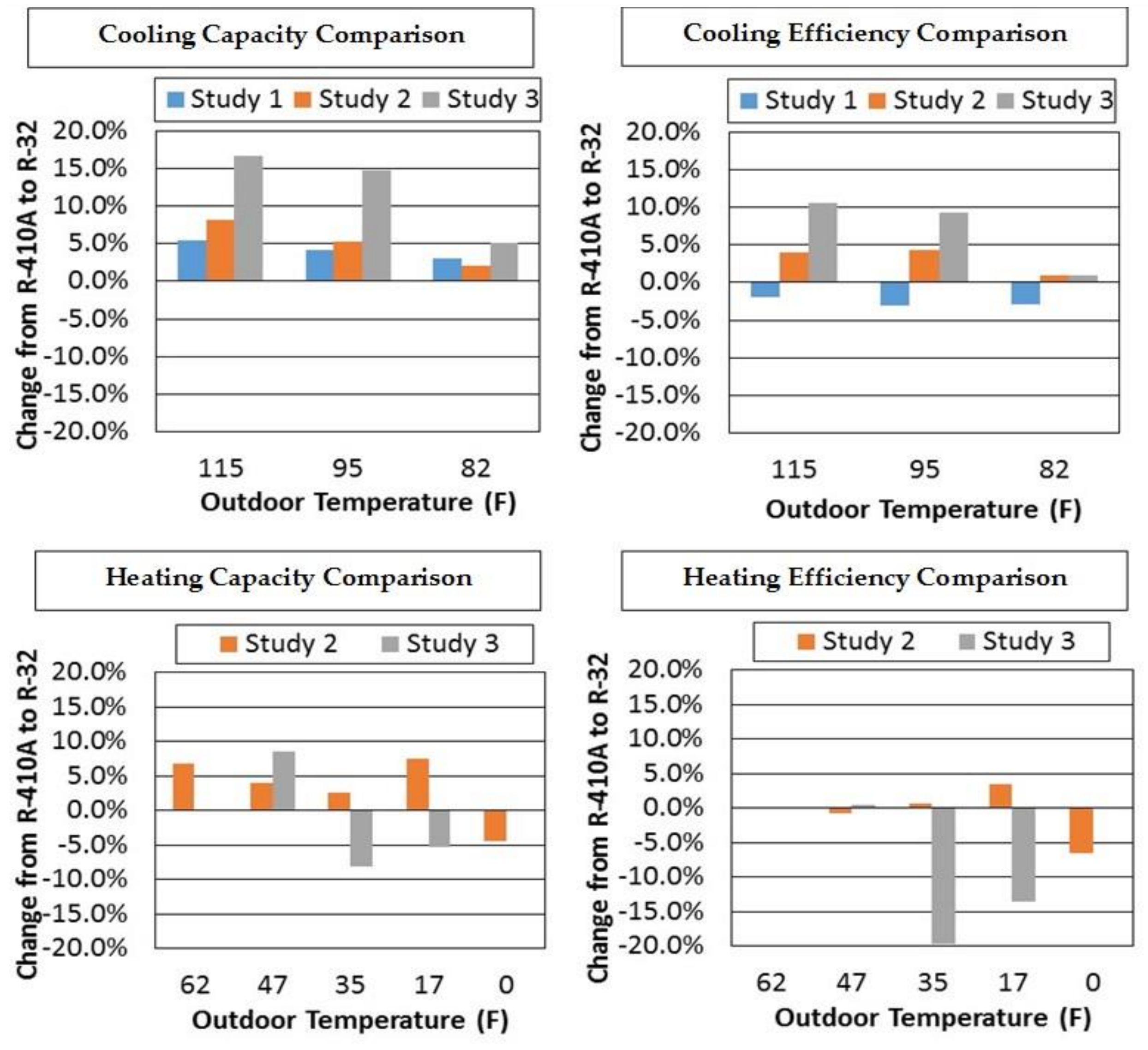
Table 21: Previous Evaluations of R-32 in Split HVAC Equipment

Study	Laboratory	System Type	Nominal Size (tons)	System Optimization
1	Oak Ridge National Laboratory	Ductless Mini-Split Air-Conditioner	1.5	Charge Quantity; Expansion Device
2	University of Maryland – Center for Environmental Energy Engineering	1-Speed Ducted Split Heat Pump	3.0	Charge Quantity; Expansion Device
3	Danfoss Laboratories	1-Speed Ducted Split Heat Pump	2.5	Charge Quantity

Source: EPRI

Table 21 and Figure 59 provide results of the laboratory comparisons of R-410A and R-32 in split HVAC equipment identified in the literature. The top two graphs provide a cooling performance comparison, while the bottom two graphs compare heating performance. The graphs on the left-side compare unit capacity, while the right-side compares the unit efficiency. Each graph provides the percent change from the R-410A baseline to R-32. For cooling operation, a trend of increased cooling performance with increasing outdoor temperature is clearly observed. The most improved cooling capacity and cooling efficiency from these previous studies occurred at an outdoor temperature of 115°F (46°C). In comparing heating operation of the previous studies, the performance of R-32 in general appears to be similar or improved at milder outdoor temperatures (62°F [17°C] and 47°F [8°C]) and similar or reduced at colder outdoor conditions (35°F [2°C], 17°F [–8°C], and 0°F [–18°C]).

Figure 59: Previous Laboratory Comparisons of R-32 to R-410 in Split Systems



Source: EPRI

In summary, using R-32 as an alternative to R-410A appears to offer some performance improvement potential and clear environmental benefits with lower GWP. The biggest downside of using R-32 is its flammability. Consumer concerns and pending global actions with environmental issues may be important enough to warrant favoring R-32 compared to R-410A. Performance improvements are also attributes that could enhance residential customer acceptance. These advantages must be traded-off against the fear of flammability and corresponding safety and insurance issues.

Zonal Control in Residential Space Conditioning

Zoning in HVAC refers to a system that separates a greater conditioned space into smaller zones all conditioned by a single HVAC system. Zoning is typically achieved with a thermostat or unit controller in each zone, which controls a damper within the ductwork for the zone. In residential applications, zoning may be implemented for the upstairs and downstairs of a two-story home or for a large room of varying occupancy, such as a bedroom. When zoning is implemented with a variable speed indoor blower, the indoor airflow can be modulated to match the load and adjust to the number of zones calling for space conditioning.

Multiple field studies on HVAC zoning were identified in the Codes and Standards Enhancement Initiative: Residential Zoned Ducted HVAC Systems for the 2013 California Building Energy Efficiency Standards. These studies highlighted that the efficiency impact of zoning with a ducted HVAC system may be dependent upon HVAC equipment type, ductwork layout, zoning operation, and the indoor temperature offset of unoccupied zones. When indoor temperature offsets were not utilized for unoccupied zones, increased HVAC energy usage was generally observed in the identified field studies. This trend likely occurs due to the altered ducting and operation of a zoned HVAC system compared to non-zoned central systems. Across the seven identified studies, energy reductions ranged from 1 percent to 29 percent and energy increases ranged from 6 percent to 48 percent.

Variable Capacity System Connected to a Ductwork System in a Multi-Zone Configuration

While variable speed equipment has been shown to achieve very high efficiencies, it is not clear how these systems perform when connected to a standard duct system in the attic which can often reach over 120 degrees in the summer. Unlike most commercial air-conditioning systems, the ductwork for most residential systems in the United States (particularly most homes in the sunbelt regions) are installed outside conditioned spaces (Stephens et al., 2011; Siegel and Walker, 2003; Modera, 1993) including attics that are generally warmer than outdoor air conditions during summer (Parker and Sherwin, 1998). According to (Lstiburek, 2013), “Attic air temperatures during the day in the summer pretty much everywhere where people live get up to the 120°F to 130°F (49°C to 54°C) range.” Sealed and insulated attics are much cooler, with temperatures 5–6°C warmer than the house in hot-dry climate regions (Less et al., 2016).

With the large loads imposed on ductwork running through an attic, combined with a longer residence time of the conditioned air in the ducts due to part-load cooling, delivery effectiveness (the ratio of cooling provided by the system (equipment + ductwork) to the ratio of cooling provided at the equipment) and therefore the overall COP of the system will be reduced (Chapter 2). Current performance standards for residential systems do not address the relationship between the air conditioning system and the ductwork. With the introduction of variable speed systems, questions regarding their efficiency at part-load will need to be resolved to properly determine their efficacy, and to create alternative control strategies for optimizing performance.

Zonal control (maintaining individual temperature set-points in different zones of a building) is another strategy that has been emphasized to optimize total HVAC energy cost (Gupta et al., 2016). While this strategy is commonplace in most commercial buildings in the United States (F. Jazizadeh, et al., 2014), most single-family houses in the United States have HVAC systems typically controlled by a single, centrally located thermostat (Alles, 2006). There has been some interest in developing airflow control strategies for zone-based temperature control in multi-zone residential systems (Foster et al., 1993) but their impact on the efficiency of the air-conditioning system is not well understood, especially when the duct-losses are included. Each zone may have supply ducts with multiple branches, thus the available surface area for heat gain from the attic will be different for each zone. This implies that the same flow rate through each supply trunk will render different delivery efficiencies through the ducts. Phase 2 of this project assessed the duct losses associated with the space-conditioning system operating in a multi-zone ducted application, with a focus on the effects of variable capacity and airflow on duct thermal losses and delivered capacity in multi-zone operation.

Fault Detection and Diagnostics

Fault detection and diagnostics (FDD) monitor HVAC system operation to detect degradation in performance (indicative of the need for maintenance or repairs), and trigger diagnostics for technicians/service personnel to remedy system issues. Methods vary on what points to measure and how to interpret the measurements. Typical methods include direct measurement of temperatures, pressures, and electric power, to identify faults relating to air flow, refrigerant charge, sensor malfunctions, fan, blower and compressor degradation, and other malfunctioning components. Early detection of degradation triggering maintenance alerts and corresponding responses can prevent system malfunctions and failures. This minimizes wasted energy and loss of functionality, and assures the occupant that system reliability and performance are optimized.

Units typically have an interface for the homeowner and for a contractor; they both can log in and see how their system has been running. E-mails could be sent alerting the user or contractor of the issues/faults along with instructions for their resolution. To save time, model number and problem information can be sent to the servicing technician before the visit, permitting them to bring the parts likely to be needed for the service call.

Opinions differ on the efficacy of FDD technology. A report issued in May 2016 (Springer 2016) cites presentations and discussions on the advantages and disadvantages of HVAC Fault detection, diagnosis, and repair/replacement:

- Mowris cited common installation and maintenance conditions that can reduce EER by 10 percent to 55 percent. Some of these problems and degradation found in the study were: low air flow (EER reduced by 3 percent to 12 percent), evaporator blockage (EER reduced by 4 percent to 7 percent), duct losses (EER reduced by 7 percent to 36 percent), refrigerant overcharge (EER reduced by 4 percent to 17 percent), and refrigerant undercharge (EER reduced by 4 percent to 56 percent). Maintenance practices are cited and the need for service and equipment improvements noted.

- Proctor looked at the effects of refrigerant charge level, failure to remove nitrogen from the refrigerant line, airflow restrictions and duct leakage as major causes of system performance degradation. A 50 percent reduction in airflow reduced EER by 25 percent. Refrigerant charge that is 70 percent of the recommended charge reduced EER by about 45 percent.
- Also in the Springer, 2016 report, others indicated that field evaluation and repair of refrigerant charge can be detrimental to system performance. Iain Walker suggested that it may be wiser to ensure that systems are correctly charged initially and leave them alone. Field studies have shown that a sample of systems that were routinely serviced had more refrigerant charge defects and problems with non-condensables than a sample that had no maintenance. Robert Mowris stated that "Every tech introduces non-condensables," and their presence is difficult to diagnose or measure. There have been multiple studies evaluating potential and realized energy savings, many of which are referenced in a document prepared to support California Title 24 standards for refrigerant charge testing (CEC, 2013 California Building Energy Efficiency Standards). Beyond this information, there is little data on the cost or cost-effectiveness of FDD and maintenance programs besides what has been determined through utility program evaluations, which so far have been uncertain or mostly unfavorable (Hunt, Heinemeyer, and Hoeschele, 2010). New programs such as PG&E's Quality Care program may provide better information on the relationship between maintenance cost and savings on a large scale, which is badly needed.
- Purdue University developed and exercised a method for testing several FDD protocols used in public utility sponsored efficiency programs (Braun and Yuill, 2014). Six results were possible in the testing: No Response, Correct, False, Alarm Misdiagnosis, Missed Detection, and No Diagnosis. Results were surprisingly poor. Protocols suffered from very high False Alarm rates (60-100 percent overall, with most categories over 95 percent), high Misdiagnosis rates and high No Diagnosis rates. The Missed Detection rates were low, suggesting that the protocols may be too sensitive. FDD provides no benefits if faults are not addressed (correctly). Handheld FDD is a tool intended to help maintenance personnel perform better service than they could with other methods. If they experience and identify False Alarms, Missed Detections, Misdiagnoses and No Diagnosis cases, it seems probable that they'll soon abandon diagnostics, or ignore them if FDD use is mandated.
- Mowris (Mowris, Jones, Eshom, 2012) presented laboratory test results of a new 3-ton split-system 13-SEER air conditioner using R-22 refrigerant. The combination of multiple faults such as low airflow, undercharge, duct leakage, and condenser coil blockage reduced EER by 54 percent and SEER by 67 percent.

As indicated by the aforementioned studies, the potential benefits of timely detection and correction of faults can provide substantial energy savings and improvements in functionality. It remains to be seen whether the FDD devices and algorithms currently on the market and the procedures used to correct detected faults provide appropriate sensitivity to be of benefit to users.

Electric Power Research Institute Thermal Lab Activities and Results

Chapter 2 described the EPRI Thermal Testing Laboratory located at EPRI's Knoxville, Tennessee facilities and the laboratory setup (Figure 2 and Figure 3) and instrumentation (Table 3) utilized for the heat pump evaluations conducted in Phase 2 of the study.

Based on the experimental measurements recorded within the laboratory setup, air-side capacity was determined for each steady-state test. Air-side capacity and corresponding air-side unit efficiency were used as the primary means of evaluating the unit performance. Refrigerant-side capacity was used to verify the air-side capacity calculation. The following equations, repeated from Chapter 2, provide the calculation of air-side capacity and efficiency, presented as energy efficiency ratio:

$$\text{Capacity (Air)} \left(\frac{\text{Btu}}{\text{h}}\right) = \text{ Air Mass Flow } \left(\frac{\text{lbm}}{\text{h}}\right) \text{x (Return Air Enthaply} - \text{Supply Air Enthalpy)} \left(\frac{\text{Btu}}{\text{lbm}}\right)$$

Equation 24

$$\text{Energy Efficiency Ratio } \left(\frac{\text{Btu}}{\text{Wh}}\right) = \frac{\text{Capacity } \left(\frac{\text{Btu}}{\text{h}}\right)}{\text{Total Power Consumption (W)}}$$

Equation 25

Table 22 provides a matrix of the cooling and heating test conditions which were imposed on the variable capacity heat pump. For both cooling and heating operation, the variable capacity system was operated at three fixed levels of output: maximum, intermediate, and minimum. The maximum and minimum conditions correspond with the maximum and minimum operating boundaries of the system, while the intermediate level falls between these two limits. Minimum and intermediate operation were predefined as approximately 30 percent and 50 percent respectively of maximum capacity, as informed by Daikin/Goodman. The indoor condition for cooling operation was a dry-bulb temperature of 75°F (24°C) and a wet-bulb of 63°F (17°C), while the heating operation tests were conducted at a dry-bulb temperature of 70°F (21°C) and a wet-bulb of 57°F (14°C). An external static pressure curve was assumed for evaluating the system under the different operating levels and indoor airflows. These indoor conditions were based on field measurements of studies by Parker and Proctor (2000).

Table 22: Test Matrix for Heat Pump Performance Evaluation

Mode of Operation	Cooling Operation	Heating Operation
Indoor Conditions (dry bulb/wet bulb) (˚F)	75/63	70/57
Outdoor Temperature (˚F)	65, 75, 85, 95, 105, 115	62, 47, 35, 25, 15
System Operating Level	Maximum, Intermediate, Minimum	Maximum, Intermediate, Minimum
Indoor External Static Pressure (in WC)	0.5 in WC at 800 CFM. Static Pressure followed assumed Indoor Resistance Curve.	

Source: EPRI

Zonal Control Evaluation

Zoning control was examined in the laboratory with the variable capacity heat pump as part of the Next-Generation Residential Space Conditioning assessment. This section describes the zoning equipment and the laboratory setup for the zoning evaluation. Table 23 provides a description of the zoning equipment including the zone control board, dampers, and the zone controllers. The zoning setup consisted of two zones, which were intended to split the conditioned air equally (50/50 split). The zoning control board and one of the two dampers are shown in Figure 60.

Table 23: Zoning Equipment in Laboratory Setup

Equipment	Description
Zone Control Board	Board could accommodate up to 3 zones. Compatible with communicating and 24V equipment.
Dampers	2 x Dampers of 10" diameter; Two position: Open / Close.
Zone Controllers (Thermostat)	2 x Unit controllers (1 for each zone) (Model CTK04)

Source: EPRI

Figure 60: Zoning Equipment Used in Lab Assessment

Source: EPRI

The laboratory setup for zoning consisted of the variable capacity heat pump, zoning equipment, and an insulated enclosure for simulating two zones. The two-zone enclosure was inside the thermal chamber (Figure 61). Within the zoning setup, the performance of the heat pump was monitored and the individual airflow to each zone was monitored. In this laboratory configuration, the zoned variable capacity system was operated in cooling under both steady-state and dynamic operation. *Under steady-state operation, the chamber maintained the indoor room condition. Under dynamic operation, the variable capacity heat pump maintained the chamber condition based on the setpoints of the two zone controllers.* In dynamic operation, electric resistance heaters in each zone were used to simulate a cooling load.

Figure 61: Laboratory Setup for Zoning Controls Evaluation

Source: EPRI

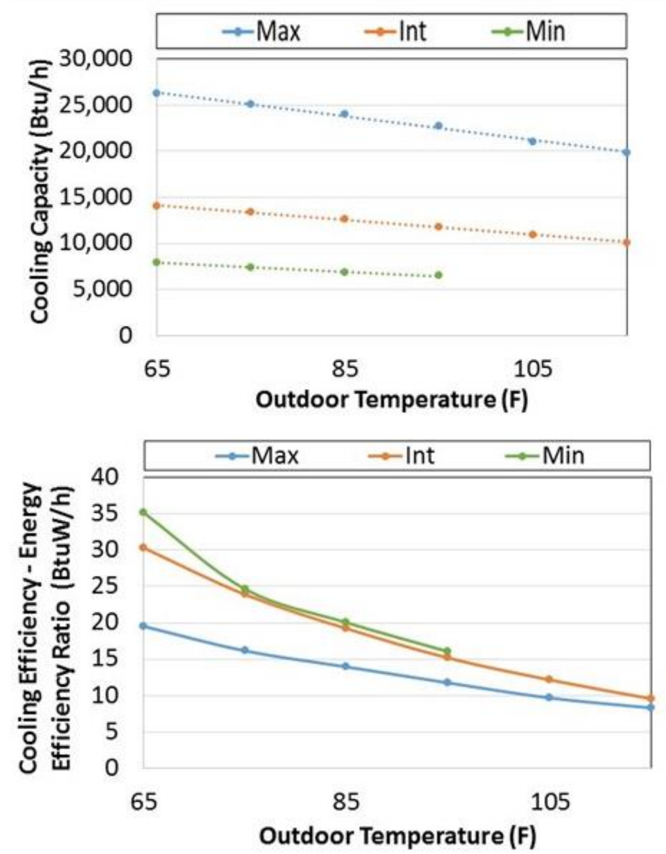
Results: Heat Pump Performance with R-32

This section provides the steady-state cooling and heating performance of the R-32 variable capacity heat pump based on the procedures outlined in the experimental setup for the EPRI laboratory.

Cooling Operation

The steady-state performance of the R-32 variable capacity system obtained in the EPRI laboratory is shown in Figure 62. The top portion of the figure provides the cooling capacity, while the bottom portion of the figure provides cooling efficiency, presented as the energy efficiency ratio. The retrofit of R-32 in the system presented no identified issues with the cooling operation of the variable capacity heat pump.

Figure 62: Steady-State Cooling Performance of R-32 VCHP



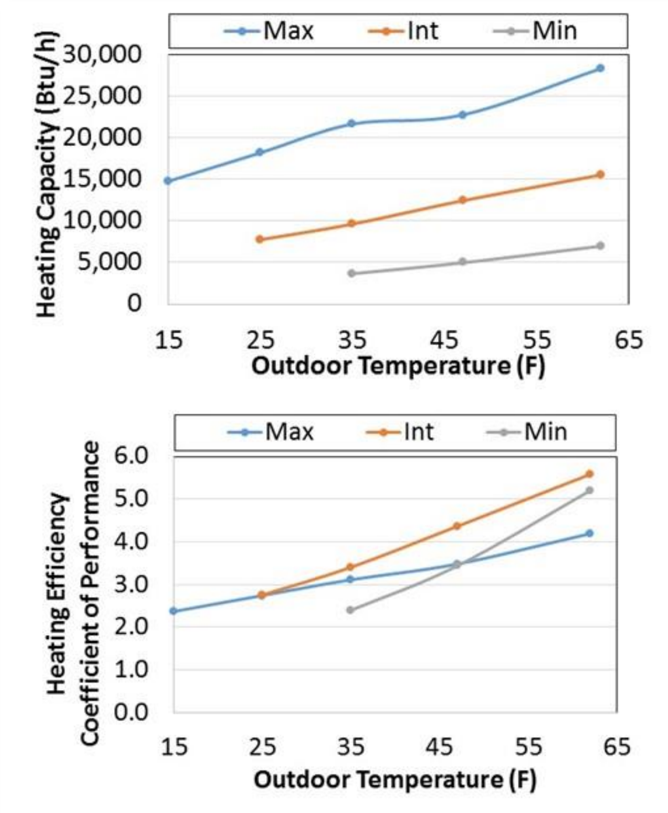
Source: EPRI

Trends observed in the cooling operation of the R-32 variable capacity heat pump are comparable to the trends observed for R-410A as the refrigerant. At 95°F (35°C) outdoor temperature, where nominal capacity is determined, the minimum output of the R-32 system was 29 percent of the maximum capacity. In R-410A testing of the variable capacity system, the minimum capacity was 30 percent of the maximum capacity at 95°F (35°C). The R-32 variable capacity system demonstrated increased efficiency at part-load operation, and the relative increase in efficiency from maximum to part-load operation increased with decreasing outdoor temperature.

Heating Operation

Figure 63 provides the heating performance of the variable capacity heat pump obtained through the experimental setup and procedures of the EPRI laboratory. This figure provides heating capacity in the upper portion of the figure and heating efficiency, presented as coefficient of performance, in the lower portion of the figure. The heating capacity values represent the steady-state heating performance, which do not account for defrost degradation at low outdoor temperatures.

Figure 63: Steady-State Heating Performance of R-32 VCHP



Source: EPRI

Operational and performance characteristics of the R-32 variable capacity heat pump in heating operation were comparable to the R-410A heat pump. The maximum heating capacity of the variable capacity heat pump decreased with decreasing outdoor temperature. The variable capacity heat pump demonstrated increased operating efficiency at intermediate heating operation. The relative increase in efficiency from maximum to intermediate operation increased with increasing outdoor temperature.

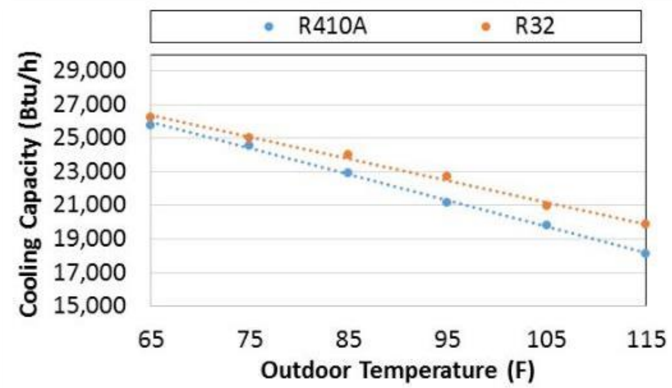
Comparison of R-410A and R-32 Heat Pump Performance

This section provides a performance comparison of the variable capacity heat pump for the two examined refrigerants, R-410A and R-32. The R-410A performance data was obtained in Phase 1 of the project, while R-32 was obtained in Phase 2 of the project. Both sets of performance data were collected utilizing the same individual variable capacity heat pump unit and identical experimental procedures. The refrigerant charge and TXV setting were optimized for each of the evaluated refrigerants. The performance comparison includes heating and cooling operation and operating capacity of the variable capacity system.

Cooling Operation

Figure 64 provides a comparison of the maximum cooling capacity curves for R-410A and R-32 in the variable capacity system. The figure illustrates a primary performance difference between R-410A and R-32 for cooling operation. The R-32 variable capacity system at maximum cooling operation demonstrated increased cooling capacity with increasing outdoor temperatures. The largest cooling efficiency increase from R-410A to R-32 was observed at 115°F (46°C).

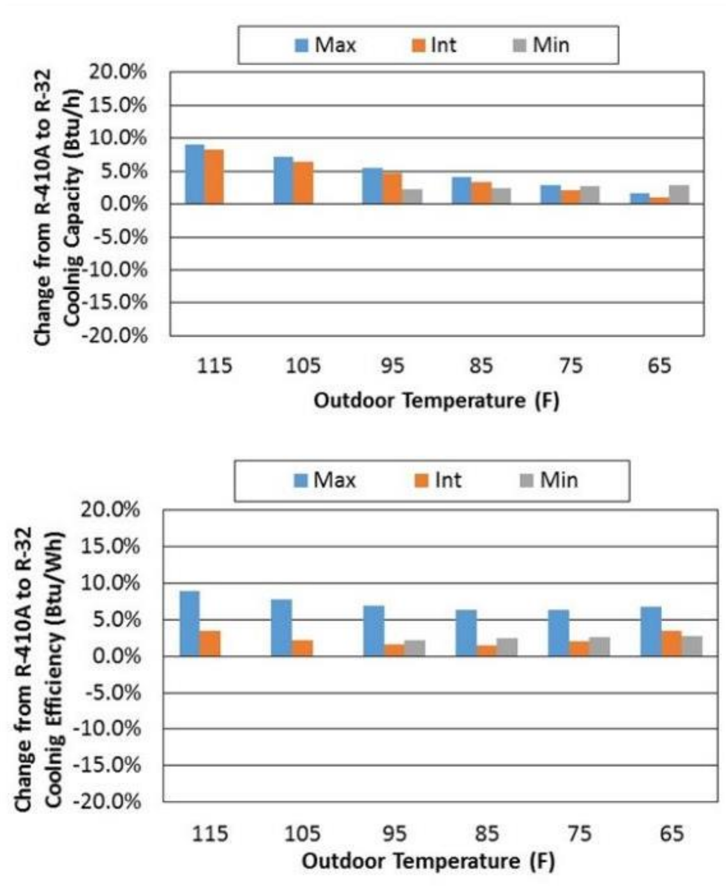
Figure 64: Maximum Cooling Capacity of R-410A and R-32 in VCHP Units



Source: EPRI

A detailed cooling performance comparison of R-410A to R-32 is provided in Figure 65.

Figure 65: Cooling Comparison of R-410A to R-32 in VCHPs



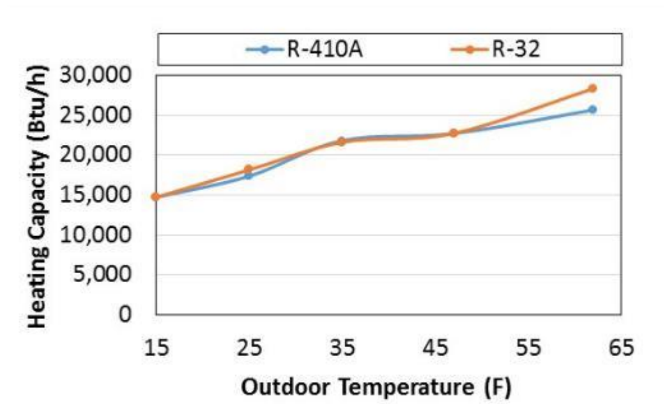
Source: EPRI

The figure shows the percent change from R-410A to R-32 for each outdoor temperature of the experimental plan and each operating level of the variable capacity heat pump. The upper portion of the figure provides a cooling capacity comparison, while the lower portion of the figure provides a cooling efficiency comparison of the examined refrigerants. For both maximum and intermediate operation, the trend of increased cooling capacity with increasing outdoor temperature was observed. At maximum operation, the cooling capacity from R-410A to R-32 increased by 6, 7, and 9 percent at 95°F (35°C), 105°F (41°C), and 115°F (46°C), respectively. The cooling capacity improvement at intermediate operation was observed to be slightly lower than occurred at maximum operation. At minimum operation, the cooling capacity increase with R-32 ranged from 2 percent to 3 percent across the examined outdoor conditions. In comparing cooling efficiency from R-410A to R-32, the largest improvements were observed at maximum operation. The retrofit of R-410A to R-32 resulted in cooling efficiency increases of 6to 9%, 1 to 3 percent, and 2 to 3 percent for maximum, intermediate, and minimum operation, respectively.

Heating Operation

Figure 66 compares the maximum heating capacity curves of the R-410A and R-32 variable capacity heat pump. The two curves are comparable with only two deviations at 62°F (17°C) and 25°F (–4°C). The R-32 system demonstrated a maximum heating capacity improvement of approximately 10 percent and 5 percent at 62°F (17°C) and 25°F (–4°C), respectively. Since the maximum heating capacity curves are comparable, R-32 and R-410A would have a similar ability to provide heating in the California climate. Both refrigerants provide near the nominal heating of two tons down to 35°F (2°C).

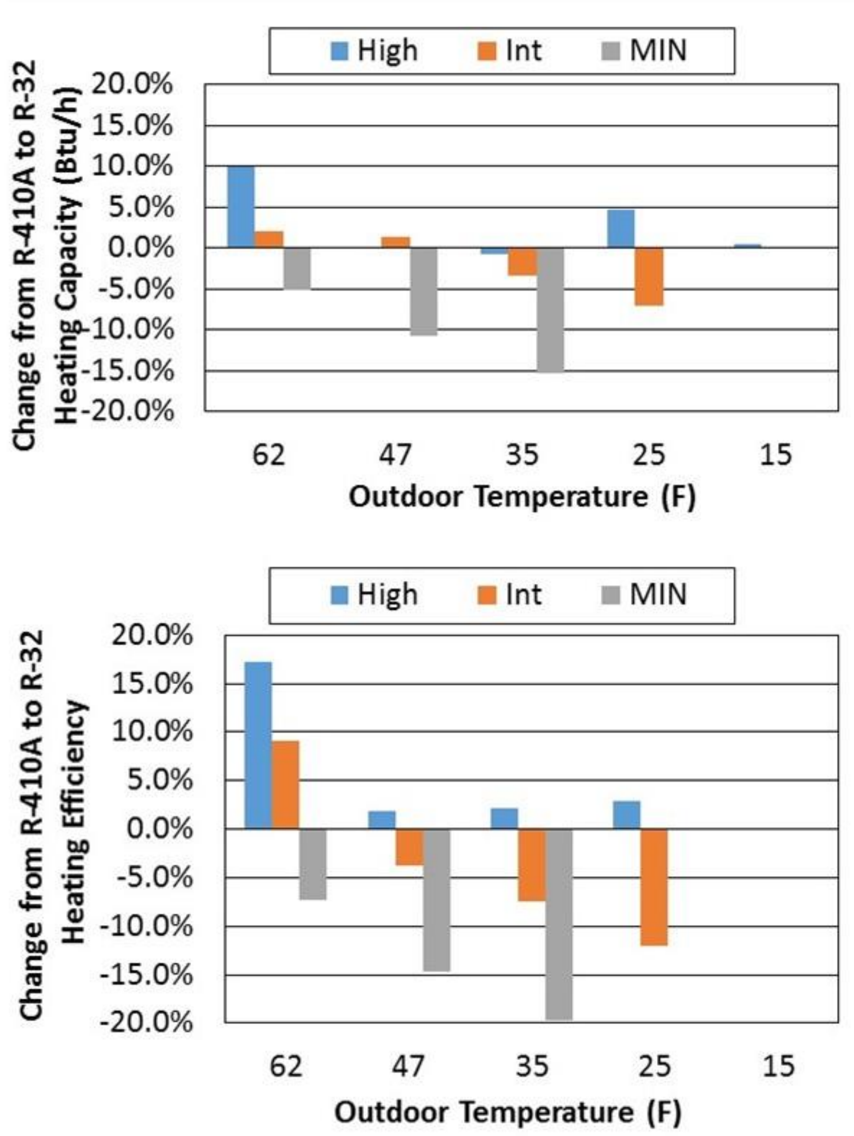
Figure 66: Comparison of Max Heating Capacity of R-410A and R-32 in VCHP Unit



Source: EPRI

Figure 67 provides a heating performance comparison of R-410A and R-32 in the variable capacity heat pump. The figure shows the percent change from R-410A to R-32 for the heating test conditions and operating levels of the variable capacity system. The upper portion of the figure compares heating capacity, while the lower portion of the figure compares heating efficiency. Through examining the heating capacity and efficiency comparison of R-410A and R-32, general trends may be observed. At maximum operation, the heating performance is generally comparable or improved with R-32 in the variable capacity heat pump. At intermediate and minimum operation, the heating performance is generally comparable or improved with R-410A as the refrigerant. In general, R-32 demonstrated comparable or improved performance at higher outdoor temperatures, while R-410A demonstrated improved heating performance at lower outdoor temperatures.

Figure 67: Heating Comparison of R-410A to R-32 in VCHP Systems



Source: EPRI

Effect of R-32 Heat Pump in California Climate

This section examines the potential impact of an R-32 variable capacity heat pump for the California climate. The section examines the annual cooling energy consumption and cooling peak power demand of the HVAC equipment studied. Due to the comparable performance of R-32 and R-410A in heating operation, a similar analysis for heating applications for California climate zones was not deemed necessary.

Cooling Energy Consumption

An energy model examining the annual performance of HVAC equipment was developed based on the bin-method described in ASHRAE Fundamentals. The model examined 15 of the 16 California climates where space cooling is common in residential applications. (Climate Zone 1 design cooling condition is less than the minimum outdoor temperature threshold for the energy model). The model examined the variable capacity system utilizing laboratory data and a baseline 14 SEER air-conditioner for comparison. In Phase 1 of the project, the model was utilized to examine the impact of the R-410A variable capacity system on annual space cooling consumption for the California climate. Typical Meteorological Year (TMY) data for a representative city was selected for each California climate zone to be utilized in the model. The model was expanded in Phase 2 to include the R-32 variable capacity system. Table 24 provides results of the energy model for an annual HVAC cooling consumption comparison by climate zone.

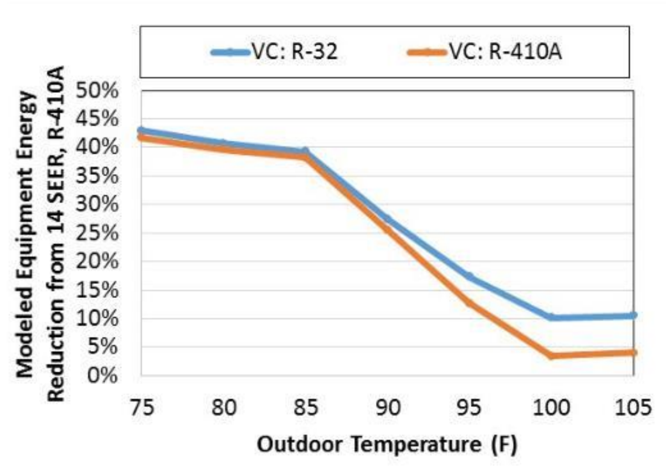
Table 24: Annual Cooling Consumption Comparison of R-32 VCHP Systems

California Climate Zone	Representative City	Annual Cooling Energy Savings Potential over Baseline 14 SEER	
		Variable Capacity Equipment (R-410A)	Variable Capacity Equipment (R-32)
1	Arcata	-	-
2	Napa	32.3%	+1.8%
3	Oakland	25.5%	+2.4%
4	San Jose	29.6%	+2.5%
5	Santa Maria	28.8%	+1.2%
6	Los Angeles	30.2%	+2.2%
7	San Diego	28.3%	+2.2%
8	Long Beach	29.9%	+3.0%
9	Burbank	29.7%	+2.7%
10	Riverside	30.3%	+2.2%
11	Red Bluff	28.5%	+1.7%
12	Stockton	28.6%	+2.5%
13	Fresno	28.2%	+2.6%
14	Twentynine Palms	25.7%	+2.8%
15	Blythe	22.4%	+2.7%
16	Bishop	28.2%	+2.5%

Source: EPRI

As previously conducted and discussed in Phase 1, the R-410A variable capacity equipment offers a potential cooling consumption improvement ranging from 22 percent to 2 percent for the California climate zones. The retrofit of R-32 into the variable capacity equipment offers an additional cooling savings potential ranging from 1.2 percent to 3.0 percent. In general, the R-32 variable capacity equipment demonstrated a greater impact in annual cooling savings in the warmer California climates. Figure 68 provides a detailed comparison of the R-410A and R-32 variable capacity system in the energy model for California Climate Zone 10. This figure provides the percent energy reduction from the baseline 14 SEER system by outdoor temperature. The outdoor design temperature used for Climate Zone 10 was 100°F (38°C). Figure 68 demonstrates that cooling energy savings from R-410A to R-32 in the variable capacity heat pump largely occur at the warmer outdoor temperatures in a climate zone.

Figure 68: Model Comparison of R-32 and R-410A for Climate Zone 10



Source: EPRI

Cooling Peak Demand

To examine the cooling peak performance of the R-410A and R-32 variable capacity heat pumps, laboratory cooling efficiency data at maximum operation was compared to a baseline system. Table 25 provides the peak cooling performance (kW / ton) of the R-410A and R-32 variable capacity heat pump along with a 14 SEER baseline system at three outdoor temperatures, 95°F (35°C), 105°F (41°C), and 115°F (46°C). A given space cooling system operating at maximum output would exhibit the shown efficiency levels at the given outdoor temperatures. With the implementation of R-32 in the variable capacity heat pump, the peak cooling performance improved by 6.7 percent to 8.2 percent with respect to R-410A. For residential equipment ranging from 2 to 4 tons, the R-410A variable capacity heat pump provides a potential peak reduction of 80–200W over a baseline 14 SEER system. Implementation of R-32 in the variable capacity heat pump provides an additional potential peak reduction of 125–475W depending upon size of the equipment.

Table 25: Peak Cooling Comparison of R-410A and R-32 in VCHP

Outdoor Temperature (F)	Peak Cooling Performance (kW/Ton)			Reduction from 14 SEER, R-410A	
	14 SEER R-410A	VCHP R-410A*	VCHP R-32*	VCHP R-410A	VCHP R-32
95	1.17	1.12	1.04	3.7%	10.4%
105	1.38	1.33	1.24	3.3%	10.3%
115	1.62	1.58	1.45	2.6%	10.8%

***Laboratory data from Phase 1 and Phase 2 of project**

Source: EPRI

Results: Zonal Control with Variable Capacity Heat Pump

This section examines the potential impact of integrating zoning with a variable capacity heat pump in the Next-Generation Space Conditioning System. The energy efficiency impact of zoning with variable capacity space conditioning is examined for a given application. In the laboratory setup, the functionality and operation of zoning controls were evaluated with the variable capacity heat pump.

Efficiency Effect of Integrating Zonal Control and Variable Capacity Heat Pump

The integration of zoning and variable capacity space conditioning allows for two energy savings mechanisms. First, zoning separates a conditioned space into zones and allows for reduced total load on the HVAC system. Reduced load allows for reduced operation or runtime of the HVAC system and thereby reduced energy consumption. Second, variable capacity equipment operates with increased efficiency under reduced load or part-load operation. This section aims to investigate the contribution of each energy savings mechanism in a zoned, variable capacity system.

Table 2826 provides example zoning scenarios with the cooling performance data obtained for the R-32 variable capacity heat pump. The example demonstrates the contribution of the two, potential energy saving mechanisms of a zoned variable capacity system: reduced load and increased system efficiency at part-load operation. The example in

Table 2826 assumes a baseline cooling load of 20,000 Btu/h at 95°F (35°C) outdoor temperature and 75°F (24°C) indoor temperature for the R-32 variable capacity system. Two load reduction cases are examined: 10 percent and 20 percent load reduction from the baseline. In a zoned system, the load on the HVAC system may reduce when an indoor temperature offset is used for unoccupied zones. For instance, a 5°F (2.8°C) temperature offset for a zone which is 50 percent of the overall conditioned space may result in a 10 percent reduction of the overall load on the HVAC system at 95°F (35°C).

As seen in the examples shown in Table 26 for a zoned variable capacity system, reduced load on the HVAC system corresponds to reduced power consumption and increased cooling efficiency.

Table 26: Example of Integrating Zoning and Variable Capacity Air-Conditioner

	Total Cooling Load on HVAC System		HVAC Power		HVAC Cooling Efficiency	
	(Btu/h)	% Reduction	(W)	% Reduction	(Btu/Wh)	% Improvement
Baseline	20,000	—	1,690	—	11.83	—
10% Load Reduction Due to Zoning	18,000	10%	1,473	12.8%	12.22	3.3%
20% Load Reduction Due to Zoning	16,000	20%	1,255	25.7%	12.75	7.8%

Source: EPRI

As load is reduced on an HVAC system, the runtime or operating level is reduced to match the new load. If the HVAC system maintained constant efficiency at reduced load, the power consumption of the HVAC unit would be proportional to the load reduction. For instance, a 10 percent load reduction would correspond with a 10 percent HVAC power reduction, and a 20 percent load reduction would correspond with a 20 percent HVAC power reduction. However, since cooling efficiency of the variable capacity equipment increases at part-load operation, a 10 percent load reduction results in an HVAC power reduction of 12.8 percent, while a 20 percent load reduction results in a 25.7 percent HVAC power reduction. The improved power reduction with reduced load occurs due to the increased cooling efficiency of the variable capacity system at part-load operation. Laboratory Assessment of Zonal Control for a Variable Capacity Heat Pump

In the laboratory setting, the two-zone setup was operated in conjunction with the R-32 variable capacity heat pump to examine performance characteristics of the integrated system. This section examines the impact of zoning on HVAC performance and the operation of the zoning controls integrated to the variable capacity system. The two-zone ducted system was split evenly, such that ~50 percent of the conditioned air would be provided in each zone for a full-load cooling situation across the two zones.

After multiple experimental tests with the zoning equipment and variable capacity system, a general control strategy and means of operating of the integrated system was established. Each zone contained a communicating thermostat, which could provide a percent demand to the zoning control board. The zoning control board received the percent demand values from the two-zones and relayed an average percent demand of the two-zones to the variable capacity system. For instance, if Zone 1 called for a 60 percent demand and Zone 2 an 80 percent demand, then the percent relayed to the variable capacity system would be 70 percent. The dampers utilized operated in either the open or closed position, and thus the conditioned air would be split evenly between the two zones. The percent demand from a single zone carried a 50 percent weight in the overall load and output of the variable capacity system. For instance, if Zone 1 called for 100 percent demand and Zone 2 was 0 percent, then the variable capacity system would operate and be capped at 50 percent nominal output. The following paragraphs and tables describe the operation of the zoning controls and variable capacity system in greater detail.

Table 27 examines the turn-down ratio of the variable capacity heat pump for multi-zone and single-zone situations. Turn-down ratio refers to the ratio of the minimum capacity to the maximum capacity for an operating state and condition. The conditions used for this example were cooling operation at 75°F (24°C), 50 percent RH indoor air and 95°F (35°C) outdoor air. The table provides the maximum cooling capacity per zone, minimum cooling capacity per zone, and turn-down ratio for the described zoning applications. The zoning application utilized in the laboratory consisted of two zones with each zone receiving an equal share of conditioned air from the variable capacity system. For a single-zone application, the variable capacity system can operate within the full boundaries of equipment. For a two-zone application with both zones calling for space cooling, the maximum and minimum cooling capacity is split evenly between the two zones. When only one zone is calling for space cooling, the minimum capacity is the minimum output of the variable capacity system, while the maximum capacity is capped at ~50 percent of the equipment output. The turn-down ratio of the variable capacity system is doubled with only one zone in operation for the two-zone setup. In a variable capacity – zoning application with only one zone in operation, the range of variable capacity operation is reduced, however in this scenario the variable capacity system would be operating at minimum and an intermediate level of operation.

Table 27: Comparison of Turn-Down Ratio for Zoning Scenarios

	1-Zone Setup	2-Zone Setup (50-50)	
		Both Zones in Operation	1 Zone in Operation
Max Cooling Capacity per Zone	22,500	11,250	11,250
Min Cooling Capacity per Zone	6,500	3,250	6,500
Turn-Down Ratio	28.9%	28.9%	57.8%

Source: EPRI

In practice, each zoning application may result in unique performance characteristics, based on number of zones, ductwork layout, and equipment type. The following comparison demonstrates an example of equipment performance differences for a variable capacity – zoning application.

Table 28 provides experimental data from the variable capacity – zoning laboratory setup under 1-zone and 2-zone operation for a similar total load on the variable capacity system. The load imposed on the variable capacity system was approximately 10,000 Btu/h in each case. In the first case, the 10,000 Btu/h load was imposed on single zone, while in the second case the load was imposed across the two zones (~5,000 Btu/h in each zone). The table provides a comparison of equipment performance under single-zone and two-zone operation. The primary observation in equipment performance between the two cases revolved around the indoor blower and indoor airflow. The two cases provided different ductwork layouts for airflow to travel, which resulted in different external static pressures and different indoor power consumption.

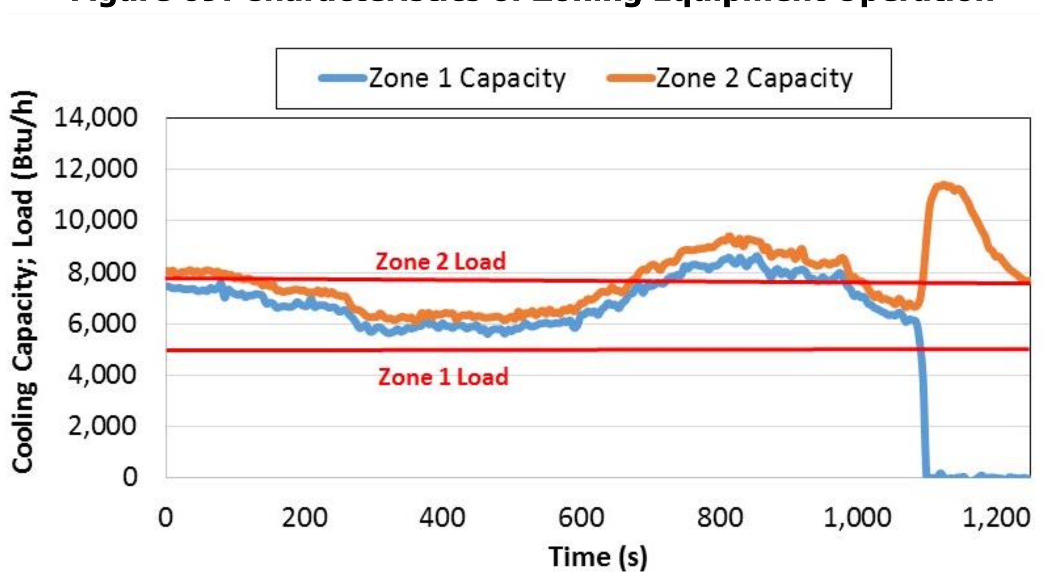
Table 28: Comparison of Equipment Performance Under Zoning Operation

	1 Zone in Operation	Both Zones in Operation
Zone 1 Airflow (CFM)	550	280
Zone 2 Airflow (CFM)	—	304
Unit External Static (in W.C.)	0.27	0.13
Unit Capacity (Btu/h)	9,800	10,600
Indoor Unit Power (W)	134	90
Outdoor Unit Power (W)	564	596
Total Unit Power (W)	698	685
Equipment Energy Efficiency Ratio (Btu/h/W)	14.0	15.5

Source: EPRI

As discussed, the variable capacity, zoning equipment examined in the laboratory provided an average capacity based on the demand of the two zones. The capacity output of the variable capacity heat pump was split evenly into each zone calling for space conditioning. Figure 69 illustrates the operation of the variable capacity, zoning equipment for an unbalanced load across the two zones. The test conditions were cooling operation with 75°F (24^{o}C) indoor temperature and 95°F (35^{o}C) outdoor temperature. In the experimental test, a thermal load of ~8,000 Btu/h was imposed on zone 1, while a load of ~5,000 Btu/h was imposed on zone 2. The variable capacity system oscillated around the combined load of the two zones for a period of time until the temperature setpoint in zone 2 was satisfied. After the zone 2 setpoint was satisfied, the variable capacity system adjusted to account for the cooling load in zone 1.

Figure 69: Characteristics of Zoning Equipment Operation



Source: EPRI

Zoning a central, variable capacity space conditioning system may alter the functionality of the variable capacity unit compared to a non-zoned application. A variable capacity system's turn-down ratio and ability to match load at part-load conditions could be affected by the number of zones calling for space conditioning. The equipment performance of a zoned, variable capacity system may also differ depending upon the number of zones calling for space conditioning. For unbalanced loads in a zoned application, a variable capacity system may provide an output based on the average percent demand of the zones to establish a level of comfort for each zone. A field evaluation could provide further understanding of zoning and variable capacity integration and performance (Chapter 4).

PG&E Lab Activities and Results

Phase 1 testing by PG&E involved laboratory testing of a prototype system using R-410A as the refrigerant under both cooling and heating modes, but with emphasis on heating and the dual-fuel capabilities. The results of these tests are reported in the EPRI Phase 1 Laboratory Evaluation Report (Beaini et al., 2017) and in Chapter 2.

Phase 2, the subject of this section, continued the system testing to evaluate the built-in fault detection and diagnostic (FDD) capabilities of the prototype system, and to test the system's performance using an alternative refrigerant: R-32. This second part of Phase 2 repeats many of the tasks from Phase 1 to assess performance differences between R-32 and R-410A.

The method, instrumentation, and facilities deployed in Phase 2 are similar to that reported in Phase 1.

Fault Detection and Diagnostics Testing

Fault detection and diagnostics (FDD) as applied to an air conditioner is analogous to the check engine light in a car. It is intended to alert the user to issues that are affecting the operation of the unit and to assist in performing maintenance or fixing a problem. With complete understanding of how a system is supposed to operate in a particular situation, FDD systems can report when issues are occurring that could cause operating problems, and not just after the problem has occurred.

The test unit is equipped with several sensors and a sophisticated computerized control system which allows for identification when fault conditions exist within the system, and report this information to the end user or service technician. As described in the manufacturer's Heat Pump Installation and Service Reference (Daikin, 2016), the heat pump can produce 51 fault codes; and the manufacturer's Furnace Installation Instructions (Daikin, 2015) list an additional 25. The fault codes are identified with LED displays on the control boards on the individual systems, and are also transmitted to the central thermostat. Only the most severe faults that cause the system to shut down are displayed on the front panel of the thermostat, while other faults that may just interrupt operation are recorded on "Diagnostics" pages for each component. The diagnostics pages are accessed through the thermostat using the following steps, which involves entering an installer password (4-digit code):

- Menu Tab ⇨ ComfortNet User Menu ⇨ (Installer Password) ⇨ Furnace or Heat Pump ⇨ Diagnostics

Fault conditions are also recorded on an "Alerts Log" in the thermostat, which provides a more descriptive record including the time and date of the fault, and is accessed through the following sequence:

- Menu Tab ⇨ Installer Options ⇨ (Installer Password) ⇨ Data Logs ⇨ Alerts Log

Both the Diagnostic pages and the Alerts Log can be cleared of recorded faults once identified and corrected. Severe faults that prevent the system from operating can only be cleared by first fixing the problem and then cycling power to the system.

Plan for Testing FDD Capabilities

System faults that can affect performance fall into two categories: installation errors and problems resulting from cumulative wear and tear or damage to the system. Faults in the first category can include:

- High or low refrigerant charge
- Non-condensable gas (air) in the refrigerant
- Refrigerant flow restrictions from poor brazing, kinked lines or unopened service valves
- Incorrect fan speed setting

Faults in the second category include:

- Low refrigerant charge (from leaks)
- Refrigerant line restrictions (due to bent or crushed tubing, clogged filter/dryer)
- Expansion device faults (clogging, stuck TXV/EXV, sensing bulb malfunction)
- Low indoor unit airflow (fouled coil or filter, closed supply dampers, restrictive ducting)
- Low outdoor unit airflow (coil fouling)
- Compressor or blower/fan electrical faults or physical blockage
- Thermostat sensor failure

Ideally, those in the first category should be avoidable by following proper installation procedures; while those in the second category are subject to how the system is operated and the environment that it is in.

While it may be desirable to test for each of the listed fault codes, this is not practical since some of these faults are fatal and would require replacement of broken components. Also, the priority is on the faults that can occur during regular system operation over those caused by installation errors, with the assumption that most knowledgeable technicians will get the installation right. Some error codes are produced as the result of a failure in a startup procedure, which also could not be easily replicated. There are also limits in terms of the effort required to cause a fault, as some faults may be triggered in some operating climates but not others and conducting a full spread of operating conditions for each stage of fault application would be time and cost prohibitive. It was also unclear initially whether a fault has to exist for a minimum period before being identified as a problem, which may be a safety feature to prevent false error reporting. Discussions with the manufacturer indicated that faults should be recorded immediately once a condition threshold has been crossed, which allowed for a shortening of the testing period.

For all of the running tests, the test unit was operated under the AHRI Standard rating conditions for cooling and heating when set to run in those modes and operated via its thermostat with at least a 5°F (2.8°C) differential between the thermostat set point and the controlled room temperature to try to push the unit to its maximum capacity. Thermostat control was applied because it was not known whether operating the system in a fixed speed mode via the Ram Monitor application would produce the appropriate response. All of these tests were run with the system having its original charge of R-410A refrigerant from the Phase 1 tests.

As described in the following section, the faults attempted are listed in increasing difficulty to perform, although not necessarily the order in which they were actually tested.

Specific FDD Tests

The following paragraphs describe specific tests designed to emulate: sensor failure, blower and fan blockage, a closed gas valve, high or low voltage, liquid line restriction, indoor airflow restriction, outdoor airflow restriction, low refrigerant charge and high discharge temperature faults.

Sensor Failure

The heat pump is equipped with several temperature sensors and a pressure sensor that allow the system to determine its correct mode of operation as well as to diagnose problems. Failure of the sensors can be the result of direct sensor issues like physical damage, or the communicating wires can be cut, shorted, or chewed through by rodents. To simulate this fault, the sensors were simply disconnected from their sockets on the outdoor unit control board.

Disconnecting the outdoor air sensor produced error code E25: AIR SENSOR FLT on the heat pump diagnostics screen, as shown in Figure 70, but no indication on the main screen, probably because the outdoor temperature sensor is not critical to system operation.

Figure 70: Thermostat Heat Pump Diagnostics Page with Air Sensor Fault

Source: EPRI

Disconnecting the plug with four temperature sensors (coil circuit, defrost, liquid line, discharge line) produced errors concerning only the discharge temperature sensor (error code E23: DISCH TEMP FAIL), and a severe fault indication on the main screen, as shown in Figure 71. There are other error codes listed in the manual associated with the other temperature sensors in the block (E27: COIL TEMP FAIL 1 for the defrost sensor, E28: COIL TEMP FAIL 2 for the coil circuit sensor, E29: LIQUID TEMP FAIL for the liquid sensor), but to trigger these would require disconnecting individual wires from the connector plug, which was not attempted.

Figure 71: Thermostat Displays with Discharge Temperature Sensor Fault

Source: EPRI

Unplugging the pressure sensor from the board produced error code E26: PRESSURE SENSOR, and another severe fault message. (The Eb0: NO ID AIRFLOW fault shown on the Diagnostics page in Figure 72 was recorded from an earlier test to be described later, and had just not been cleared.)

Figure 72: Thermostat Displays with Pressure Sensor Fault

Source: EPRI

The test unit did not need to be operating to trigger these faults.

Blower and Fan Blockage

For this test, the indoor blower and outdoor unit fan were individually restrained from turning and then the unit was activated. Fan problems can be caused by debris fouling or other physical damage that prevents the fan from turning freely.

With the outdoor unit fan blocked, the diagnostics page showed three faults of code E53: PCB OR FAN MIN as it attempted three times to start up, followed by fault code E19: PCB OR FAN FAIL and a severe fault message on the main screen, as shown in Figure 73.

Figure 73: Thermostat Displays with Faults from Outdoor Fan Blockage

Source: EPRI

Blocking the furnace blower from turning produced error messages from both the heat pump and the furnace. The furnace diagnostics page reported B0: MOTOR NOT RUN and B4: MOTOR TRIPS, and the heat pump diagnostics page reported error Eb0: NO ID AIRFLOW, as shown in Figure 74.

Figure 74: Thermostat Displays with Faults from Indoor Blower Blockage

Source: EPRI

Closed Gas Valve

If the gas supply valve is closed and the system is instructed to use gas for heating (backup source), the furnace will attempt three starts; and if no ignition is achieved, will display severe fault code E0: LOCKOUT. The furnace may be reset by cycling power to it alone (and not the heat pump). (If the furnace is not operating, the unit can provide heat from the heat pump so long as the backup source is not called for.)

High or Low Voltage

230V to the maximum for the voltage regulator at 277V (+20 percent) did not produce a fault, and the system continued to operate normally. Lowering the voltage, the unit shut off and a fault was displayed on the thermostat when the voltage dropped below 140V (–40 percent). The correct fault code of E42: LOW LINE VOLT was recorded on the heat pump diagnostics page, as shown on Figure 75.

Figure 75: Thermostat Displays with Low Voltage Fault at the Heat Pump

Source: EPRI

Restoring the voltage to a normal level and cycling power to both the furnace and outdoor unit cleared the error on the thermostat. Cooling performance metrics were not significantly affected by the changes in voltage.

Liquid Line Restriction

Closing the liquid service valve to restrict refrigerant flow did not produce a fault until the valve was completely closed while the unit was operating in cooling mode. With the valve slightly cracked opened, the liquid would flash passing through the valve causing the liquid line to frost over and leaving very little pressure drop for the TXV at the evaporator, as shown in Figure 76.

Figure 76: Frosting of Liquid Line with Service Valve Almost Closed

Source: EPRI

This also caused the unit to cycle on and off but did not produce any fault indication at the thermostat after two cycles. Shutting the service valve off completely resulted in fault code E16: LOW PRESSURE MINOR, as shown in Figure 77, which still allowed the system to operate. It is possible that after more cycles the error condition could have advanced to E15: LOW PRESSURE CRITICAL, but lack of time prevented the continued operation needed to test this assumption.

Figure 77: Thermostat Display with Minor Low Pressure Fault

Source: EPRI

Surprisingly, shutting the liquid valve while in the heating mode did not produce any fault code even after 20 minutes of operation. Other fault codes besides E15 and E16 that could be triggered by the service valve being closed (according to the "Probable Causes" in the Heat Pump Installation & Service Reference (Daikin, 2016) include:

- E13: HIGH PRESSURE CRITICAL

- E14: HIGH PRESSURE MINOR
- E17: COMPRESSOR FAIL
- E35: HIGH CURRENT
- E52: COMP FAIL MINOR

Indoor Airflow Restriction

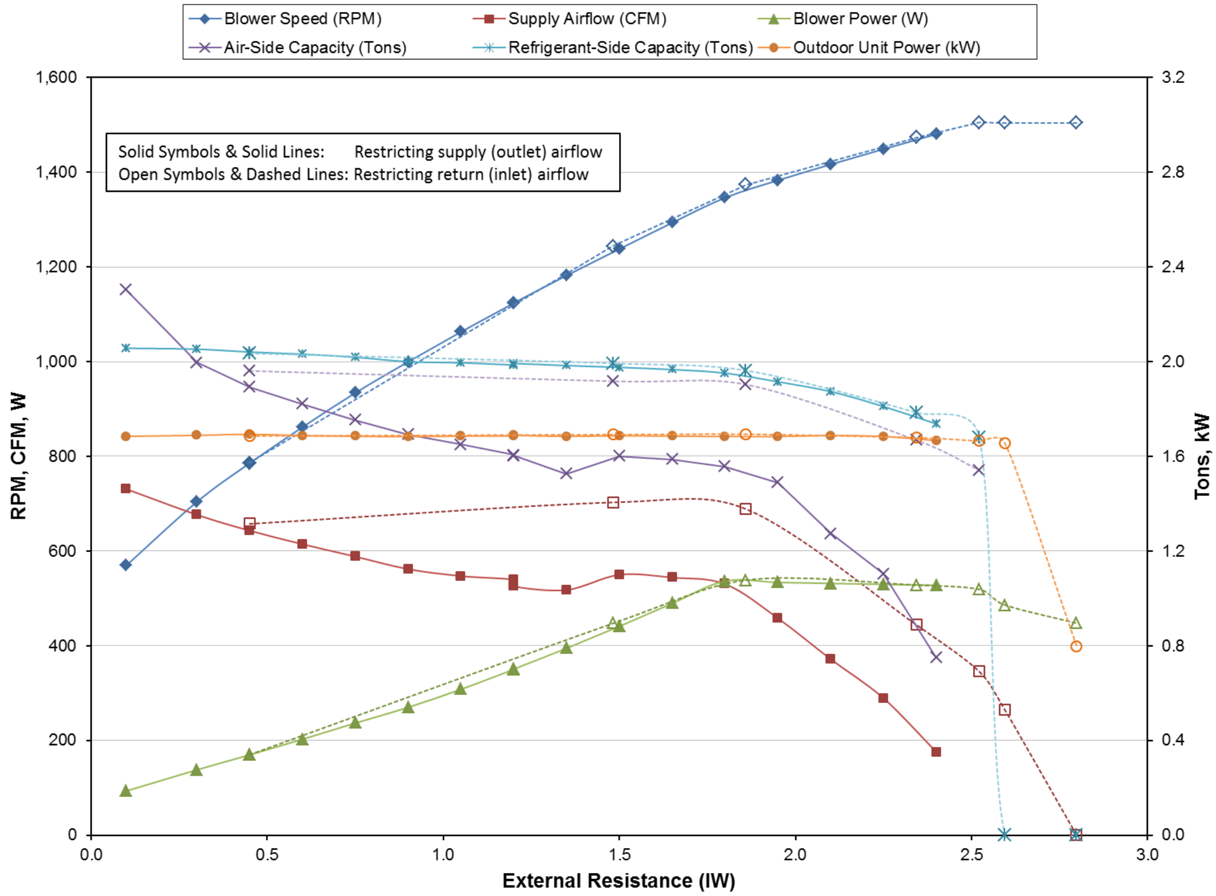
Several factors can contribute to a low indoor airflow situation. Flow may be restricted on the downstream side of the blower as the result of closed supply dampers, crushed ducts, or a fouled coil. On the upstream side of the blower, the flow may be restricted by a dirty filter (the most likely cause) or blocked return registers or restricted return ducting. Each of these situations could result in a different response from the unit, so both situations were examined.

With the variable speed blower in this system, increases in airflow resistance are usually compensated for by the blower speeding up to maintain its airflow or motor torque. As such, there is a wide range of airflow resistance that the system is capable of operating at without identifying that there is a problem.

Several tests were conducted with different applications of flow restriction. The first set was done with the unit operating in cooling mode with the airflow being restricted on the downstream side of the coil. This was accomplished via a combination of controlling the speed of the blower on the airflow measurement apparatus and by closing off its nozzles. One problem with this test method is that as the flow is decreased, the accuracy of the flow measurement is also decreased. Also, the increased backpressure exposed a few leaks in the test ducting and the test unit itself that allowed air to escape upstream of the measurement apparatus. *It is possible that the pressure was increased beyond the point where it blew out the water trap in the condensate line allowing air to escape through that path.* A separate set of tests were then conducted with the airflow restricted on the upstream side by blocking the intake, also in cooling mode. Any leaks in the furnace case would likely be occurring on the upstream side of the blower and would not affect the downstream airflow measurement.

Figure 78 presents the trend of some performance characteristics as the airflow resistance was increased, as indicated by the measure of differential pressure across the unit. In the figure, the results from the first round of testing with the supply restricted are shown with solid symbols and solid lines, while the results from the second round with the restriction on the return side are shown with open symbols and dashed lines. For comparison purposes, the figure includes both the primary capacity measurement on the air side, and the secondary capacity measurement taken on the refrigerant side of the coil (refrigerant mass flow rate multiplied by its enthalpy difference across the coil, less the blower input power). For the case with the restriction on the return side, the capacity metrics are in reasonable agreement. However, the results from the test with the supply side restricted show a significantly lower capacity from the air side measurements as a direct result of the incorrectly low airflow measurement, which is also included in the figure. The response of the measurements of blower speed and power and outdoor unit power are about the same for the two test sequences, and display the general increase in fan speed and power as the flow is restricted. The blower power eventually settled out at a maximum and actually began to decrease once the blower reached its maximum speed but was no longer producing much airflow. The outdoor unit power was mostly unaffected by the indoor airflow restriction up until the point that airflow was essentially completely cut off and the outdoor unit had no load from the evaporator coil.

Figure 78: Restricting Indoor Airflow in Cooling Mode



Source: EPRI

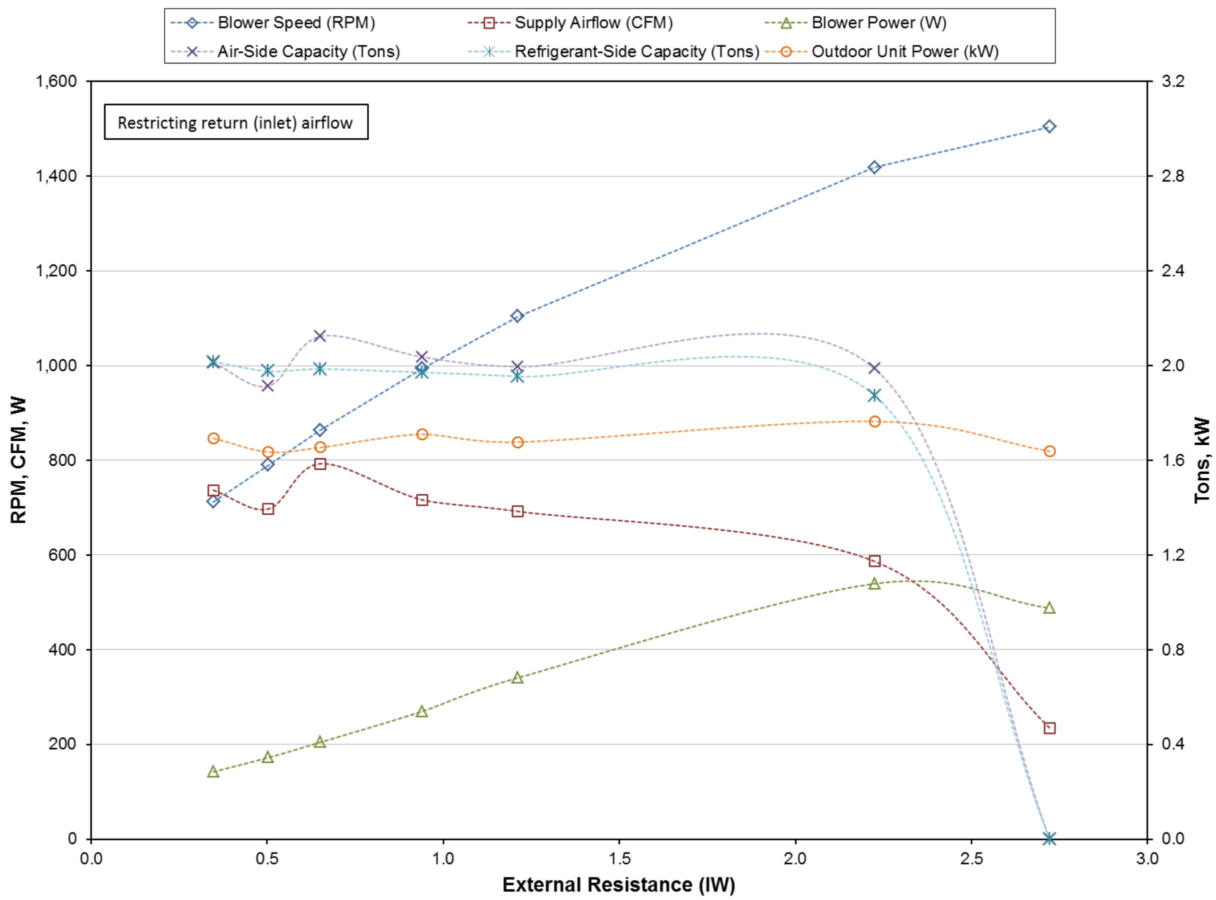
In regards to the response from the fault detection system, most were reported by the furnace as it was the system directly affected by the fault. The first indications appeared around the 1.5 IW resistance point with the furnace reporting a B3: MOTOR LIMITS fault on its Diagnostics page. This warning fault is triggered when the "circulating blower motor is operating in a power, temperature, or speed limiting condition" and does not actually interrupt operation. Several of these were recorded over the course of both tests at the higher resistance points. Eventually, the fault messages progressed to several B9: LOW ID AIRFLOW on the furnace diagnostics page, which also did not interrupt operation. Once the airflow was completely cut off at the intake, the heat pump diagnostics page recorded fault code Eb9: LOW ID AIRFLOW, as shown in Figure 79, as the heat pump controller correctly identified that it could not absorb any heat from the non-moving indoor air.

Figure 79: Thermostat Diagnostic Pages with Restricted Indoor Airflow Faults

Source: EPRI

The test sequence with the intake air restricted was repeated with the heat pump in heating mode with a similar response in system performance metrics, as shown in Figure 80.

Figure 80: Restricting Indoor Airflow in Heating Mode



Source: EPRI

Additional B3 and B9 faults were recorded on the furnace Diagnostics page during these tests, as shown in Figure 81, but even with the intake again completely closed off, nothing was recorded on the heat pump Diagnostics page as a fault. In neither cooling nor heating modes was a fault condition serious enough to be displayed on the main screen of the thermostat or shut the system off, meaning the problem would have needed to be noticed in order for the user to purposely access the diagnostics or alerts pages to investigate the problem.

Figure 81: Indoor Airflow Restriction and Thermostat Diagnostic Page (Heating)

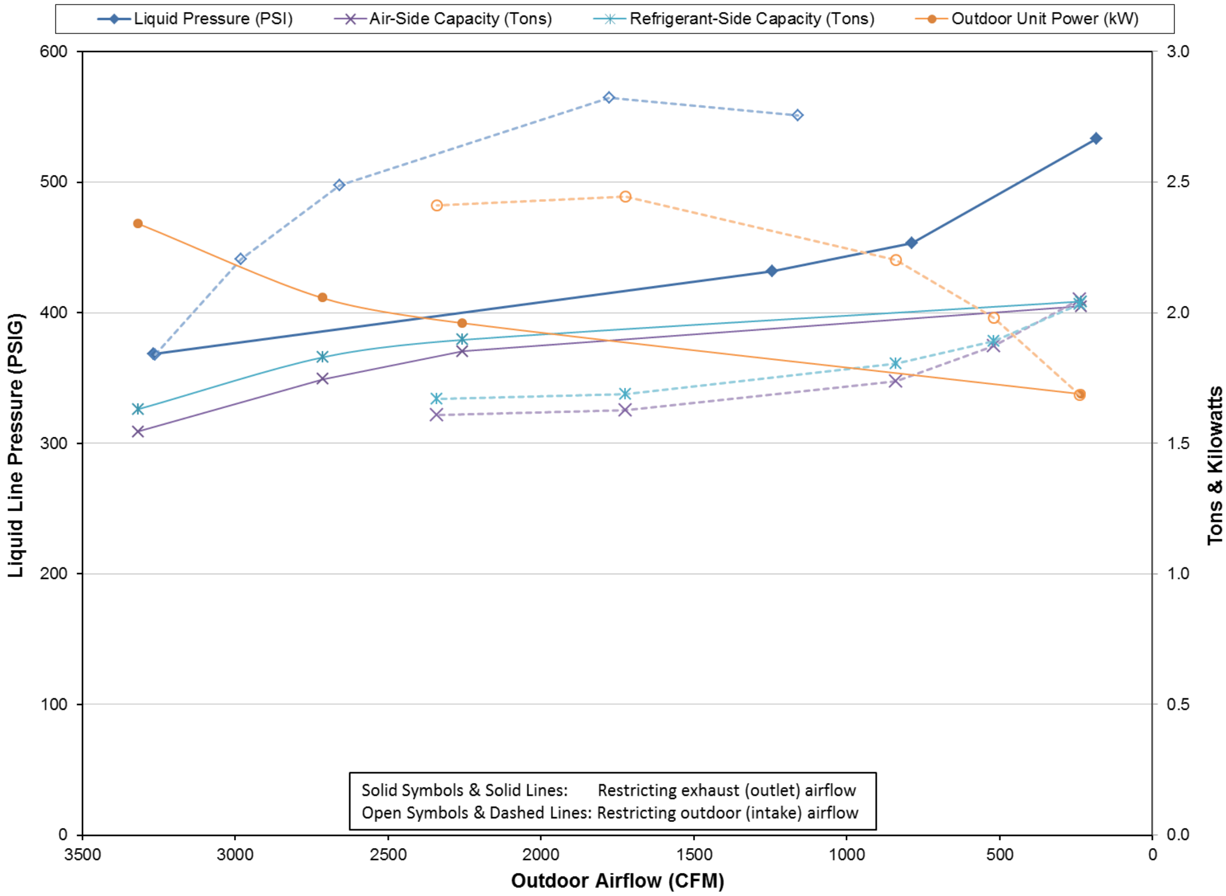
Source: EPRI

Outdoor Airflow Restriction

Similarly, to the indoor side, the outdoor unit airflow can be restricted on the discharge side due to fan discharge blockage, or on the intake side due to coil fouling. As before, both scenarios were examined with the unit operating at the AHRI Standard cooling and heating rating conditions. Outlet resistance was restricted on the discharge side by the same methodology of closing nozzles on the airflow measurement apparatus and controlling its booster fan speed, similar to the indoor side. On the intake side, resistance was added in the form of paper towels wrapped around the condenser coil (from the bottom up), while the booster fan was set to maintain the usual zero exhaust static pressure. Although the fan on the outdoor unit is also variable speed for the purposes of capacity control, its speed remained at a constant 650 revolutions per minute during all of these tests. Propeller fans also do not have a large capability to handle flow resistance, so small changes resulted in relatively large changes in performance.

Figure 82 shows the trends of various performance metrics as the airflow was restricted. Rather than trend the results as a function of the differential pressure across the test unit, this chart uses the measured exhaust side airflow, which decreases as the resistance is increased. As before, as the flow is restricted, the airflow measurement becomes less and less accurate, particularly below about 260 CFM, which is the lowest airflow that the measurement apparatus can measure within its prescribed lower accuracy limit. Restricting the airflow on the outlet side produced strikingly different results than restricting the airflow on the intake side. This is possibly because restricting the outlet side lowers airflow across the entire coil, while incrementally wrapping the intake side of the condenser coil initially accelerates the airflow across the remaining exposed coils.

Figure 82: Restricting Outdoor Airflow in Cooling Mode



Source: EPRI

As far as the fault detection events, none were recorded in the scenario when the restriction was applied to the outlet (exhaust) side of the outdoor unit. Somehow, the condenser coil still achieved some circulation, drawing air in across the lower coils and creating a backwards flow across the upper coil sections due to the extra centrifugal air pressure from the fan tips. *In the scenario where the condenser coil intake was restricted, once it was completely wrapped in paper towels, the system went into a cyclic operating mode where it repeatedly tripped off due to high discharge pressure*. Several high-pressure fault codes E14: HI PRESSURE M were recorded on the heat pump Diagnostics page, as shown in Figure 83, but these were classified as minor and nothing was displayed on the main thermostat screen, and the system simply cycled back on after a short resting period.

Figure 83: Applied Outdoor Airflow Restriction and Thermostat Diagnostic Page

Source: EPRI

In heating mode, with the intake airflow again restricted by paper towel wrapping, no errors were reported even on the heat pump Diagnostics page, and the system continued to operate, although at reduced capacity. *One observed effect of operating in heating mode with the condenser intake restricted was that the system appeared to interpret the flow restriction as high frost buildup, and increased its frequency of defrost cycles.*

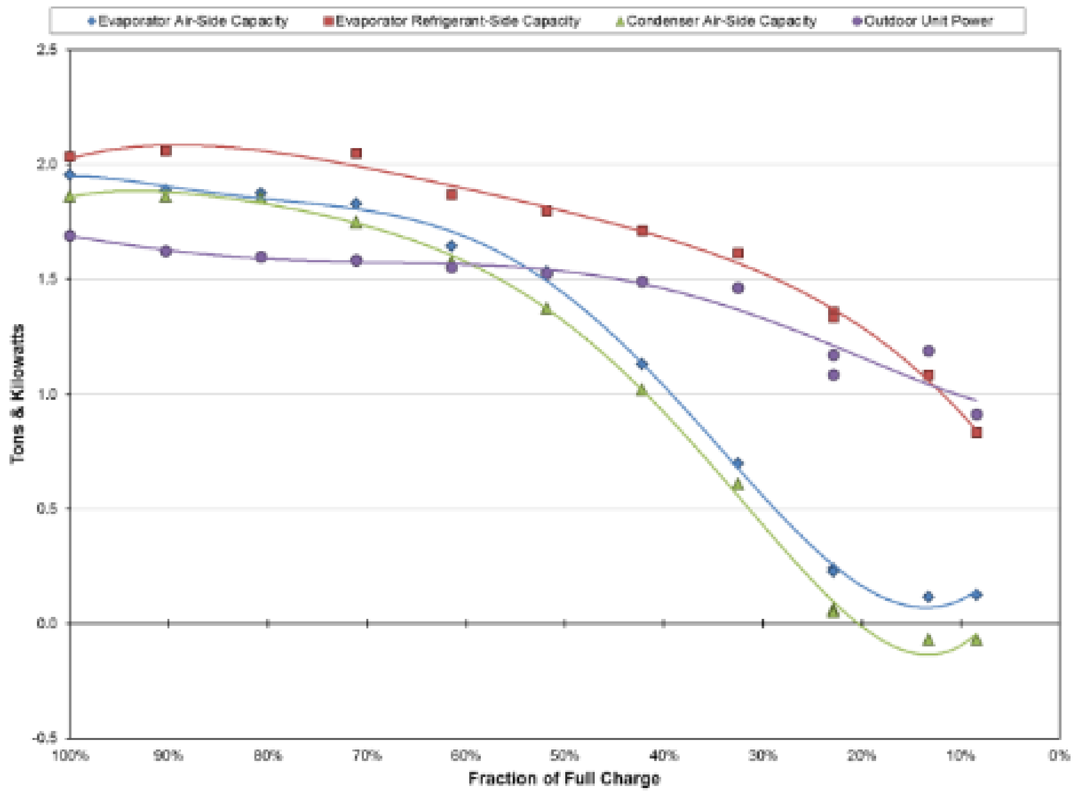
Low Refrigerant Charge

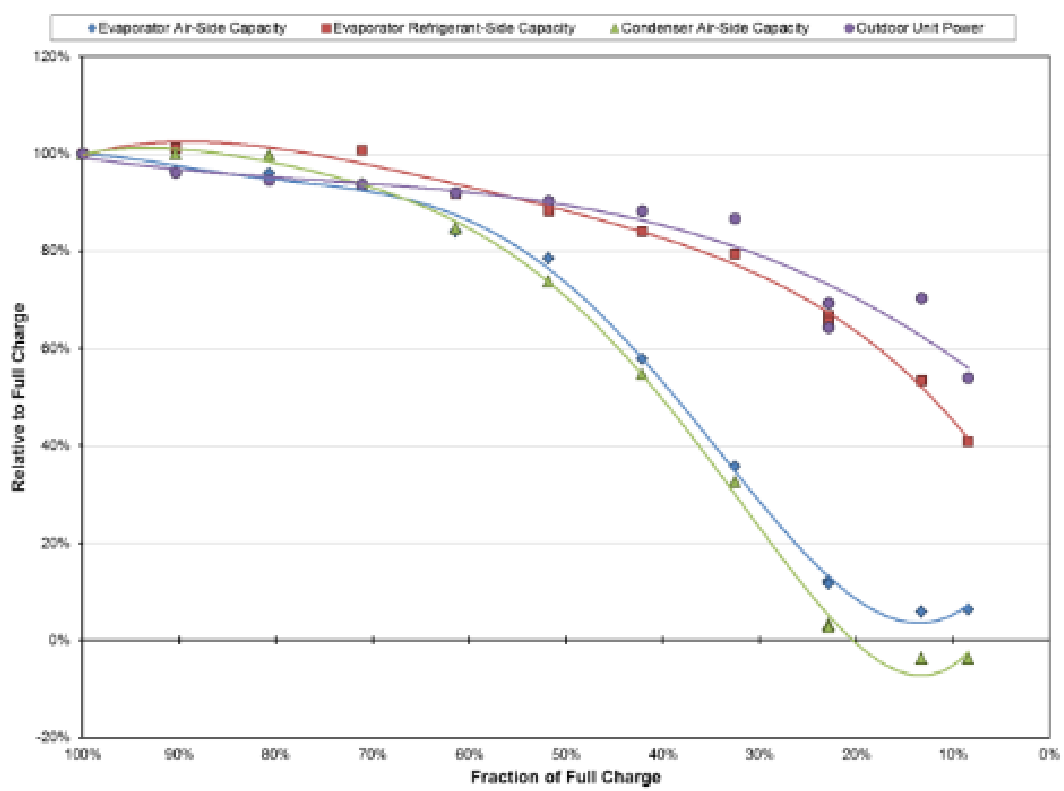
The last test of the system's fault detection capabilities was an attempt to trigger a low refrigerant charge fault. Deviating from the test plan, a test intended to trigger an overcharge fault was not conducted after discussions with the manufacturer suggested it would take a significant level of overcharge to create a fault condition due to the refrigerant capacity of the liquid accumulator. The low charge test was done last as it worked into the next step in the project involving the use of an alternative refrigerant. The methodology employed for this test was to remove refrigerant in steps into a recovery tank mounted on a scale in order to measure the weight of refrigerant removed. At each step, a test was run in both cooling mode and heating mode for at least one hour at the standard AHRI Standard rating conditions (A_2 and $H1_2$). System performance was recorded over a period that was relatively steady-state to gauge the relative change in performance as a function of charge. Refrigerant was removed in steps of 1 pound from the written full charge of 10 pounds, 6 ounces.

The results from these tests are summarized in Figure 84 for cooling and Figure 85 for heating. Each figure includes two charts, the first with absolute performance metrics and the second showing the relative performance to that at full charge. Three measures of system capacity are shown, including the primary measurement of evaporator air side capacity (evaporator airflow rate and change in air enthalpy across the indoor unit), and the secondary measurements of evaporator refrigerant side capacity (refrigerant flow and change in refrigerant enthalpy across the evaporator, less the indoor blower power) and condenser air side capacity (condenser airflow and change in air enthalpy across the outdoor unit, less total power input).

The results show a fairly good agreement between the evaporator and condenser air-side capacity metrics, but less so for the refrigerant side capacity. One factor that will affect the refrigerant capacity measurement is the assumption that the refrigerant passing through the coil is changing state from a subcooled or saturated liquid to a superheated or saturated vapor. At very low charge conditions, this may not be the case, particularly on the liquid side which may be a liquid/vapor mixture. At extremely low charge conditions, there is likely no liquid refrigerant left anywhere in the system and the refrigerant is all in a vapor state.

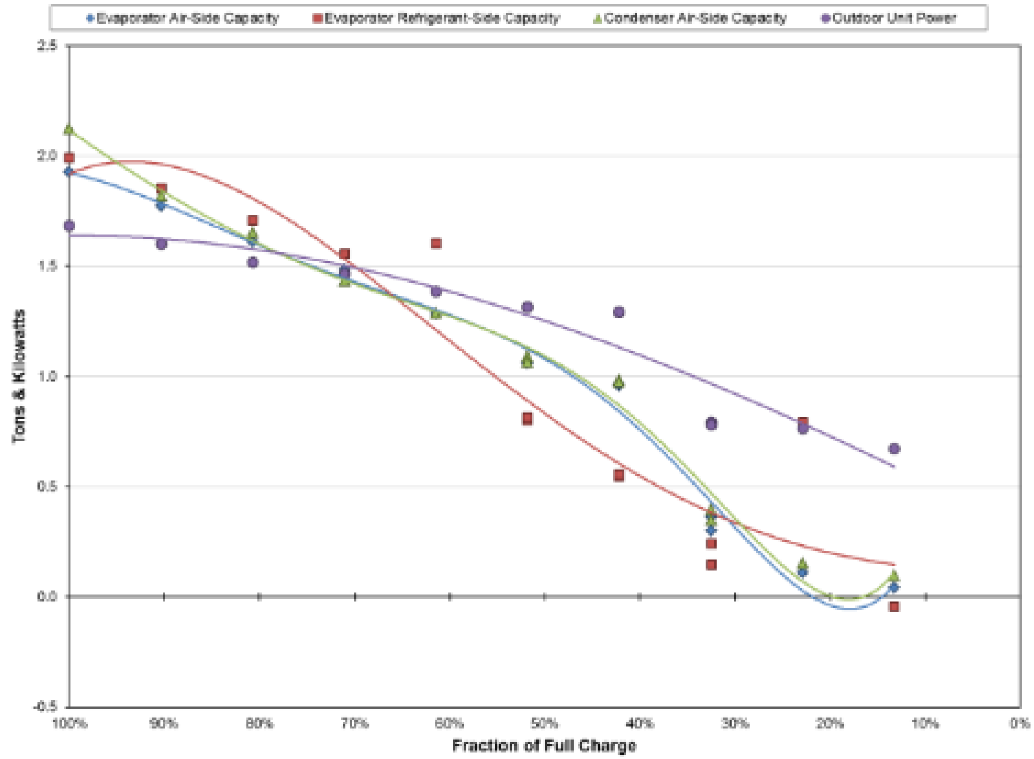
Figure 84: Cooling Performance as a Function of Refrigerant Charge

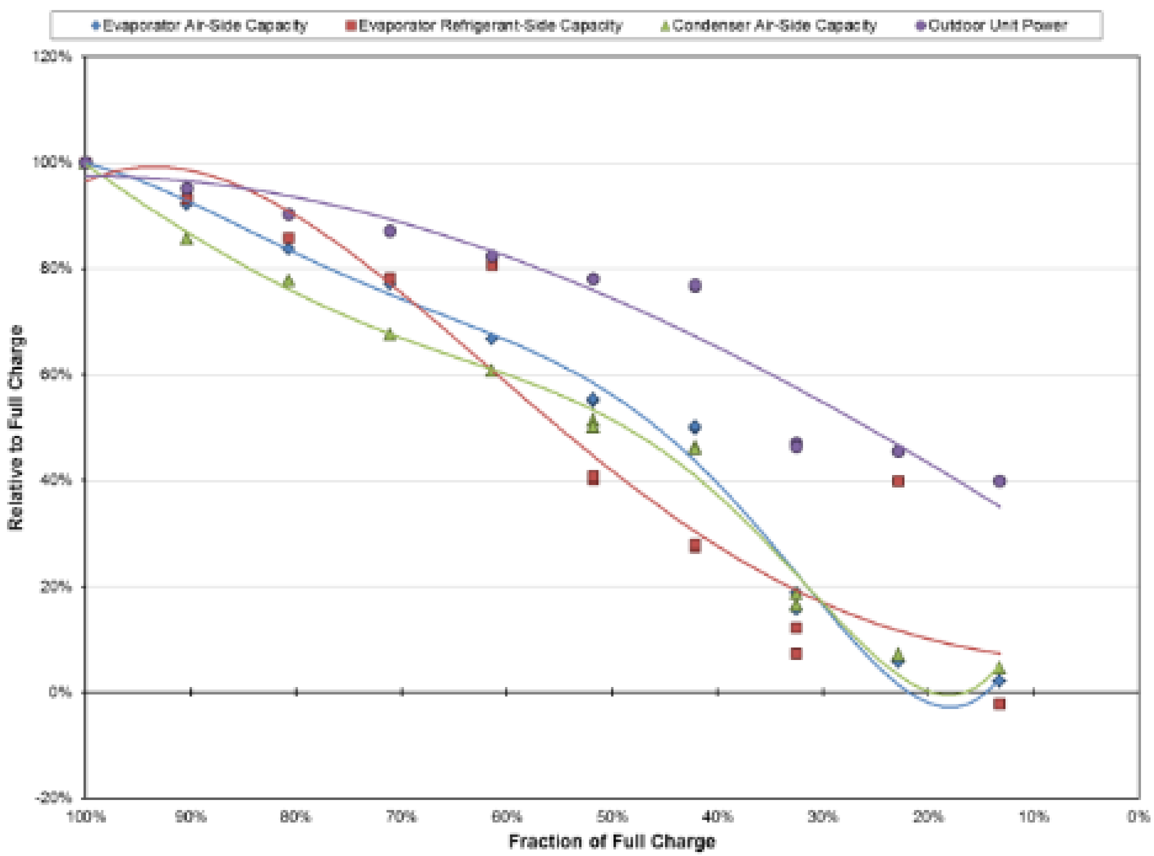




Source: EPRI

Figure 85: Heating Performance as a Function of Refrigerant Charge





Source: EPRI

The results from these tests show a significant drop-off in performance as the refrigerant is removed, with a higher sensitivity in heating mode versus cooling mode. One of the initial indications of performance issues when in heating mode was an increased frequency of defrost cycles, which made it difficult to capture long periods of steady state operation. The lack of smooth curves in the results is also affected by the system control through the thermostat, as the system appeared to compensate for the lack of capacity by reducing the indoor airflow.

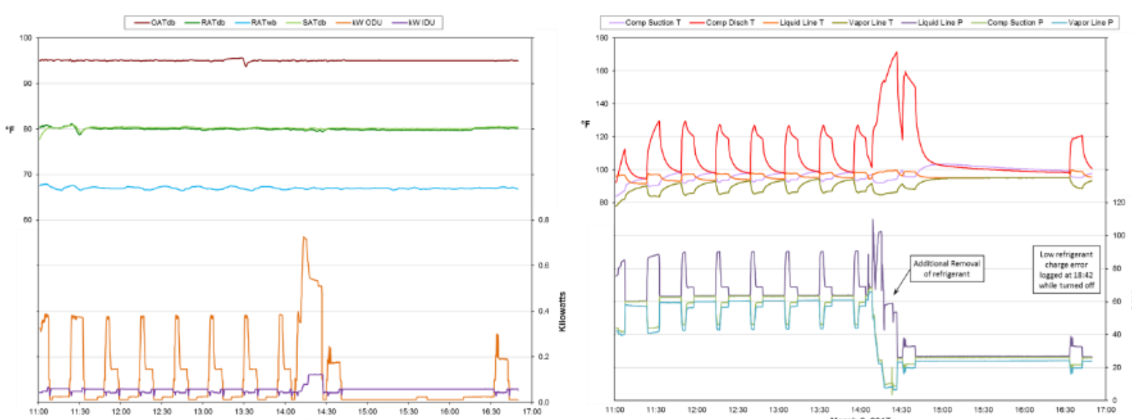
Despite removing nearly 90 percent of the refrigerant charge and the loss of all capacity, the thermostat never displayed a low charge fault up to this point. The only fault indication that was recorded on the diagnostics page was at the 10 percent charge level in heating mode when the unit experienced multiple trips due to high discharge temperature. Three fault code E55: HI DIS TEMP MIN were recorded when the measured compressor discharge temperature exceeded 250°F (121°C), as shown in Figure 86.

Figure 86: Thermostat Display with High Discharge Temperature Faults

Source: EPRI

Further removal of refrigerant was attempted, although this was becoming more difficult as it was not easy to create a pressure difference between the system and the recovery tank except by making the recovery tank and its contents colder than the environment around the test unit by placing it in an ice bath. Because of this, the weights of the last removals were not accurately recorded. e 87Figure 87 shows measurements and performance metrics with the unit in cooling mode from this final attempt.

Figure 87: Cooling Mode Test at Very Low Charge



Source: EPRI

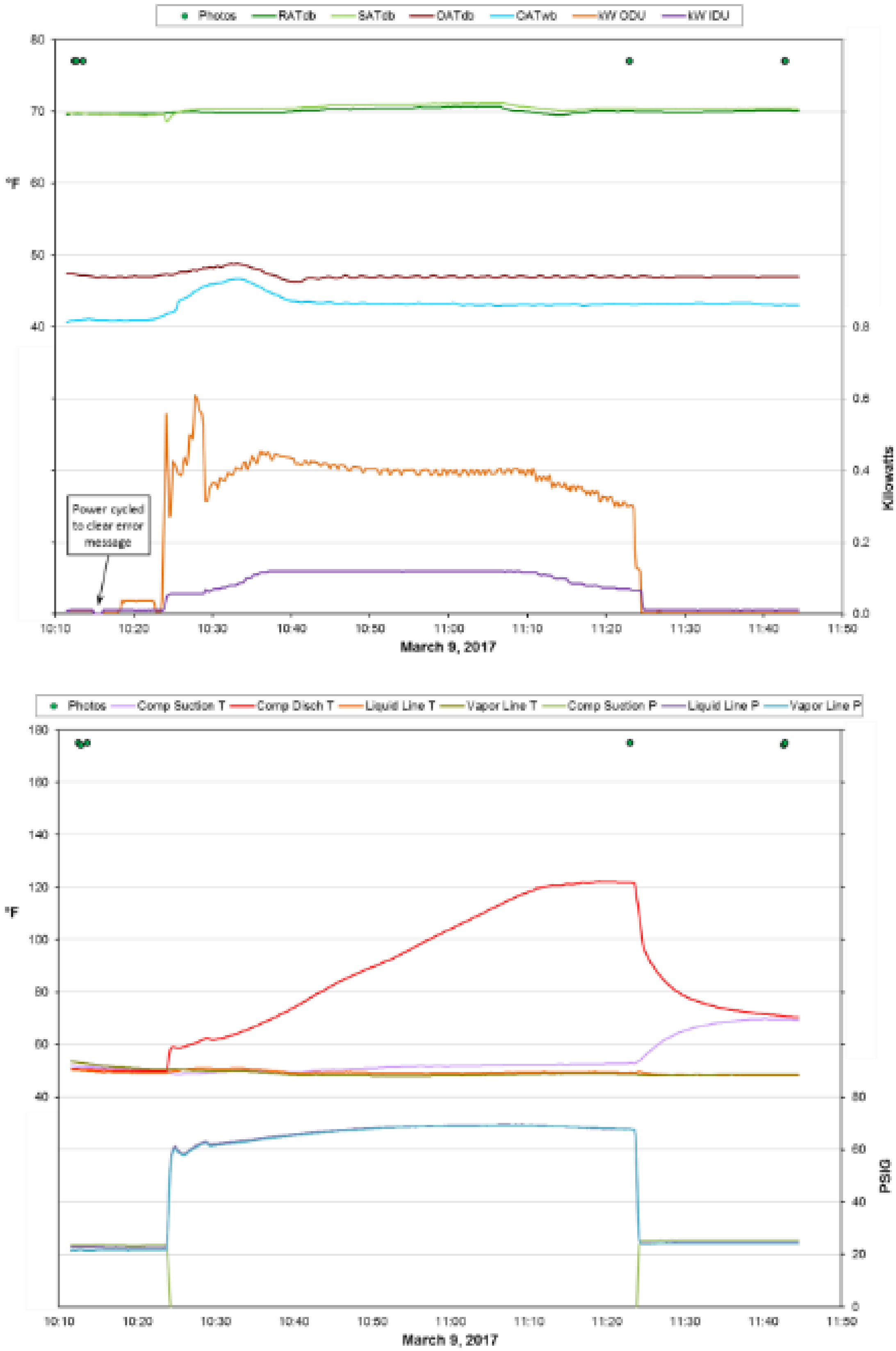
The air temperature trends in the left figure show the conditions being held at the AHRI Standard rating conditions for the entire 6-hour test window. Control of the unit was via its thermostat with a cooling set point of 72°F (22^{o}C) although the room temperature was being held at 80°F (27^{o}C). The resulting operation of the test unit was cyclic, although no fault messages were recorded by the thermostat. Note that the supply air temperature was actually slightly above the return air temperature due to blower heating and no cooling from the coil. After 3 hours of this cyclic behavior, another refrigerant removal was attempted resulting in a significant drop in pressure. Considering the pressure indication in the right figure when the compressor was off at around 26 PSI on both the liquid and vapor lines (which for R-410A would mean a saturation temperature of around –21°F (–29^{o}C)) there is likely no liquid refrigerant left in the system and it is all vapor. After this removal, the controls tried to activate the compressor twice for short periods, and kept the indoor blower running. At 16:50, the test was concluded by switching the thermostat from “Cool” to “Off” and shutting down the test room conditioning systems, with still no fault indication. However, later into the evening, the thermostat did finally log an E41: LOW REFRIGERANT fault at 18:42 while the system was sitting idle, as shown in Figure 88.

Figure 88: Thermostat Display of Low Refrigerant Fault on Alerts Log

Source: EPRI

The error message was found the next morning, and cleared by cycling power to the unit. The test unit was then put into heating mode with the rooms conditioned to AHRI Standard rating conditions for heating to see if the error would re-occur. The unit did run for about an hour without actually doing any heating, as shown in Figure 88, and with the suction pressure actually below atmospheric before the unit shut down and another E41: LOW REFRIGERANT fault was recorded, as shown in Figure 89.

Figure 89: Heating Mode Test at Very Low Charge



Source: EPRI

Figure 90: Thermostat Displays of Low Refrigerant Fault in Heating Mode

Source: EPRI

Synopsis of FDD Testing

Through the course of exercising the fault detection and diagnostic features of the test unit, 12 of the 51 fault codes listed for the heat pump and 5 of the 25 fault codes listed for the furnace were triggered. (One additional fault code for the heat pump was triggered in the next part of the Phase 2 testing, and will be discussed later.) Although this is a fairly small fraction of what the system is capable of reporting, a general conclusion about the FDD features may be drawn. *The FDD system produced fault codes were mostly precise in identifying the cause when something goes wrong, but was less useful at warning the end user that something is going wrong and should be addressed.* The self-diagnostics are mainly focused on preservation of the system and shutting the system down when continued operation would lead to possibly costly damage. This is a significant improvement over what is available on most systems, but there is room for improved features that can alert the end user of performance degradation that could be remediated through maintenance and associated preventive measures that would extend efficient operation of the unit.

The FDD testing also did not follow a very broad methodology, and under different circumstances of environmental conditions or time of operation, faults could be triggered sooner than they would have occurred during this evaluation.

Alternative Refrigerant Testing

The general plan for evaluating the performance of the test unit with an alternative refrigerant was to replace the original charge of R-410A with R-32, following the existing charging instructions to a specified range of subcooling while in cooling mode, and then repeat the tests conducted in Phase 1 to provide a direct comparison. These tests included several conducted in accordance with the current rating standards, specifically AHRI Standard 210/240-2008

Table 9, which is also incorporated into the DOE regulations (DOE Title-10). In addition to the Standard tests, several performance mapping tests as a function of outdoor temperature were conducted to fill in the gaps in performance not covered under the Standard tests, and to examine performance under a more California-climate appropriate return air condition; specifically, 75°Fdb/62°Fwb versus the AHRI Standard 80°Fdb/67°Fwb.

Most of the tests outlined in Table 2729 are steady-state tests, where stable conditions are to be maintained for at least 30 minutes within specified tolerances. The exception is the I_1 test, which is an on/off cycling test at minimum speed. For variable speed systems like the test unit, the cycling period is specified to be 48 minutes off followed by 12 minutes on to complete an hour cycle. The cyclic test is conducted immediately following a G_1 test to record the steady state performance at the same conditions for comparison. Both tests are to be conducted at a return air dew point temperature that is below the operating temperature of the evaporator coil to avoid condensation, as the measurement of humidity in the supply duct is not considered to be accurate under unsteady conditions. The on/off cycling of the unit had to be done manually using the Ram Monitor program, toggling between "Cooling_Test_Mode" settings of 0 and 1.

Table 29: Planned Cooling Mode Standard Tests

Test Description	Air Entering Indoor Unit Temperature: Dry-Bulb (°F)	Air Entering Indoor Unit Temperature: Wet-Bulb (°F)	Air Entering Outdoor Unit Temperature: Dry-Bulb (°F)	Compressor Speed and Cooling Air Volume Rate
AHRI A_2 Test	80	67	95	Maximum
AHRI B_2 Test	80	67	82	Maximum
AHRI E_V Test	80	67	87	Intermediate
AHRI B_1 Test	80	67	82	Minimum
AHRI F_1 Test	80	67	67	Minimum
AHRI G_1 Test	80	Dry Coil	67	Minimum
AHRI I_1 Test (Cyclic)	80	Dry Coil	67	Minimum
AHRI Maximum Conditions	80	67	115	Maximum
Performance Mapping AHRI Indoor Conditions	80	67	75, 85, 95*, 105, 115*	Maximum & Minimum
Performance Mapping California Indoor Dry Climate	75	62	75, 85, 95, 105, 115	Maximum & Minimum

*** Mapping test conditions already included in standard tests at maximum**

Source: EPRI

As in Phase 1, the listed tests were doubled to include tests at two different levels of external resistance: 0.10 IW (24.884 Pa) and 0.45 IW (111.978 Pa). The resistance specification only applies to when the unit was operated at its maximum airflow setting. At the minimum and intermediate settings, the external resistance would be reduced while maintaining a constant duct coefficient, defined as follows:

$$C_{DUCT} = \frac{CFM}{\sqrt{IW/\rho_{SA}}} \qquad \textbf{Equation 26}$$

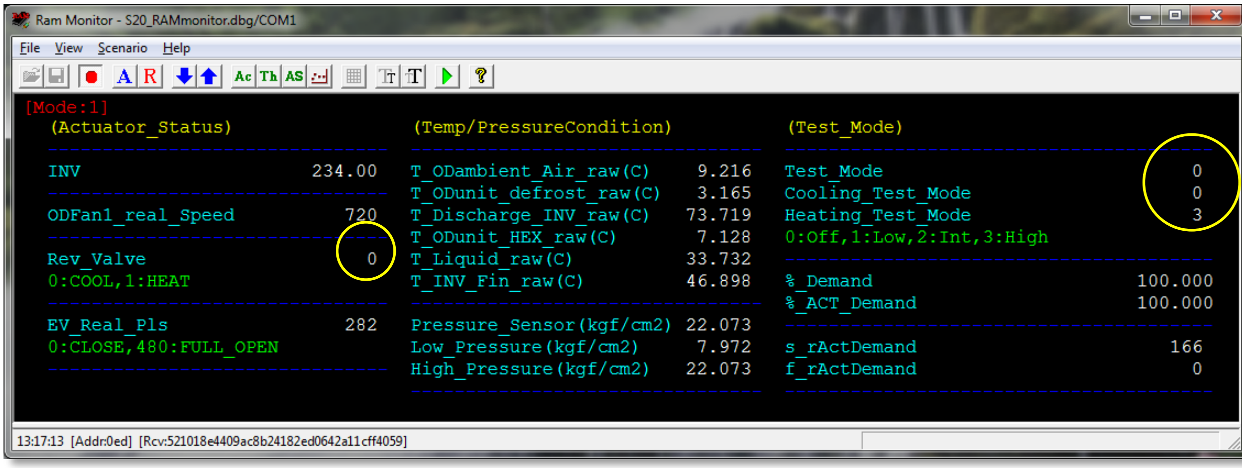
where ρ_{SA} is the supply air density. The test conducted at the maximum compressor speed and the corresponding indoor blower speed is done at the prescribed fixed external resistance, and the measured airflow rate through the indoor unit is then used to establish the duct coefficient. This is then used as a constant with this equation rearranged to calculate the appropriate external static pressure setpoint for the measured airflow rate.

The refrigerant replacement took place on March 15, 2017, although actual testing would not begin until a month later. An HVAC contractor was instructed to obtain the supply of R-32, which took some time as it is not a commonly stocked item. The remaining small amount of R-410A left in the system after the low charge FDD testing was recovered and a vacuum held on the system for several hours to confirm that no R-410A remained and that there were no leaks in the system. An initial charge of R-32 was then added to the system (~7 pounds) and the unit started up with the thermostat set in "Charge Mode". Additional refrigerant was then added in steps while watching the liquid subcooling on the outdoor unit until it reached a level specified in the installation manual (7 to 9°F [-14^{o} to -13^{o}C]). The resulting full charge was nearly the same as for the R-410A (10 pounds) although it was expected to be less based on information provided by the manufacturer (which said that charge and flow rates would be 70 percent of the R-410A levels). Other than the change in refrigerant, no other adjustments were made to the heat pump.

The Standard tests under control of the Ram Monitor program did not go as smoothly with R-32 as before with R-410A, as the test unit or the program appeared to have developed some issues that were not present in the earlier testing. The most disruptive of which was that when the unit was put into a fixed speed mode, it would run for 4-5 hours and then abruptly shut off – the "Cooling_Test_Mode" or "Heating_Test_Mode" settings would return to zero. Since many test scenarios were set up to run overnight via a script, they were usually cut short and had to be continued at a later time when the system could be put back into test mode, thus increasing the testing time. Two issues that did not impact the testing were in the Ram Monitor display. Even if the system had been put into a fixed speed condition via the "Cooling_Test_Mode" or "Heating_Test_Mode" controls, the "Test_Mode" indicator would always stay at zero. (For comparison, note the "1" indication in Source: EPRI

Figure 19 for "Test_Mode".) Also, when in heating mode, the reversing valve indicator "Rev_Valve" also stayed at zero when it should have indicated 1. Figure 91 shows an example shot of the Ram Monitor screen in high heating test mode, and zero indications for "Test_Mode" and "Rev_Valve".

Figure 91: Ram Monitor Screen in Heating Mode in Phase 2



Source: EPRI

It is not clear that this change in behavior is due to the refrigerant replacement, or possibly some issue caused by the strain put on the system during the FDD testing, or just a software anomaly. The issue was raised with the manufacturer, but they did not have an answer at that time. Testing could still be conducted; but it took longer.

The other observation was that the system operation appeared to be unstable. Several of the test conditions were repeated, and the response of the system was often not consistent. For example, an hour into a test with steady-state environmental conditions and the unit controlled to fixed speed via Ram Monitor, the system speed would unexpectedly shift to a lower level. The tabulated results that follow are averages over the multiple tests if run and may not be repeatable. A system designed specifically to use R-32 would likely show more consistent and optimized performance than this unit with just a drop-in replacement.

Cooling Mode Standard Test Results

The majority of the cooling mode tests were conducted between April 17 and May 12, 2017. The results from the standard tests with R-32 are summarized in Source: EPRI

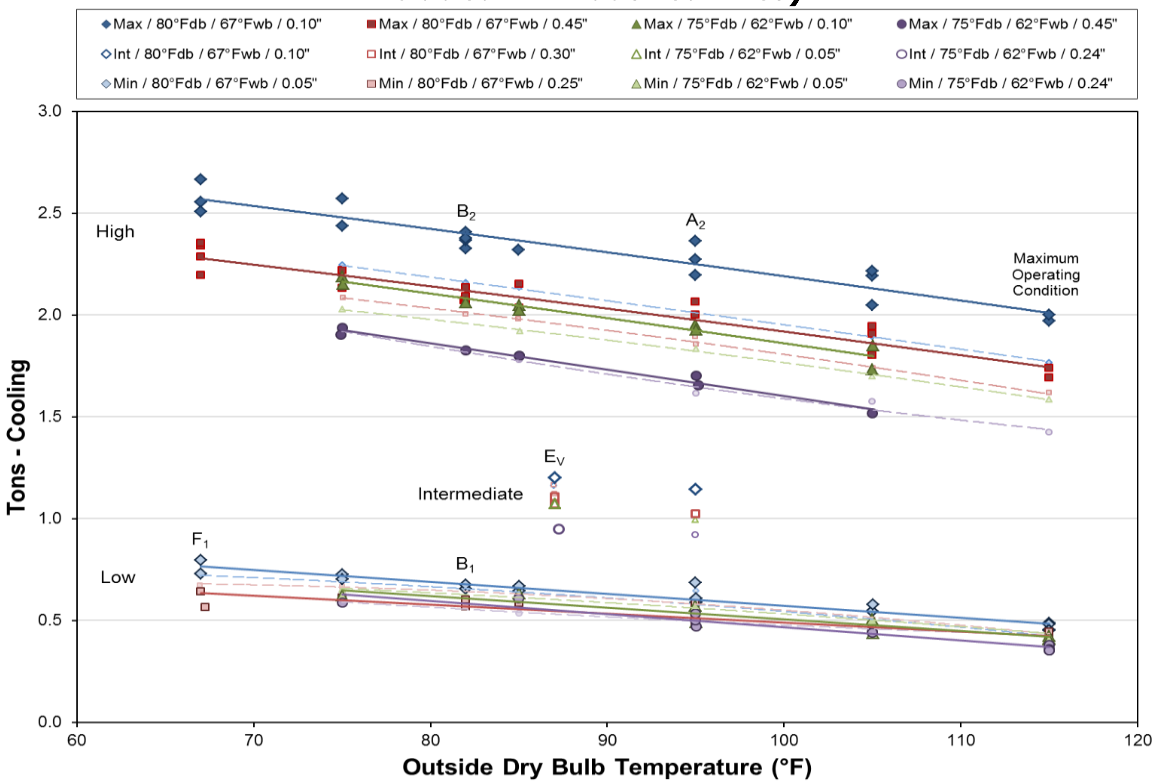
Table 30 (corresponding to the earlier results with R-410A presented in Table 11), and the relative percent change in the metrics between the two refrigerants is given in Table 31. As in Phase 1, the performance metrics are better viewed in graphic form. Three figures are provided that trend capacity (Figure 92), power (Figure 93), and efficiency[4] () as a function of the outdoor dry bulb temperature. The data in the figures is again grouped by the two return air conditions and two external resistance levels into four sets, and these are also split into groups for the High, Intermediate, and Low setting of the Ram Monitor "Cooling_Test_Mode". The trends from the previous testing with R-410A are included for direct comparison and indicated with small symbols and dashed lines.

The resulting capacity metrics followed a consistent pattern with the earlier tests, with the best performance with the more humid return air and lower external resistance. At the High

[4] Energy Efficiency Ratio (EER) (Btu/Wh) = 3.41214 x COP (Coefficient of Power). To convert the scale in Figure 21 to COP, divide by 3.42124.

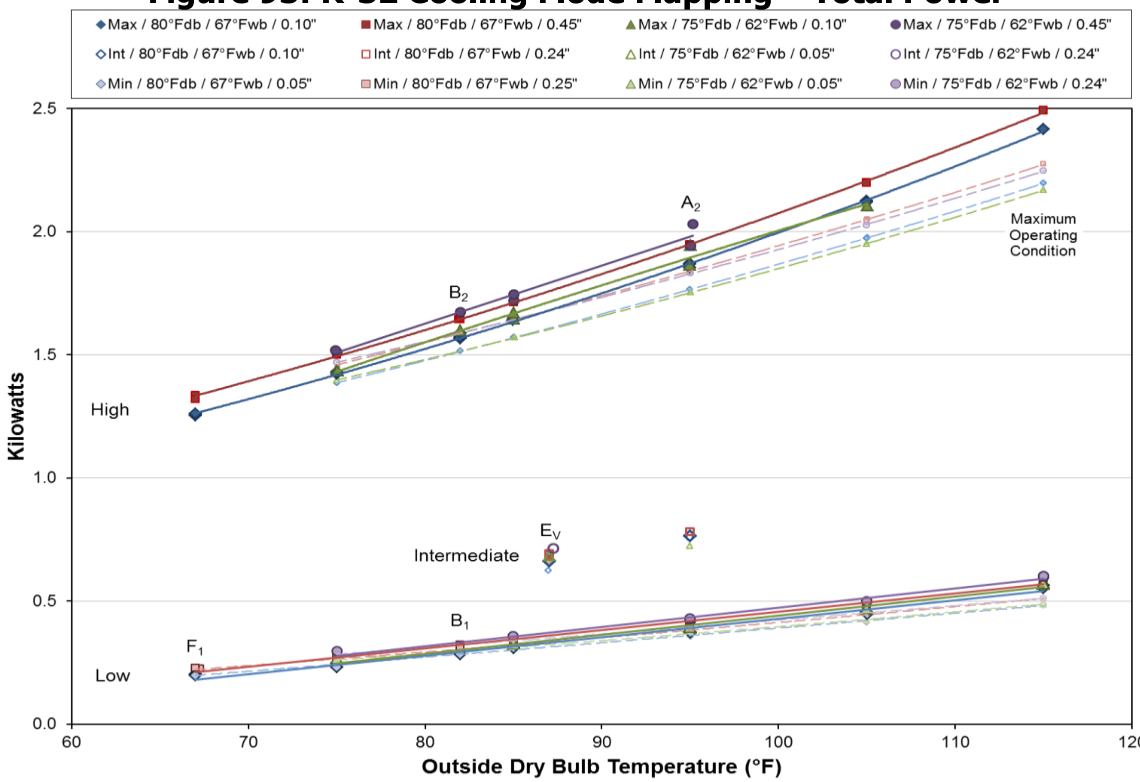
setting, the values were mostly higher than what was measured with R-410A, except for the results with the dry return air and higher resistance which produced nearly identical performance with the earlier results. However, this higher capacity was accompanied by an increase in power consumption, with the resulting change in system efficiency being a mix of values both higher and lower than the system with R-410A.

Figure 92: R-32 Cooling Mode Mapping – Capacity - (Phase 1 results with R-410A included with dashed lines)



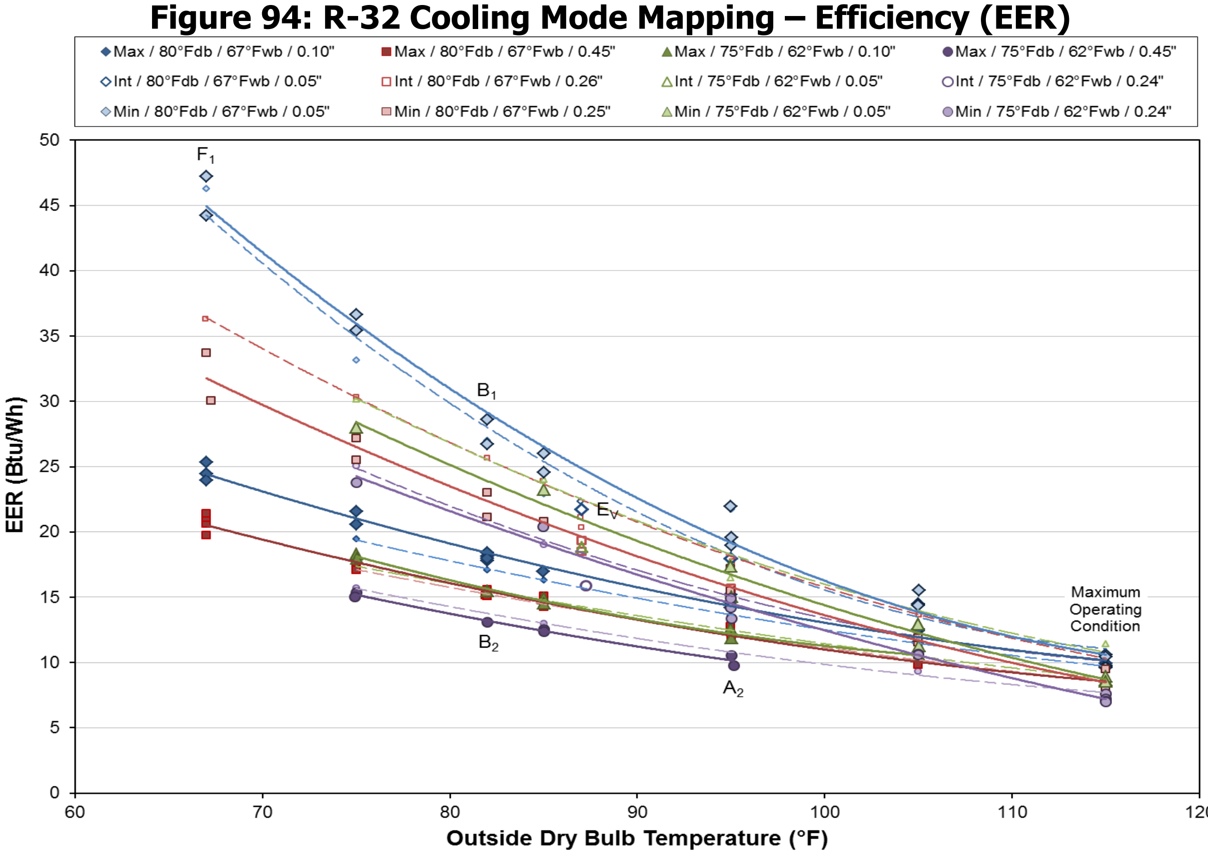
Source: EPRI

Figure 93: R-32 Cooling Mode Mapping – Total Power



Source: EPRI

Figure 94: R-32 Cooling Mode Mapping – Efficiency (EER)



Source: EPRI

Table 30: Cooling Mode Standard Test Results (R-32)

Test Description	Compressor Speed and Cooling Air Volume Rate	Outside Air Tdb (°F)	Return Air Tdb (°F)	Return Air Twb (°F)	Ext. Res. (IW)	Supply Air Tdb (°F)	Cooling Capacity (Tons)	Power (kW) IDU	Power (kW) ODU	Power (kW) Total	EER (Btu/Wh)	Indoor Blower Airflow (CFM)	Indoor Blower Speed (RPM)	Outdoor Fan Airflow (CFM)	Outdoor Fan Speed (RPM)
A_2	Maximum	95	80	67	0.10	57.1	2.28	0.091	1.77	1.86	14.7	740	565	3,311	649
					0.45	56.8	2.00	0.167	1.78	1.94	12.3	641	781	3,310	649
B_2	Maximum	82	80	67	0.10	56.1	2.37	0.091	1.48	1.57	18.1	740	566	3,301	649
					0.45	55.7	2.09	0.167	1.48	1.65	15.2	640	782	3,300	649
E_V	Intermediate	87	80	67	0.05	60.3	1.20	0.044	0.62	0.66	21.8	541	427	2,552	499
					0.24	60.0	1.07	0.076	0.62	0.69	18.5	476	590	2,553	499
B_1	Minimum	82	80	67	0.05	68.0	0.67	0.042	0.25	0.29	27.7	545	411	2,550	499
					0.25	67.9	0.58	0.073	0.24	0.32	22.0	480	580	2,548	499
F_1	Minimum	67	80	67	0.05	66.9	0.76	0.042	0.16	0.20	45.7	545	413	2,530	499
					0.25	67.2	0.60	0.074	0.15	0.23	31.9	481	582	2,527	499
G_1	Minimum	67	80	61	0.05	67.0	0.72	0.042	0.14	0.18	47.7	540	410	2,505	499
					0.24	67.0	0.63	0.074	0.13	0.20	37.3	477	581	2,514	499
I_1	Minimum (Cyclic)	*67*	*80*	*61*	*0.05*		0.114			0.050	27.4	Cyclic			
					0.24		0.102			0.070	17.5				
Performance Mapping	Maximum	115	80	67	0.10	59.1	1.99	0.091	2.32	2.42	9.9	743	566	3,320	650
					0.45	58.8	1.72	0.168	2.33	2.49	8.3	645	782	3,320	650
			75	62	Unit tripped on high discharge temperature										
		105	80	67	0.10	58.1	2.15	0.091	2.03	2.12	12.2	742	565	3,311	650
					0.45	57.8	1.88	0.167	2.03	2.20	10.3	642	781	3,311	650
			75	62	0.10	53.2	1.85	0.090	2.02	2.11	10.5	741	562	3,314	650
					0.45	54.1	1.52	0.166	2.40	2.56	7.1	648	779	3,325	650
		95	80	67	0.10	57.1	2.28	0.091	1.77	1.86	14.7	740	565	3,311	649
					0.45	56.8	2.00	0.167	1.78	1.94	12.3	641	781	3,310	649
			75	62	0.10	52.6	1.94	0.090	1.82	1.91	12.2	742	562	3,317	649
					0.45	52.5	1.68	0.166	1.82	1.99	10.1	642	778	3,318	649
		85	80	67	0.10	56.4	2.32	0.090	1.55	1.64	17.0	737	563	3,304	649
					0.45	56.0	2.09	0.167	1.55	1.71	14.7	639	780	3,304	649
			75	62	0.10	51.8	2.04	0.090	1.57	1.66	14.7	741	562	3,305	649
					0.45	51.6	1.80	0.166	1.57	1.73	12.5	642	778	3,306	649
		75	80	67	0.10	55.6	2.50	0.091	1.34	1.43	21.1	742	566	3,295	649
					0.45	55.3	2.18	0.167	1.34	1.50	17.4	641	783	3,295	649
			75	62	0.10	50.9	2.17	0.089	1.35	1.44	18.1	738	560	3,292	649
					0.45	50.6	1.92	0.165	1.35	1.52	15.2	639	777	3,295	649
	Minimum	115	80	67	0.05	71.9	0.47	0.042	0.52	0.56	10.1	548	410	2,578	500
					0.25	71.9	0.41	0.074	0.51	0.58	8.4	484	581	2,576	500
			75	62	0.05	67.7	0.41	0.042	0.53	0.57	8.7	544	409	2,569	500
					0.25	67.4	0.36	0.073	0.53	0.60	7.2	479	577	2,569	500
		105	80	67	0.05	70.3	0.56	0.042	0.41	0.45	14.8	547	410	2,565	499
					0.25	70.5	0.47	0.074	0.41	0.48	11.7	483	581	2,565	499
			75	62	0.05	65.9	0.48	0.042	0.43	0.47	12.4	543	409	2,560	499
					0.24	66.1	0.44	0.073	0.43	0.50	10.6	478	575	2,555	499
		95	80	67	0.05	69.2	0.63	0.042	0.33	0.37	20.2	546	410	2,556	499
					0.25	69.1	0.53	0.074	0.33	0.40	16.0	481	580	2,557	499
			75	62	0.05	64.3	0.53	0.042	0.35	0.39	16.2	542	411	2,560	499
					0.24	64.4	0.50	0.073	0.35	0.43	14.2	477	577	2,557	499
		85	80	67	0.05	68.3	0.66	0.042	0.27	0.31	25.3	545	411	2,549	499
					0.25	68.2	0.59	0.073	0.26	0.34	20.8	481	580	2,559	499
			75	62	0.05	63.3	0.63	0.042	0.28	0.33	23.3	542	411	2,542	499
					0.25	63.8	0.61	0.073	0.28	0.36	20.4	478	578	2,544	499
		75	80	67	0.05	67.4	0.71	0.042	0.20	0.24	36.0	545	413	2,540	499
					0.25	67.5	0.60	0.074	0.20	0.27	26.3	481	581	2,539	499
			75	62	0.05	62.4	0.62	0.042	0.22	0.27	28.0	540	412	2,532	499
					0.24	61.6	0.59	0.074	0.22	0.30	23.8	476	579	2,531	499

Source: EPRI

Table 31: Relative Change from R-410A to R-32 in Cooling Mode

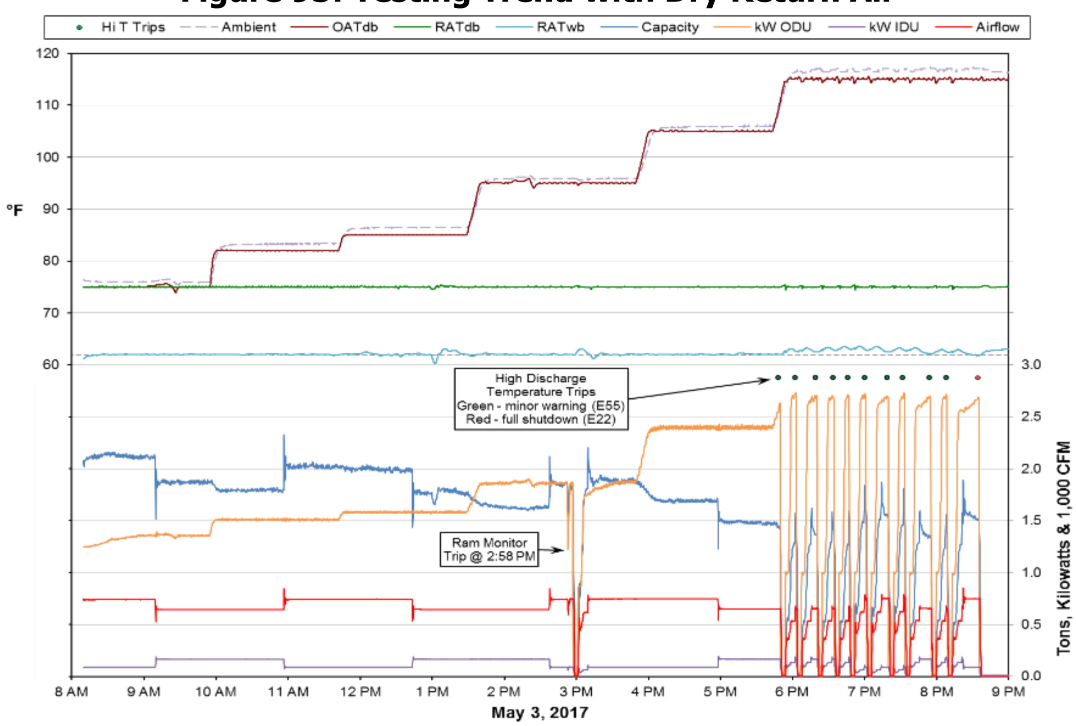
Test Description	Compressor Speed and Cooling Air Volume Rate	Outside Air Tdb (°F)	Return Air Tdb (°F)	Return Air Twb (°F)	Ext. Res. (IW)	Supply Air Tdb (°F)	Cooling Capacity (Tons)	Power (kW) IDU	Power (kW) ODU	Power (kW) Total	EER (Btu/Wh)	Indoor Blower Airflow (CFM)	Indoor Blower Speed (RPM)	Outdoor Fan Airflow (CFM)	Outdoor Fan Speed (RPM)
A_2	Maximum	95	80	67	0.10	+3%	+14%	+31%	+5%	+6%	+8%	+23%	+10%	+1%	-0%
					0.45	+1%	+6%	+17%	+5%	+6%	+1%	+11%	+3%	+1%	+0%
B_2	Maximum	82	80	67	0.10	+4%	+10%	+31%	+3%	+4%	+6%	+23%	+10%	+1%	-0%
					0.45	+2%	+4%	+17%	+2%	+4%	+1%	+11%	+3%	+1%	-0%
E_V	Intermediate	87	80	67	0.08	-1%	+4%	-15%	+8%	+6%	-3%	+7%	-8%	-0%	-0%
					0.28	-2%	-6%	-14%	+8%	+5%	-11%	-2%	-7%	+1%	-0%
B_1	Minimum	82	80	67	0.08	+2%	+3%	-1%	+1%	+0%	+2%	+25%	-4%	+2%	+0%
					0.24	+2%	-10%	+15%	+2%	+4%	-14%	+12%	+4%	+1%	-0%
F_1	Minimum	67	80	67	0.08	+3%	+3%	-1%	+6%	+5%	-1%	+25%	-4%	+2%	-0%
					0.24	+3%	-10%	+16%	-4%	+2%	-12%	+13%	+5%	+1%	+0%
G_1	Minimum	67	80	61	0.08	+1%	+34%	-1%	+7%	+5%	+27%	+22%	-4%	+0%	+0%
					0.23	+3%	+12%	+20%	-12%	-3%	+15%	+15%	+7%	+1%	-0%
I_1	Minimum (Cyclic)	*67*	*80*	*61*	0.08		+9%			-28%	+51%	Cyclic			
					0.23		+3%			-3%	+6%				
Performance Mapping	Maximum	115	80	67	0.10	+4%	+12%	+32%	+9%	+10%	+2%	+23%	+11%	+1%	+0%
					0.45	+3%	+6%	+18%	+9%	+10%	-3%	+12%	+4%	+1%	-0%
			75	62	R-32 unit tripped on high discharge temperature										
		105	80	67	0.10	+4%	+13%	+31%	+7%	+7%	+5%	+23%	+10%	+1%	-0%
					0.45	+2%	+9%	+18%	+7%	+7%	+1%	+11%	+3%	+1%	-0%
			75	62	0.10	+4%	+9%	+30%	+7%	+8%	+1%	+23%	+10%	+1%	+0%
					0.45	+5%	-4%	+17%	+27%	+26%	-24%	+13%	+3%	+1%	+0%
		95	80	67	0.10	+3%	+14%	+31%	+5%	+6%	+8%	+23%	+10%	+1%	-0%
					0.45	+1%	+6%	+17%	+5%	+6%	+1%	+11%	+3%	+1%	+0%
			75	62	0.10	+5%	+6%	+30%	+8%	+9%	-3%	+23%	+10%	+1%	+0%
					0.45	+3%	+4%	+17%	+8%	+8%	-4%	+12%	+3%	+1%	+0%
		85	80	67	0.10	+4%	+9%	+29%	+3%	+4%	+4%	+22%	+9%	+1%	-0%
					0.45	+2%	+6%	+17%	+3%	+4%	+1%	+11%	+3%	+1%	-0%
			75	62	0.10	+5%	+6%	+30%	+4%	+6%	+0%	+24%	+10%	+1%	+0%
					0.45	+4%	+1%	+17%	+4%	+6%	-4%	+13%	+3%	+1%	+0%
		75	80	67	0.10	+4%	+12%	+32%	+1%	+3%	+8%	+23%	+10%	+1%	-0%
					0.45	+2%	+4%	+18%	+1%	+3%	+1%	+11%	+3%	+1%	-0%
			75	62	0.10	+3%	+7%	+29%	+1%	+3%	+4%	+23%	+10%	+1%	-0%
					0.45	+2%	-0%	+18%	+2%	+3%	-3%	+12%	+3%	+1%	-0%
	Minimum	115	80	67	0.08	+4%	+12%	-1%	+17%	+15%	-3%	+24%	-4%	+1%	+0%
					0.24	+3%	-8%	+19%	+13%	+14%	-19%	+16%	+7%	+2%	+0%
			75	62	0.08	+4%	-5%	+0%	+18%	+17%	-19%	+25%	-3%	-0%	+0%
					0.24	+4%	-15%	+18%	+18%	+18%	-28%	+14%	+6%	-0%	+0%
		105	80	67	0.08	+4%	+10%	-1%	+11%	+10%	+1%	+24%	-4%	+1%	+0%
					0.24	+2%	-6%	+19%	+8%	+9%	-14%	+15%	+7%	+2%	-0%
			75	62	0.08	+3%	-5%	+0%	+12%	+11%	-15%	+24%	-3%	-0%	+0%
					0.24	+5%	-4%	+18%	+9%	+10%	-13%	+14%	+6%	-0%	+0%
		95	80	67	0.08	+5%	-2%	-1%	+6%	+5%	-7%	+24%	-4%	+1%	+0%
					0.24	+1%	-6%	+19%	+2%	+4%	-10%	+15%	+6%	+2%	-0%
			75	62	0.08	+2%	-2%	+1%	+8%	+7%	-8%	+24%	-3%	-0%	-0%
					0.24	+4%	+1%	+18%	+6%	+8%	-7%	+14%	+6%	+0%	+0%
		85	80	67	0.08	+2%	+13%	-2%	+3%	+2%	+10%	+25%	-4%	+2%	+0%
					0.24	+2%	-9%	+15%	+2%	+5%	-13%	+13%	+4%	+1%	-0%
			75	62	0.08	+1%	+0%	-1%	+4%	+3%	-3%	+22%	-4%	+1%	-0%
					0.24	+5%	+14%	+19%	+3%	+6%	+7%	+15%	+6%	+0%	+0%
		75	80	67	0.08	+2%	+4%	-1%	-5%	-5%	+9%	+25%	-4%	+2%	+0%
					0.24	+3%	-10%	+15%	+0%	+4%	-13%	+13%	+5%	+1%	-0%
			75	62	0.08	+1%	-5%	-1%	+3%	+3%	-7%	+22%	-4%	+1%	+0%
					0.24	+2%	-1%	+19%	+0%	+4%	-5%	+15%	+6%	+0%	+0%

Source: EPRI

One issue that came up over the course of this steady-state testing was that the system would not operate at the highest temperature (115°F [46°C]) with the drier return air. Having dry return air means there is less latent load or condensation occurring that takes place at mainly a constant temperature. Without that, capacity has to be made up in sensible temperature reduction, which requires a lower temperature for the evaporating refrigerant. This results in a higher compression ratio and much higher compressor discharge temperatures, which is enhanced with the R-32 and its slightly higher saturation curve. The Heat Pump Installation & Service Reference (Daikin, 2016) mentions high pressure faults can occur at pressures above 490-PSI and high discharge temperature faults can occur above 200°F (93°C). Measured liquid pressures during these tests reached nearly 500 PSI, and the compressor discharge temperature reached 250°F (121°C). In the Phase 1 tests with R-410A at the same conditions, measured liquid pressures were at about 485 PSI, which also supports a 490-PSI threshold for a fault condition; and discharge temperatures were also below the mentioned threshold at about 198°F (92°C).

Figure 95 presents a trend of the testing done with the Ram Monitor program set with "Cooling_Test_Mode" at High, and the return air at 75°Fdb and 62°Fwb. At each outdoor temperature step, the external resistance was toggled between 0.10 IW and 0.45 IW. There was a unit shutdown at 2:58 PM when the Ram Monitor program put the "Cooling_Test_Mode" to "Off", but this was corrected and the test continued. At the highest ambient temperature of 115°F (46°C), the test unit entered a cycling mode tripping off with high discharge temperature faults. Ten minor warning faults (code E55: HI DIS TEMP MIN) were recorded to the heat pump's Diagnostics page and Alerts Log from which the system would try to recover, before finally giving up for good and recording fault code E22: HI DISCH TEMP and displaying a severe fault message on the front of the thermostat, as shown in Figure 96.

Figure 95: Testing Trend with Dry Return Air



Source: EPRI

Figure 96: Thermostat Display of High Discharge Temperature Fault

Source: EPRI

Because operation at this test condition could not be achieved, corresponding data could not be included in the tables and figures.

Calculation of Seasonal Energy Efficiency Ratio (SEER)

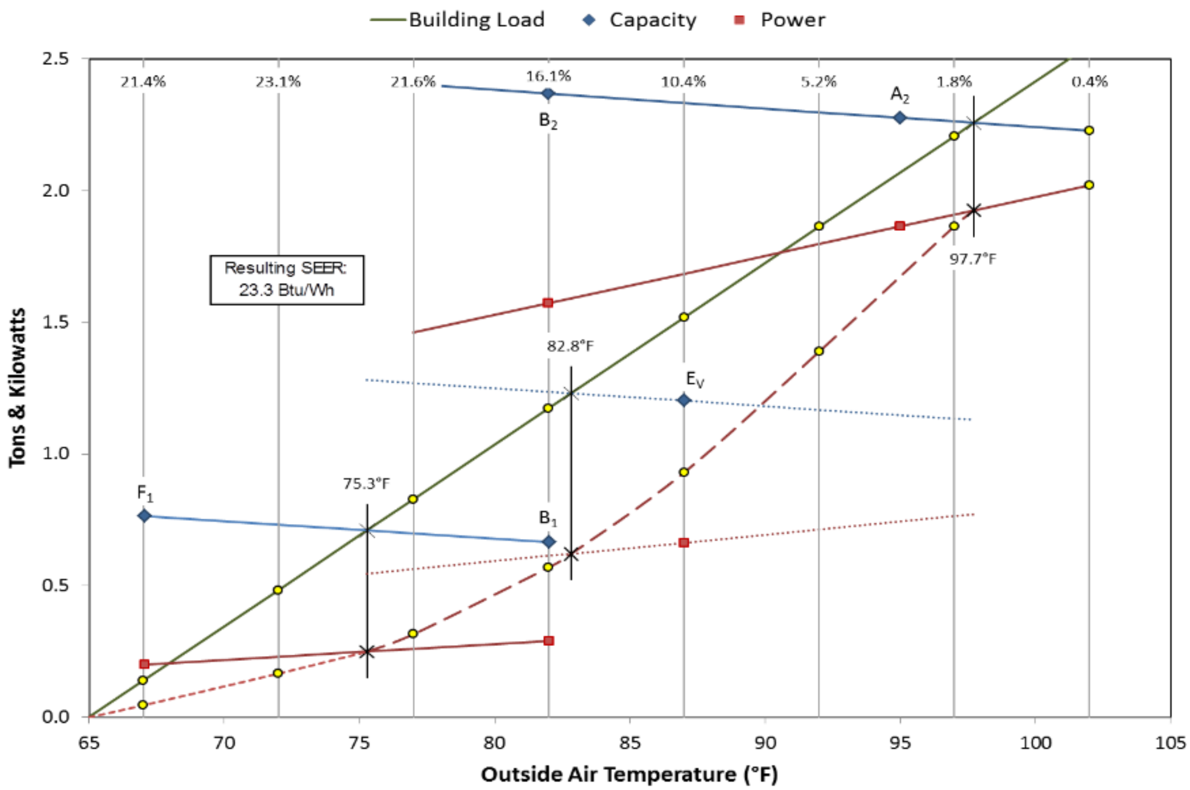
The calculation of SEER was performed in the same manner as in Phase 1. Repeating the description from Phase 1, it is a bin method calculation using representative temperatures in 5°F (2.8°C) increments to calculate the cooling load and power consumed by the air conditioner at those temperatures, which are then weighted by the number of hours in a particular cooling season that the temperatures in the bins occur. The binned ton-hours and kWh are then summed, and the sums divided into each other to determine the SEER.

The building load is defined in the AHRI and DOE Standards as:

$$BL = \frac{(OAT-65)}{95-65} \times \frac{\dot{Q}_c^{k=2}}{1.1}$$ **Equation 27**

When graphed against outside temperature (see Figure 2396), this produces a line that extends from zero at 65°F (18°C) through the standard rated capacity (A_2 test) divided by 1.1 at 95°F (35°C). The next step is to determine how much power the system will use to satisfy this building load, and that involves finding the temperatures where this building load line intersects the system capacity trends. At maximum speed, the system capacity is represented by a line drawn through the capacity values from the A_2 and B_2 tests. In Figure 97, which uses the results from the Phase 2 cooling mode tests, the intersection with the building load line occurs at a temperature of 97.7°F (36.7°C). At outside temperatures above this, the system will be running constantly at maximum capacity, and its power is that extrapolated from the line drawn through the power values from the A_2 and B_2 tests. Under these conditions, the building load is greater than the system capacity, so it will not be able to maintain the indoor thermostat setpoint temperature.

Figure 97: Test Data Used to Derive SEER (at 0.10 IW)

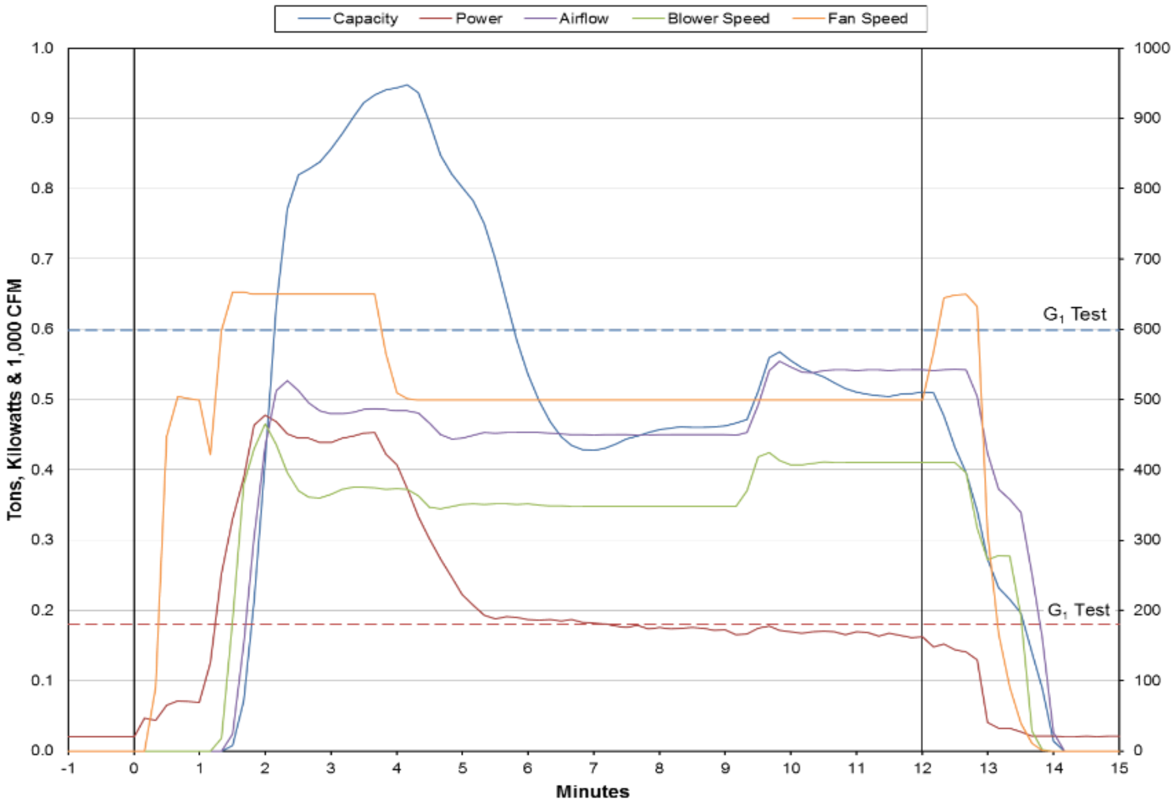


Source: EPRI

The steady-state minimum speed capacity is represented by a line drawn through the capacity measurements from the B_1 and F_1 tests, which in this case intersects the building load line at 75.3°F (23.9°C). Below this temperature, the minimum speed capacity will be greater than the building load and the air conditioner is assumed to cycle between off and minimum speed. To determine the power for the temperature bin, the calculation involves finding the fraction of time that the system will need to operate to meet the load, and also applies a degradation coefficient to account for the non-steady-state operation.

The degradation coefficient may be derived from the optional G_1 and I_1 tests, which compares the integrated total cooling and power consumed during a 48 minute off / 12 minute on cycle (I_1) to what it would have been under steady-state operation (G_1). In a typical fixed speed compressor system, the capacity rises from zero at the start of an on cycle to asymptotically approach the steady-state capacity; but this system behaves differently. An example of the on cycle for this variable capacity unit is given in Figure 98 (the third of three cycles conducted following the G_1 test conducted in Phase 2). The trends show that for the first minute after the unit is triggered to come on, only the outdoor unit fan is operating. Eventually, the compressor and indoor blower begin to operate, but at a level higher than minimum to get the system moving. The capacity and power during this period actually exceed their steady-state values. After a few minutes, the system begins to slow down to the point where the total power is on par with that from the steady-state test. Capacity then dips below the steady-state value before rebounding when the blower speed is increased. When the system is triggered to turn off at the end of 12 minutes, the outdoor unit fan speeds up for about one minute; after which the compressor and indoor blower finally turn off. Because of this unusual operational trend, the actual degradation coefficient calculated from these tests was 0.51; but the standard caps it at 0.25; making these two tests unnecessary. If the uncorrected value was used, it would have reduced the SEER result by about 2 percent.

Figure 98: On Cycle from I_1 Test



Source: EPRI

Between the minimum and maximum speed intersection temperatures (75.3°F to 97.7°F [23.9ºC to 36.7ºC]), the system is assumed to be running at a variable intermediate speed. To determine the power consumption at these temperatures, pseudo-performance trend lines are drawn through the capacity and power values measured from the single intermediate speed test (E_V). These trends (shown as dotted lines in Figure 23) are weighted averages of the slopes of the maximum and minimum speed trend lines. Once again, the intersection of the capacity line and the building load line is determined, and in this case, it is at 82.8°F (28.3ºC).

The EER of the system is then determined at each of the three intersection temperatures, and the three points are used to create a second-order curve fit of EER as a function of outside temperature. When combined with the building load line, the power for the intermediate speed temperatures can be determined. The trend of power derived from the building load and EER curve fit is shown as the dashed parabolic curve in Figure 23, showing that it passes through the three points on the minimum, pseudo-intermediate, and maximum power trend lines at the intersection temperatures.

Finally, the load or capacity and power are calculated for each of the bin temperatures (indicated by the yellow circles), and multiplied by the weighting factors prescribed in the AHRI Standard, as shown at the top of the temperature lines. The end result of the bin calculation is a SEER of 23.3 when the external resistance at maximum speed is held to

0.10 IW; reducing to a SEER of 19.4 when the external resistance at maximum speed is raised to 0.45 IW.

In comparison with the Phase 1 tests with R-410A, the results (summarized in Table 32) show a generally small decrease in SEER for both levels of external resistance. (Note that this external resistance is only valid for the highest fan speed, and was controlled to decrease with the square of the airflow rate for the intermediate and low speed tests per Equation 2.)

Table 32: Comparison of Calculated SEER for R-410A and R-32

SEER	R-410	R-32	Change
0.01 IW	24.1	23.3	–3%
Change	–9%	–17%	

Source: EPRI

Cooling Mode Dynamic Test Results

Several tests were conducted using the dynamic algorithm for controlling the indoor room temperature developed in Phase 1 in order to observe how the system's own controls will operate the unit. As before, a sensible building load was calculated from the current measurements of inside and outside temperatures, and combined with constants derived from a steady-state test at the design conditions, using the following equation:

$$\mathrm{BL_s} = \dot{\mathrm{Q}}\mathrm{s_D} \times \frac{[(\mathrm{OAT}-\mathrm{OAT_0})-(\mathrm{RAT}-\mathrm{RAT_D})]}{\mathrm{OAT_D}-\mathrm{OAT_0}} \qquad \textbf{Equation 28}$$

where:

- BL_S = Sensible Building Load (tons)
- OAT = Current measured outside dry bulb temperature (°F)
- OAT_0 = Outside dry bulb temperature selected for zero BL_S (constant, 65°F [18°C])
- OAT_D = Design outside dry bulb temperature corresponding to the design cooling load (constant, 95°F [35°C])
- RAT = Current measured room or return dry bulb temperature (°F)
- RAT_D = Cooling design interior dry bulb temperature or thermostat set point (constant, 72°F [22°C])
- $\dot{Q}s_D$ = Sensible cooling capacity of the subject air conditioning unit at maximum speed under design outside (OAT_D) and inside (RAT_D) conditions (constant, tons)

The current measured sensible capacity of the system is then subtracted from the building load to derive a change in the room temperature set point.

$$\mathrm{BL_s} - \dot{\mathrm{Q}}_\mathrm{s} = \mathrm{m} \times \mathrm{c_p} \times \frac{\Delta\mathrm{RAT}}{\Delta\mathrm{t}} \qquad \textbf{Equation 29}$$

where:

m = mass of air and other materials in the space

c_p = specific heat of air and other materials in the space (0.24 Btu/lb-°F for standard air)

t = time (in appropriate units)

One minor adjustment was made to the algorithm from Phase 1 in that the concept of a volume of air was discarded in favor of a straight capacitance number (the product of mass and specific heat shown in Equation **29**). For comparison, a house volume of 10,000 cubic feet of air translates to a capacitance of 180 Btu/°F. In this form, the capacitance can be inclusive of other masses in the space that act to slow down the rate of temperature change. The value of capacitance used in this phase of testing was also more than doubled to 400 Btu/°F (or 200 Btu/°F per ton of rated cooling capacity) to better dampen out larger temperature swings.

$\mathrm{BL_s} - \dot{\mathrm{Q}}_\mathrm{s} = \mathrm{m} \times \mathrm{c_p} \times \frac{\Delta\mathrm{RAT}}{\Delta\mathrm{t}}$ **Equation 29** can be simplified to:

$$\mathrm{RAT(t + \Delta t) = RAT(t) + \Delta t} \times \frac{\mathrm{BL_S} - \dot{\mathrm{Q}}_\mathrm{S}}{\mathrm{C}}$$ **Equation 30**

where:

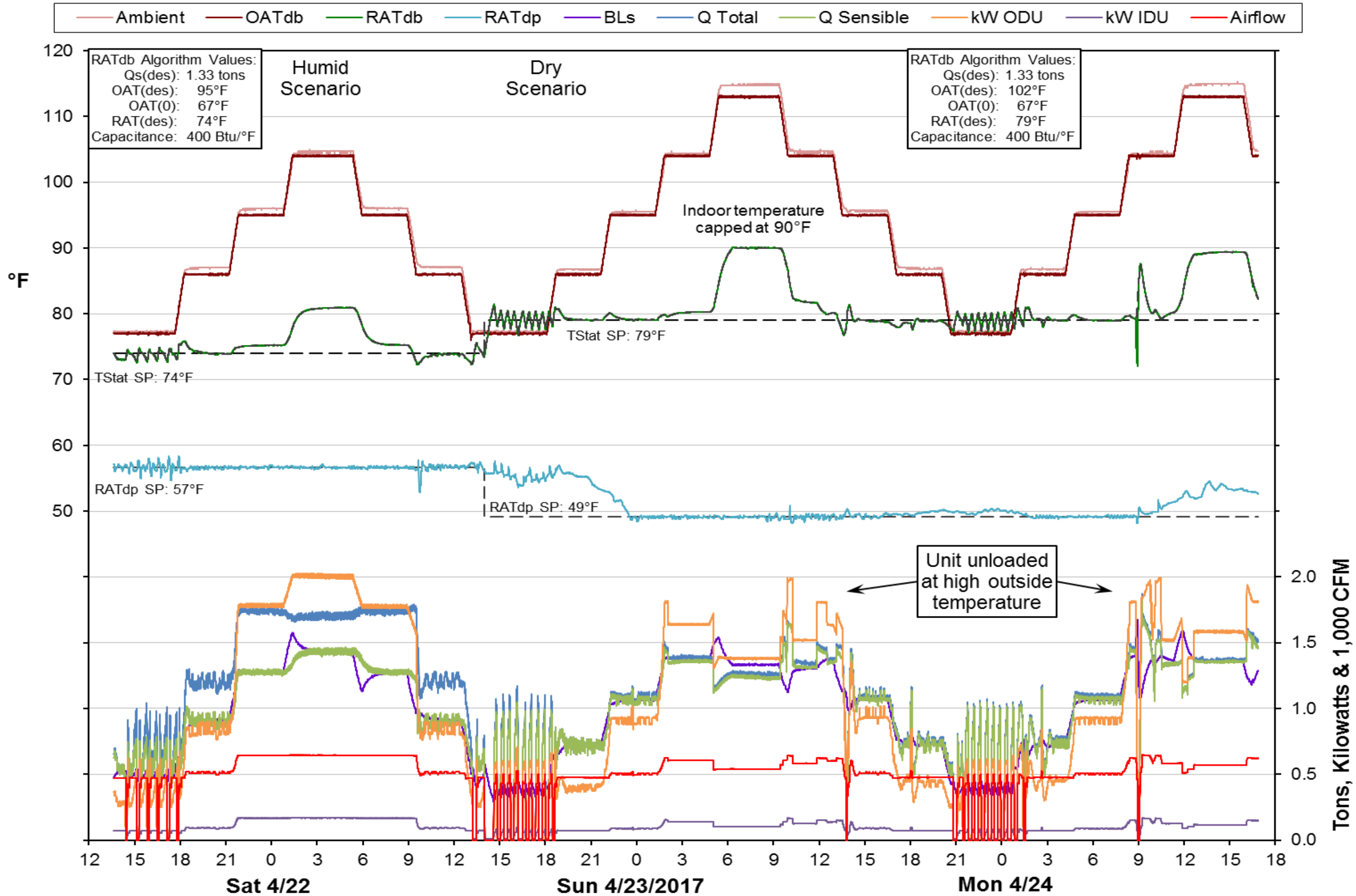
C = capacitance of the space (Btu/°F), product of mass and specific heat

The return air temperature term on the left side of the equation is then applied to the temperature set point for the space conditioning system. For all of the dynamic tests, Δt was 1 second, with all measurements and calculations done on this short time basis.

As in Phase 1, there are two ways to run the dynamic mode tests: semi-dynamic where the outdoor temperature is held constant at specific levels; and fully dynamic where the outdoor temperature follows a typical outdoor trend for a hot day. The first dynamic test run was a semi-dynamic test following the conditions specified in a draft Canadian Standards Association (CSA) procedure for testing variable speed heat pumps (Canadian Standards Association, 2017). The testing lab was set up to run a script over a weekend with the test unit controlled by its own thermostat in the indoor room. The results from this test are shown in Figure 99. (Note: "Ambient" refers to the outside temperature sensor of the test unit that was recorded via Ram Monitor, while "OATdb" is the average outside temperature from the 16 individual sensors around the unit.) The test was begun on a Friday afternoon with the indoor conditions based on the draft standard's "humid" conditions with a nominal set point of 74°Fdb and 63°Fwb (57°Fdp), the thermostat was set to 74°F (23°C), and the outdoor temperature was held at levels of 77, 86, 95, and 104°F (25, 30, 35, and 40°C) (5°C increments) for three hours at each step, first rising then falling. The temperature control algorithm and Equation **30** were set to vary the indoor temperature in response to the measured sensible capacity of the unit with the constants as shown in the figure.

The results from this first phase may be interpreted from the data shown in Figure 100. At the 77°F (25°C) outside temperature, the unit cycles to meet load. At 86°F (30°C), the operation settles out to a narrow range of operating speeds, switching back and forth between those speeds. At 95°F (35°C), the system reached its full speed and was just able to maintain the room temperature within a small tolerance band. Finally, at 104°F (40°C), the unit was still at high speed, but without enough capacity to maintain the thermostat set point, resulting in the indoor temperature rising to the point where the building load was in equilibrium with the sensible capacity of the test unit.

Figure 99: Cooling Dynamic Test #2-1 (Semi-Dynamic, R-32)



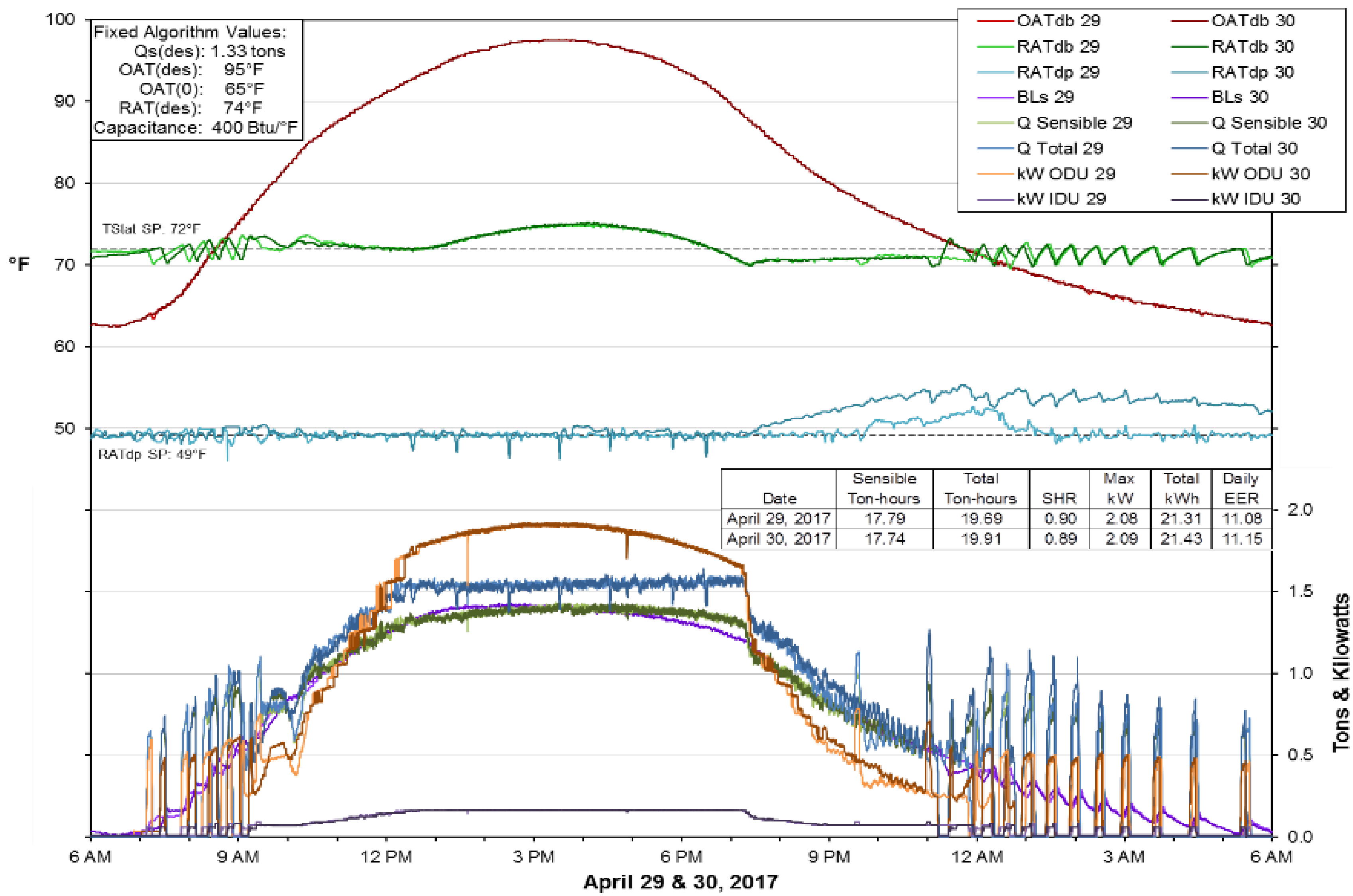
Source: EPRI

A little after noon on Saturday, the settings were changed to the “Dry” climate scenario. The thermostat set point was raised to 79°F (26°C), the nominal return temperature was based on 79°Fdb and 61°Fwb (49°Fdp), the design outdoor temperature for the control algorithm was raised to 102°F (39°C) from 95°F (35°C), and an additional temperature step of 113°F (45°C) was added to the outdoor temperature script. With the very dry return air, there was essentially no latent load on the system, and the trends for total and sensible cooling are nearly identical. The resulting unit operation was similar to the previous results, up until the unit reached the temperature plateaus above 100°F (38°C). Likely due to the higher compressor discharge pressures and temperatures with the alternative refrigerant, under extreme high outside temperatures, the system reduced its speed and capacity to avoid compressor overload. This reduction in capacity when it was most needed caused the indoor temperature to rise to an uncomfortable level. (A 90°F [32°C] cap had been applied to the algorithm; otherwise the indoor temperature would have continued to rise, at least for the first incidence of this temperature.) This unloading at high temperature occurred during both cycles of the higher outside temperatures.

The next dynamic test was a full dynamic profile using a more realistic outdoor temperature trend. The temperature profile was built with a low of 62.5°F (16.7°C) (which is below the chosen zero load temperature of 65°F (18°C) and will allow the indoor temperature to drop below the thermostat set point), and a high of 97.5°F (36.7°C). This profile was run over both days of a weekend to observe the repeatability in the system’s response. The resulting trend of performance shown in Figure 100 was again very repeatable (right down to the times at which it would cycle on under low load conditions), and very similar to what was produced in Phase 1, except with less extreme indoor temperature changes due to the chosen increase in the capacitance constant.

Due to the observed unloading from the semi-dynamic test in Figure 99, another fully dynamic test was run with an outside temperature profile that ranged from 60°F (16°C) to a high of 105°F (41°C). The results from this scenario did result in the system unloading on both days and causing uncomfortable indoor temperatures, as shown in Figure 101. Despite the unit unloading, the unit response was still very consistent across the two days, with a good match in the daily EER values. The EER numbers for these tests were higher than the ones from the previous test in Figure 100 despite the higher outside temperatures as a direct result of the unloading during the hottest part of the day.

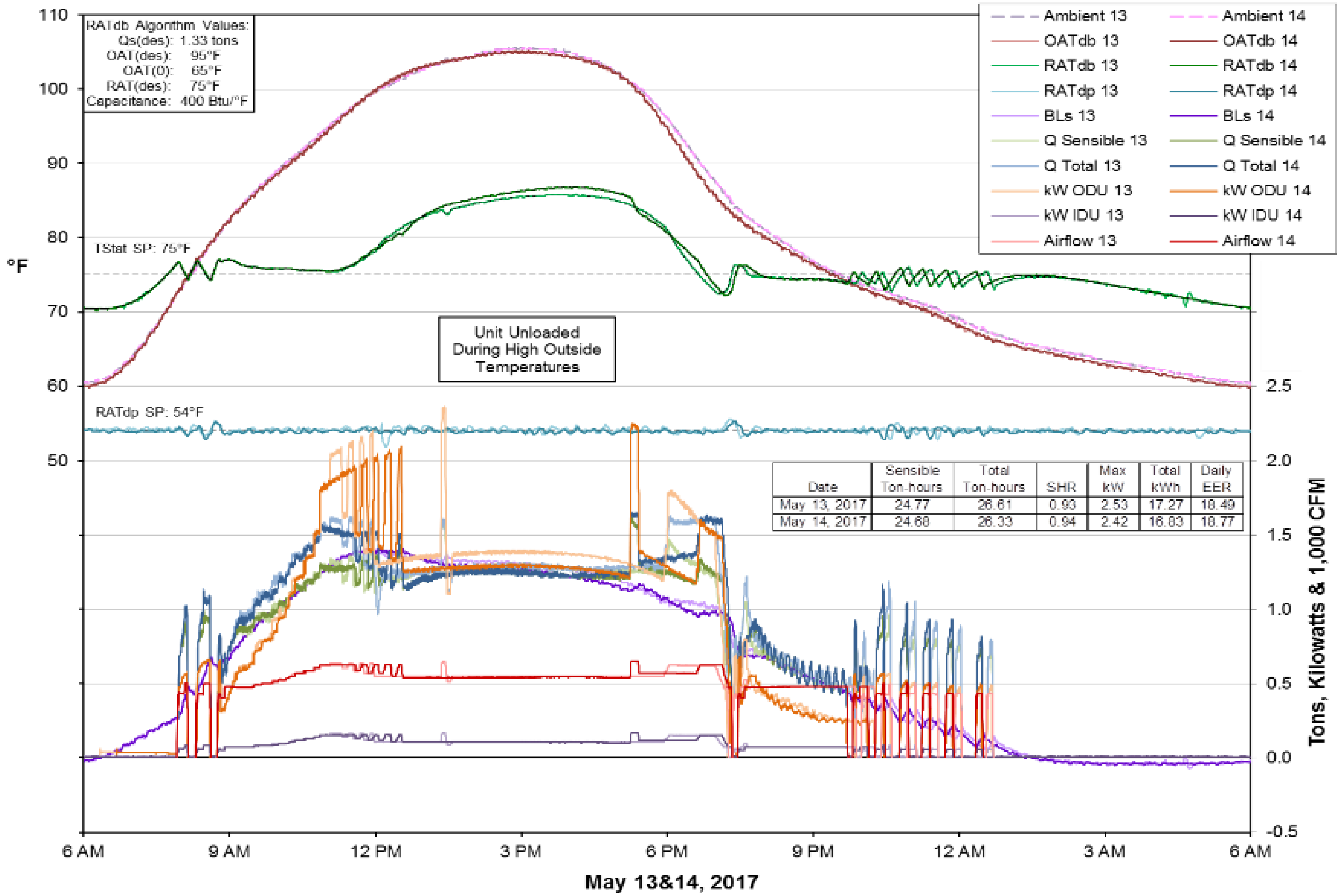
Figure 100: Cooing Dynamic Test #2-2 (Fully-Dynamic, Moderate Temperature)



Date	Sensible Ton-hours	Total Ton-hours	SHR	Max kW	Total kWh	Daily EER
April 29, 2017	17.79	19.69	0.90	2.08	21.31	11.08
April 30, 2017	17.74	19.91	0.89	2.09	21.43	11.15

Source: EPRI

Figure 101: Cooling Dynamic Test #2-3 (Fully Dynamic, High Temperature, R-32)



Source: EPRI

Two additional semi-dynamic tests were run to repeat much of what was seen in Figure 99, and are included in Appendix E. These include Figure E-1 recorded May 5-9 and Figure E-2 recorded May 11-12, 2017. In both cases, the unit continued to unload at temperatures above 100°F (38°C).

Plan for Standard Tests in Heating Mode

As with cooling, the conditions applied for the heating mode tests included those used for rating purposes in AHRI Standard 210/240-2008, Table 33 (for variable-speed compressor systems), plus some additional outdoor conditions to produce a performance map. Once again, the Ram Monitor program was used to put the system into a fixed operating mode, except now using the "Heating_Test_Mode" parameter to operate the unit as a heat pump. The planned performance tests are listed in Table 33, including their AHRI designation. The $H2_2$ test and $H2_1$ tests are optional for a variable speed system as they can be estimated from other test results. ($H2_1$ and Maximum are not used in the calculation of HSPF.)

Table 33: Planned Heating Mode Standard Tests

Test Description	Air Entering Indoor Unit Temperature Dry-Bulb (°F)	Air Entering Outdoor Unit Temperature Dry-Bulb (°F)	Air Entering Outdoor Unit Temperature Wet-Bulb (°F)	Compressor Speed and Heating Air Volume Rate
AHRI $H0_1$ Test	70	62	56.5	Minimum
AHRI $H0C_1$ Test (Cyclic)	70	62	56.5	Minimum
AHRI $H1_2$ Test	70	47	43	Maximum
AHRI $H1_1$ Test	70	47	43	Minimum
AHRI $H2_2$ Test (Optional)	70	35	33	Maximum
AHRI $H2_V$ Test	70	35	33	Intermediate
AHRI $H2_1$ Test (Optional)	70	35	33	Minimum
AHRI $H3_2$ Test	70	17	15	Maximum
AHRI Maximum Conditions	80	75	65	Maximum
Performance Mapping	70	65, 55, 45, 35*, 25, 15		Maximum & Minimum

*** Mapping test condition already included in standard tests**

Source: EPRI

For the performance mapping tests, the decision was made to try to achieve a constant 70 percent relative humidity in the outdoor room for each of the dry bulb temperature steps. This is consistent with the H0, H1, and H3 test conditions, but not the H2 or Maximum tests.

The outdoor space conditioning apparatus was actually able to achieve temperatures into the low 40's through continuous recirculation of the chamber air, and the secondary refrigeration system was not needed. The difficulty with performing any of the heating mode tests at outside temperatures below about 45°F (7°C) is the tendency for ice to form on the evaporator coils. This problem is not just experienced by the sub-chamber refrigeration coil but can also happen on the coil of the outdoor chamber space conditioning system. As ice forms, it cuts off airflow through the coil, and will make it impossible to properly control the space temperature. It becomes necessary then to periodically turn off the cooling systems and defrost their coils. For the outdoor room space conditioning system, this can be done relatively quickly by toggling its reversing valve to put it into heat pump mode just long enough to melt the formed ice. For the sub-chamber refrigeration coil, defrosting required manually toggling the defrost cycle which turned off the compressor and switched on electric resistance heat on the coil. These defrost cycles can be set to run periodically, but these may not happen at opportune times. Defrosting the refrigeration coil can be a long process, and the coil is not cooling the chamber during these times, which leads to unstable temperature conditions.

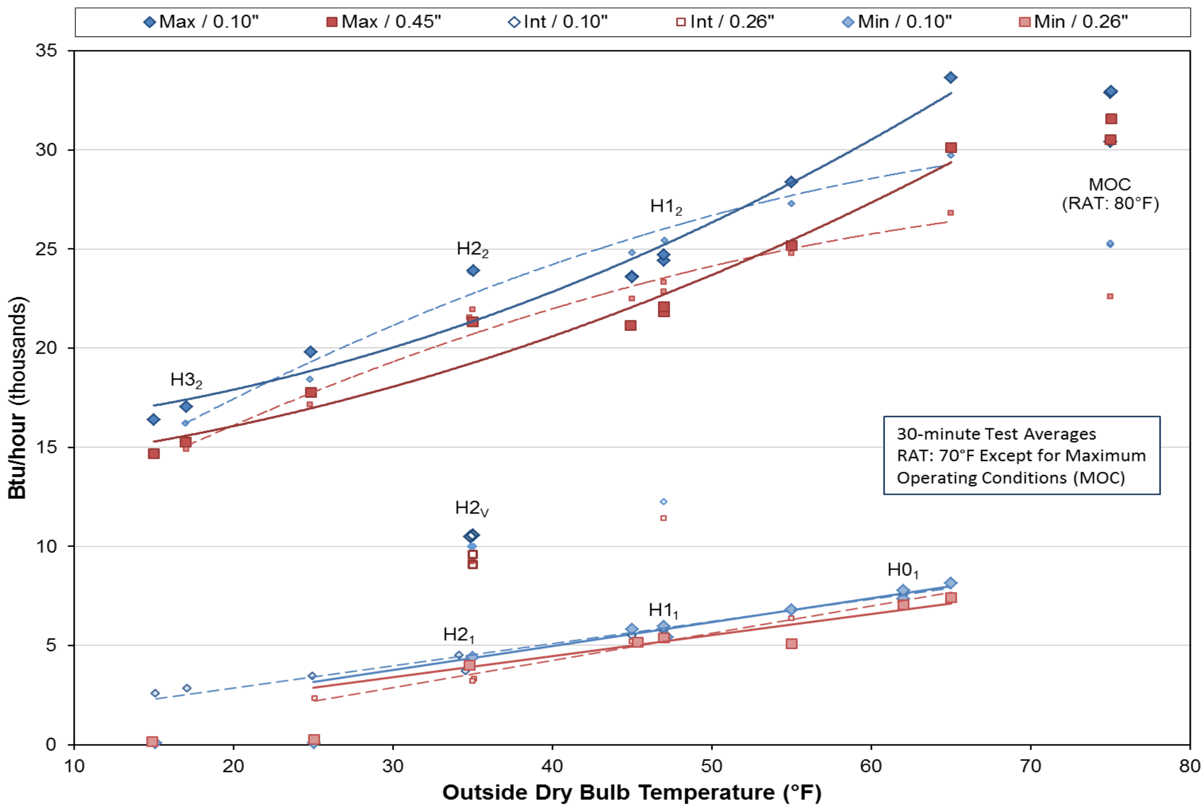
The test unit itself also has a built-in defrost cycle when operating as a heat pump. The frequency of these defrosts can be set in the thermostat to a maximum interval of two hours but could not be disabled. Operating the heat pump via the Ram Monitor program does not override these periodic defrosts, and this leads to extended testing to capture a 30-minute stable period. Compounding this is that the system can take a long time to return to steady-state operation following a defrost cycle, and as much as an hour is needed for recovery when operating at minimum speed.

Heating Mode Standard Test Results

Most of the standard tests were completed between May 10 to May 30, interspersed with some dynamic cooling mode tests on overnights or weekends. The heating mode tests are more "hands on" due to the need for keeping the space conditioning system from frosting over at very low temperatures, and also coordinating tests around the test unit's two hour defrost cycles. Thus, there was not much opportunity to run overnight tests by script to reduce the testing time. There were also some observed capacity drops in the middle of a test with constant environmental conditions and while supposedly in a fixed operating mode. This resulted in some inconsistencies in the test results that do not neatly follow a trend.

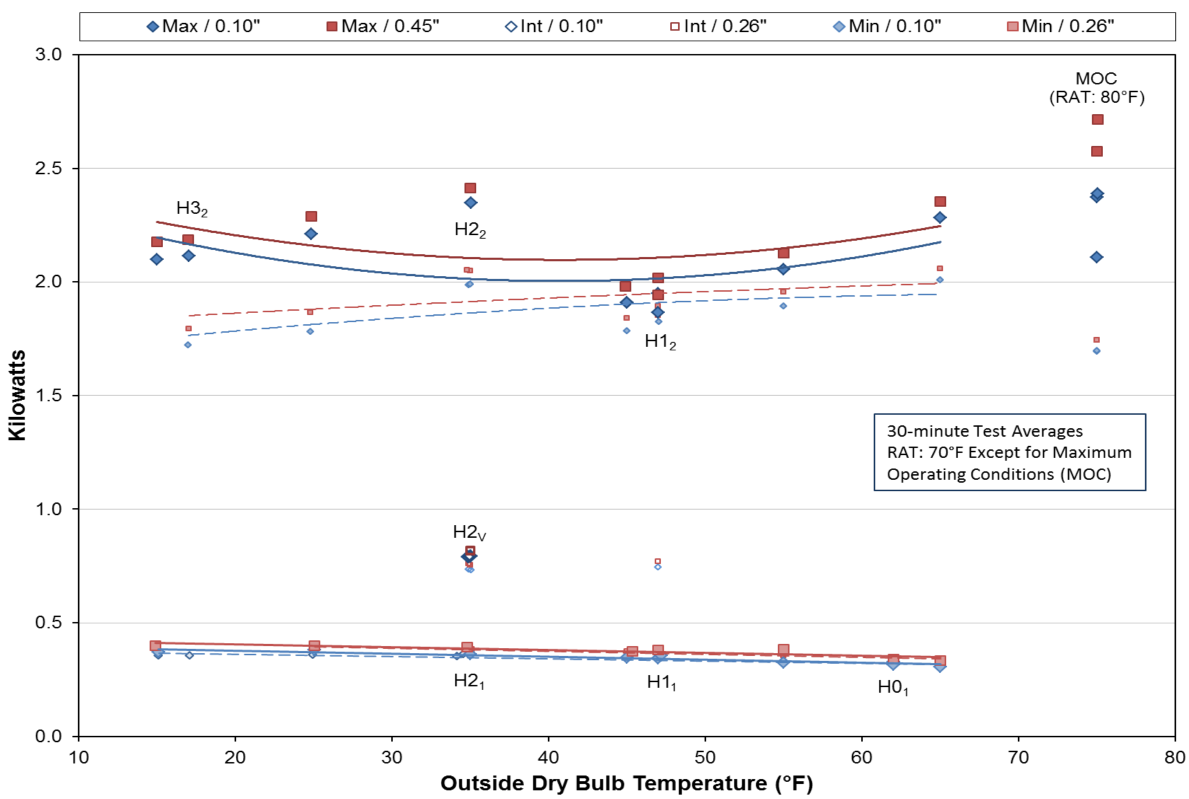
Once again, the results from the heating mode tests are presented in three figures and two tables. Table 34 summarizes the R-32 results using the standard heating mode tests run with R-410A, and Table 35 contains the relative change in the results. Figure 102 presents the trends of heating capacity in thousands of Btu/hr, Figure 103 presents total power in kilowatts, and Figure 104 presents efficiency in terms of the dimensionless COP. In all of the three figures, the results from the earlier tests with R-410A are included in the background with small symbols and dashed lines.

Figure 102: Heating Mode Mapping – Capacity (R-32)



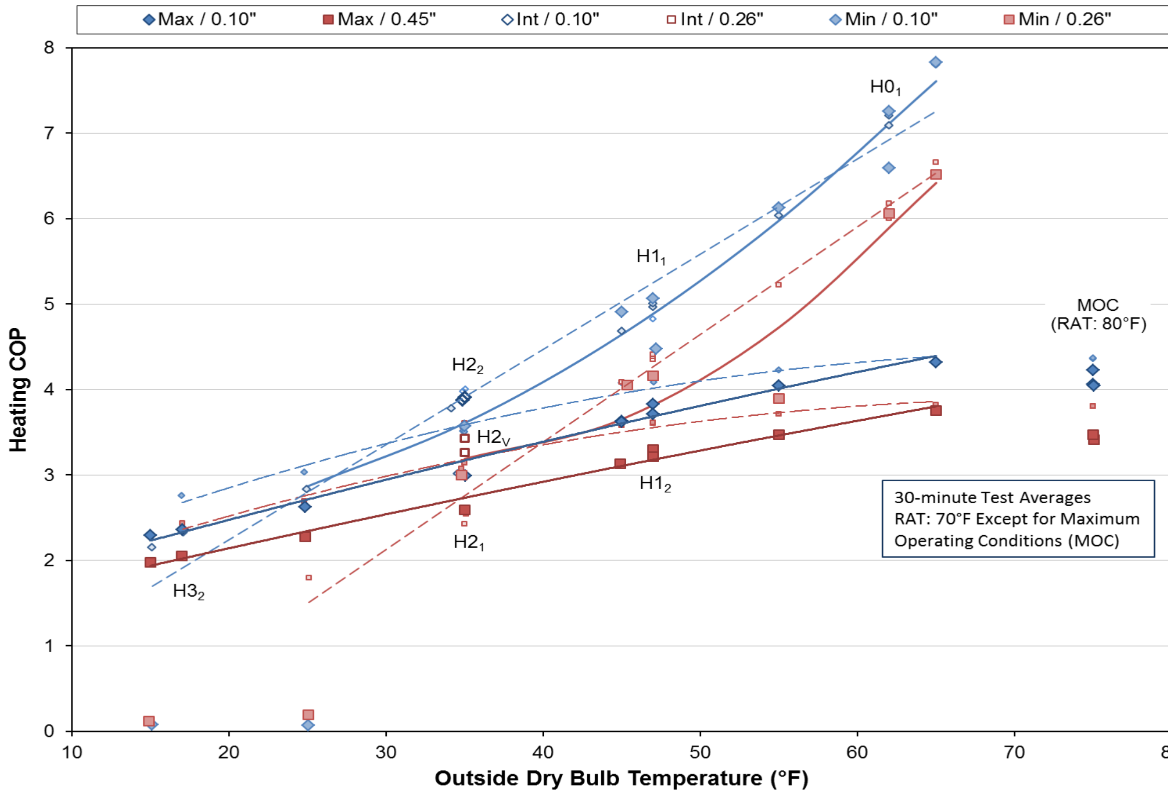
Source: EPRI

Figure 103: Heating Mode Mapping – Total Power (R-32)



Source: EPRI

Figure 104: Heating Mode Mapping – Efficiency (COP, R-32)



Source: EPRI

In all of these figures, there is a clear shift in system operation between the $H1_2$ Standard Rating outdoor condition of 47°F (8°C) and the $H2_2$ condition at 35°F (2°C) when operated at "High" speed. In the Phase 1 testing, this phenomenon was also observed and thought to be due to the higher outdoor humidity level in the outdoor room. The suspicion is that it may be built into the control system to vary its capacity as a function of the outside temperature, and fixing the speed via the "Heating Test Mode" parameter to "High" in Ram Monitor may not actually result in the same compressor speed depending on the temperature. The inverter speed reading in Ram Monitor (INV) is not included in its data log and was not otherwise recorded manually to verify this possibility.

Table 34: Heating Mode Standard Test Results (R-32)

Test Description	Compressor Speed and Cooling Air Volume Rate	Outside Air Tdb (°F)	Outside Air Twb (°F)	Return Air Tdb (°F)	Ext. Res. (IW)	Heating Capacity (kBtu/hr)	Power (kW) IDU	Power (kW) ODU	Power (kW) Total	COP	Indoor Blower Airflow (CFM)	Indoor Blower Speed (RPM)	Outdoor Fan Speed (RPM)
$H0_1$	Minimum	63	57.0	70	0.05	7.56	0.042	0.28	0.32	6.92	557	405	599
					0.23	6.98	0.071	0.27	0.35	5.91	498	566	599
$H0C_1$	Minimum (Cyclic)	*63*	*57.0*	*70*	0.05	1.17			0.109	3.14	Cyclic		
					0.23	1.27			0.119	3.11			
$H1_2$	Maximum	47	44	70	0.10	24.54	0.087	1.82	1.91	3.77	798	550	719
					0.45	21.97	0.164	1.82	1.98	3.25	704	769	719
$H1_1$	Minimum	47	43	70	0.05	5.68	0.042	0.31	0.35	4.77	553	405	599
					0.24	5.39	0.072	0.31	0.38	4.16	492	569	599
$H2_2$	Maximum	35	34	70	0.10	23.91	0.086	2.26	2.35	2.99	796	545	719
					0.45	21.31	0.162	2.25	2.41	2.59	704	766	719
$H2_V$	Intermediate	35	33	70	0.05	10.53	0.041	0.75	0.79	3.89	561	403	599
					0.23	9.60	0.070	0.75	0.82	3.43	503	562	599
$H2_1$	Minimum	35	33	70	0.05	4.39	0.041	0.32	0.36	3.57	553	403	599
					0.23	4.01	0.070	0.32	0.39	3.00	494	562	599
$H3_2$	Maximum	17	16	70	0.10	17.05	0.086	2.03	2.11	2.36	783	546	719
					0.45	15.27	0.163	2.02	2.19	2.05	691	766	719
Max	Maximum	75	65	80	0.10	32.91	0.082	2.30	2.38	4.05	790	545	719
					0.45	31.05	0.165	2.48	2.65	3.44	723	778	719
Performance Mapping	Maximum	65	59	70	0.10	33.62	0.087	2.20	2.28	4.32	814	550	719
					0.45	30.13	0.164	2.19	2.35	3.75	720	770	719
		55	51	70	0.10	28.36	0.087	1.97	2.05	4.05	807	550	719
					0.45	25.17	0.164	1.96	2.13	3.47	712	770	719
		45	42	70	0.10	23.60	0.086	1.82	1.91	3.62	797	548	718
					0.45	21.13	0.163	1.82	1.98	3.13	704	768	719
		35	34	70	0.10	23.91	0.086	2.26	2.35	2.99	796	545	719
					0.45	21.31	0.162	2.25	2.41	2.59	704	766	719
		25	23	70	0.10	19.80	0.087	2.12	2.21	2.63	796	550	719
					0.45	17.75	0.164	2.12	2.29	2.27	703	771	719
		15	14	70	0.10	16.38	0.088	2.01	2.10	2.29	792	551	719
					0.45	14.67	0.165	2.01	2.17	1.98	700	771	719
	Minimum	65	59	70	0.05	8.15	0.042	0.26	0.31	7.83	561	406	599
					0.23	7.42	0.071	0.26	0.33	6.52	502	566	599
		55	51	70	0.05	6.81	0.042	0.28	0.33	6.13	559	406	599
					0.23	5.09	0.071	0.31	0.38	3.89	498	564	599
		45	42	70	0.05	5.80	0.041	0.31	0.35	4.91	554	401	599
					0.23	5.17	0.070	0.30	0.37	4.05	495	562	599
		35	33	70	0.05	4.39	0.041	0.32	0.36	3.57	553	403	599
					0.23	4.01	0.070	0.32	0.39	3.00	494	562	599
		25	24	70	0.05	0.09	0.041	0.34	0.38	0.07	548	404	599
					0.23	0.26	0.070	0.33	0.40	0.19	490	562	599
		15	15	70	0.05	0.09	0.041	0.33	0.37	0.07	550	405	599
					0.23	0.16	0.070	0.33	0.40	0.12	490	563	599

Source: EPRI

Table 35: Relative Change from R-410A to R-32 in Heating Mode

Test Description	Compressor Speed and Cooling Air Volume Rate	Outside Air Tdb (°F)	Outside Air Twb (°F)	Return Air Tdb (°F)	Ext. Res. (IW)	Heating Capacity (kBtu/hr)	Power (kW) IDU	Power (kW) ODU	Power (kW) Total	COP	Indoor Blower Airflow (CFM)	Indoor Blower Speed (RPM)	Outdoor Fan Speed (RPM)
H0$_1$	Minimum	62	56.8	70	0.08	-1%	-16%	+5%	+2%	-3%	+5%	-9%	+0%
					0.26	-3%	-11%	+4%	+0%	-3%	+1%	-5%	+0%
H0C$_1$	Minimum (Cyclic)	*62*	*56.8*	*70*	0.08	-4%			-3%	-1%	Cyclic		
					0.26	+25%			+1%	+24%			
H1$_2$	Maximum	47	43	70	0.10	-3%	+17%	+4%	+5%	-8%	+13%	+7%	+0%
					0.45	-5%	+10%	+5%	+6%	-10%	+7%	+2%	+0%
H1$_1$	Minimum	47	43	70	0.08	-2%	-16%	+6%	+3%	-4%	+5%	-10%	+0%
					0.25	-2%	-6%	+6%	+4%	-5%	-0%	-2%	+0%
H2$_2$	Maximum	35	34	70	0.10	+0%	+14%	+18%	+18%	-15%	+12%	+5%	+0%
					0.45	-2%	+9%	+18%	+18%	-17%	+7%	+2%	+0%
H2$_V$	Intermediate	35	34	70	0.08	+6%	-18%	+10%	+8%	-2%	+5%	-11%	+0%
					0.25	+4%	-9%	+10%	+8%	-4%	+1%	-4%	+0%
H2$_1$	Minimum	35	33	70	0.07	+6%	-18%	+4%	+1%	+5%	+5%	-10%	+0%
					0.24	+23%	-8%	+4%	+2%	+21%	+1%	-4%	+0%
H3$_2$	Maximum	17	16	70	0.10	+5%	+15%	+23%	+23%	-14%	+12%	+5%	+0%
					0.45	+2%	+9%	+23%	+22%	-16%	+6%	+2%	+0%
Max	Maximum	75	65	80	0.10	+33%	+33%	+45%	+44%	-8%	+24%	+12%	+0%
					0.45	+37%	+24%	+54%	+52%	-9%	+21%	+5%	+0%
Performance Mapping	Maximum	65	59	70	0.10	+13%	+15%	+14%	+14%	-1%	+13%	+6%	+0%
					0.45	+12%	+9%	+15%	+14%	-2%	+8%	+2%	+0%
		55	50	70	0.10	+4%	+15%	+8%	+8%	-4%	+12%	+6%	+0%
					0.45	+2%	+9%	+9%	+9%	-7%	+7%	+2%	+0%
		45	42	70	0.10	-5%	+15%	+7%	+7%	-11%	+11%	+5%	-0%
					0.45	-6%	+10%	+7%	+7%	-13%	+7%	+2%	+0%
		35	34	70	0.10	+0%	+14%	+18%	+18%	-15%	+12%	+5%	+0%
					0.45	-2%	+9%	+18%	+18%	-17%	+7%	+2%	+0%
		25	24	70	0.10	+7%	+17%	+24%	+24%	-13%	+13%	+6%	+0%
					0.45	+4%	+10%	+24%	+23%	-16%	+8%	+2%	-0%
		16	15	70	0.10	+1%	+17%	+22%	+22%	-17%	+13%	+6%	+0%
					0.45	-2%	+11%	+22%	+21%	-19%	+8%	+2%	+0%
	Minimum	65	60	70	0.08	-0%	-17%	+2%	-1%	+0%	+5%	-10%	+0%
					0.25	-2%	-8%	+3%	+0%	-2%	+1%	-3%	+0%
		55	50	70	0.08	-0%	-17%	+1%	-2%	+2%	+5%	-10%	+0%
					0.25	-20%	-8%	+12%	+7%	-26%	+1%	-4%	+0%
		45	42	70	0.07	+5%	-18%	+4%	+1%	+5%	+4%	-11%	+0%
					0.25	-1%	-9%	+3%	+0%	-1%	+1%	-4%	+0%
		35	33	70	0.07	+6%	-18%	+4%	+1%	+5%	+5%	-10%	+0%
					0.24	+23%	-8%	+4%	+2%	+21%	+1%	-4%	+0%
		25	24	70	0.08	-97%	-18%	+10%	+6%	-98%	+4%	-10%	-0%
					0.24	-89%	-8%	+8%	+5%	-89%	+1%	-4%	-0%
		20	19	70	0.15	-96%	-46%	+8%	-3%	-96%	+13%	-30%	-0%

Source: EPRI

Calculation of Heating Season Performance Factor (HSPF) for R-32

The methodology for calculating the HSPF for the unit with R-32 was the same as described in the Phase 1 report. In fact, the spreadsheet developed previously for the tests with R-410A was simply updated with the new readings contained in Table 34.

As before, there is a multiplier of two for the two different external resistance values at which the tests were conducted. There is another factor of two in that the optional H2$_2$ test results could be used, or a mathematical approximation of this test can be obtained from a calculation based on the values from the H1$_2$ and H3$_2$ tests. Both these situations were looked at, with the calculated approximation indicated by an asterisk (*) in the results. The main reason for the proliferation of HSPF calculations is that the standards call for it to be calculated for six "Regions" with different temperature bin weighting factors; and in each region, two building load lines referenced to region-specific outdoor design temperatures. Thus, instead of two SEER results, there are forty-eight HSPF results. Reporting of HSPF is normally just done for Region IV at a single external resistance, so there would be just the range between the minimum and maximum load lines provided. Figure 37 describes graphically how four values of HSPF are derived for one particular region (Region IV) using the very complicated HSPF calculation method. Another wrinkle in this calculation from what is done for SEER is that the slope of the capacity and power at maximum speed within the range of 17°F (–8°C) to 45°F (7°C) is to be derived from the $H2_2$ (or $H2_2$*) and $H3_2$ tests, and outside this range the slopes are derived from the $H1_2$ and $H3_2$ tests. (The trends determined using the calculated $H2_2$* test values are shown as thin dotted lines.) The minimum building load line is based on the capacity of the system from the $H1_2$ test rounded off to a "standard" heating value and referenced to a region-specific design outdoor temperature (5°F [–15°C] for Region IV), and Blmax is about twice the slope of Blmin, also subject to rounding.

Once again, the intersections between the load lines and the available capacity trends for the minimum and maximum speeds, and the pseudo-trend for the intermediate speed are used to determine a second order curve fit for the power demand at intermediate speeds. (Since there are two alternative slopes for the maximum speed capacity and power trends, there are also two alternative slopes for the pseudo-intermediate speed capacity and power trends, and two intermediate speed power curves.)

In the SEER calculation when the building load exceeded the available cooling at maximum speed, the capacity and power followed just what the system could provide. For HSPF, when the heating load exceeds the available heat pump capacity at maximum speed, electric resistance heat is assumed to pick up the difference[5]. The regions where the system would be using some form of backup heat are shaded in the upper left corner of the figure.

Four values of HSPF are calculated based using the two load lines and the two values of $H2_2$ for one Region. Figure 37 presents the results from all 48 combinations of variables. Note that all of the results for each of the two external resistance levels are derived from the same set of measurments, but just with different weighting in the analysis.

Table 36 presents the numerical values of the results shown in Figure 38, along with the relative change from the HSPF values calculated from Phase 1. The change was usually very small, ranging from 8 percent less to 7 percent more, except for one outlier in the Region 5 calculation which was 20 percent higher.

[5] Although the test unit uses a natural gas furnace for its backup heat source, the HSPF calculation still assumes electric resistance heat since it is applied to an outdoor unit that could be combined with either. The furnace combination is also an either/or function where the system will operate as a heat pump or using the furnace, but not together, while the electric resistance option is providing supplemental heat with both systems running.

Table 36: Multiple HSPF Values Derived from Same Two Test Data Sets (R-32)
(Results in Btu/h/W and the Relative Change from the R-410A Results)

External Resistance at Maximum	Measured or Calculated $H2_2$	Region I		Region II		Region III		Region IV		Region V		Region VI	
		Min	Max	Min	Max	Min	Max	Min	Max	Min	Max	Min	Max
0.10	Measured	15.57	15.40	14.64	13.50	13.39	11.16	11.04	8.31	9.82	7.05	15.42	14.60
		-3%	-4%	-3%	-5%	-4%	-5%	-4%	-5%	-3%	+5%	-3%	-4%
	Calculated	15.61	14.42	14.68	12.55	13.42	10.15	11.07	7.67	8.95	6.83	15.43	13.60
		-3%	-4%	-3%	-4%	-3%	-5%	-4%	-4%	-3%	+7%	-3%	-4%
0.45	Measured	13.68	13.74	12.84	11.41	11.66	10.06	10.88	8.02	9.95	6.25	13.52	13.03
		-4%	-5%	-5%	-8%	-5%	-8%	+7%	+3%	+20%	-6%	-4%	-6%
	Calculated	13.70	13.19	12.85	10.39	11.66	9.26	9.99	7.56	8.13	6.02	13.51	12.37
		-4%	-5%	-5%	-7%	-5%	-8%	-2%	+6%	-2%	-6%	-4%	-6%

Source: EPRI

Heating Mode Dynamic Test Results

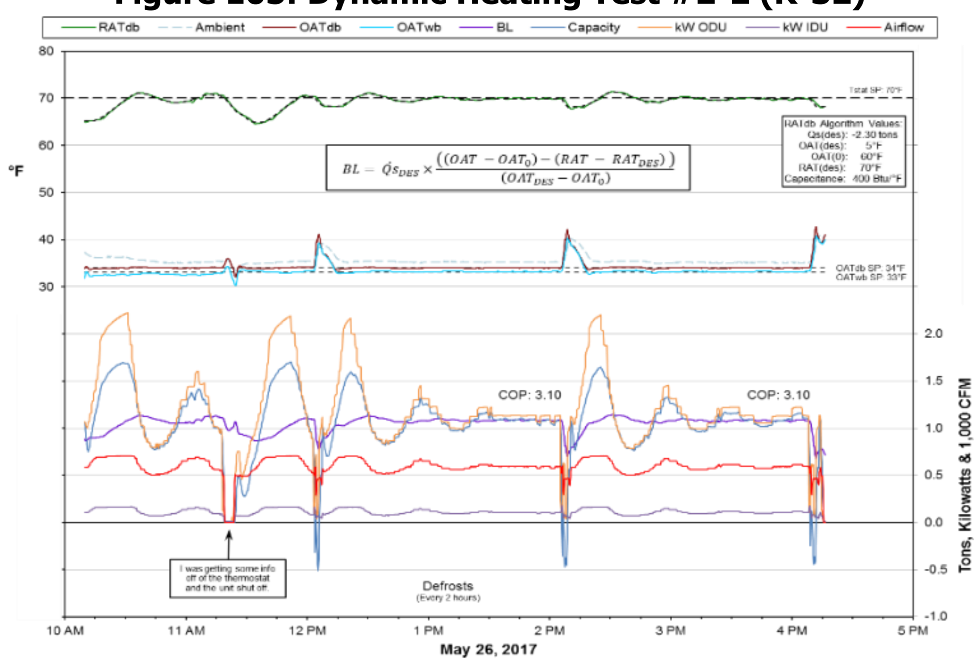
The heating dynamic tests used the same algorithmic procedure developed for the cooling mode tests, although not as many tests were attempted in heating mode due to the additional attention required to keep the space conditioning systems from freezing. What was done was mainly at higher temperatures to delay system freezing as long as possible. Figure E-3 through Figure E-5 in Appendix E were captured from a single continuous test, but have been broken up into segments based on a fixed outside temperature. The first segment (#2-1A) was at the highest outside temperature setting of 54°Fdb. This was first run in combination with the CSA Draft Standard “marine” wet bulb temperature of 49°F (9ºC) for the first six hours, and then reset to the “dry” 45°F (7ºC) for the next six hours, although the system wasn’t able to achieve this due to local climate conditions (too humid outside). The response of the system was mostly short cycles at a nearly consistent frequency, starting with an initial surge in capacity and power followed by a leveling off. While the system had been set up with a minimum defrost cycle setting of two hours, these occurred more erratically and more frequently with time. It seems odd that the system would be running defrosts at all with the relatively warm outdoor temperatures. The system cycling and the defrost cycles created some instability in the recorded temperature in the outdoor room sub-chamber, but the indoor temperature did not vary appreciably, even with the dynamic algorithm controlling the temperature and the thermostat controlling system operation.

For the second segment (#2-1B), the outside room temperature was reduced to 47°Fdb and 45°Fwb for the first six hours, and then the wet bulb set point was reduced to 41°F (5ºC) for the next six hours. Towards the end of the second segment, the outside temperature was starting to gradually rise as the test chamber space conditioning system could no longer keep up with the defrost heat input. The response of the unit to these outdoor temperatures changed from on/off cycling to hunting for an appropriate speed setting. There was one off cycle recorded at about 3:25 AM for no apparent reason, and no faults were recorded. The off cycle at about 8:30 AM exposed a problem in the live calculation of capacity used in the room temperature control algorithm when the supply air dew point instrument ran its periodic self-cleaning cycle and created a false capacity spike (shown as a dashed line). This caused a sharp rise in the room temperature and resulted in the unit cycling off when the thermostat was satisfied.

For the last segment (#2-1C), the sub-chamber was isolated and its chiller coil activated to attempt the target temperature of 33°F (1ºC), although the apparatus was not able to hold this steady. The thermostat had not been adjusted from the previous settings for backup heat, which had settings with the gas heat locked out above 40°F (4ºC) and the compressor locked out below 35°F (2ºC). Thus, shortly after reaching the 33°F (1ºC) set point, the system switched over to gas heating. The first cycle of the gas heat also triggered a B3: BLOWER MOTOR LIMIT error to the alerts log, probably due to the higher airflow rate when the furnace was activated. Later cycles did not have as high of a flow rate and no other faults were recorded. The gas heating capacity is so much greater than the heat pump capacity that the indoor room temperature (green line) had a bit more difficulty following the algorithm set point (underlying dashed line) with the heat pump. After defrosting the chiller coil in the middle of the test and allowing the sub-chamber temperature to rise for a bit, the system returned to heat pump operation for almost 1-1/2 hours (including a defrost cycle) before returning to gas heating.

In Figure 105, the control logic for the outside room sub-chamber temperature was modified such that much of the oscillation that occurred in the previous heating mode tests was dampened out. This test was nearly a repeat of the previous test with an outside temperature set point of 34°F (1ºC), but this time the lockouts were adjusted down 5°F (2.8ºC) each so that the system would stay in heat pump mode as long as it could maintain the temperature. The test lab was actually able to hold the desired temperature relatively constant, except for the times when the test unit went into defrost and the outdoor coil started to heat up the test chamber instead of helping to keep it cold. The unit fell into a repeatable pattern between defrosts where it would first overcompensate for the lost heating coming out of the defrost cycle before settling out at a constant output that matched the calculated dynamic building load. The unit was also fully capable of holding the thermostat set point at these temperatures without the use of the gas backup heat.

Figure 105: Dynamic Heating Test #2-2 (R-32)



Source: EPRI

Only one more dynamic test was conducted, and was similar to a full dynamic test. As shown in Figure E-6, it was a test in which the unit was run overnight with the outdoor room supplied with cooled ventilation air from outside the building (economizer open). The outdoor room temperature never dropped below 42°F (6°C), and the heat pump easily met the calculated building load and usually leveled off between defrost cycles. Due to the variability of these dynamic tests and the changes made to the test method, there is no real direct comparison to the previous tests in Phase 1, other than providing insight into how the system will operate in an actual installation.

To put into context, the project results thus far, a summary of Phase 1 laboratory evaluation results are included from Phase 1 interim report, followed by a summary of Phase 2 laboratory evaluation results.

Phase 2 WCEC Lab Activities and Results

The experiments for this project were conducted in two phases – Phase 1 and Phase 2. The objective of Phase 1 was to study the effect of evaporator airflow and compressor speed on the overall system efficiency of a single-zone residential air-conditioning system that includes equipment and ductwork running through an attic. Phase 1 results, described in Chapter 2, showed that for no-zoning operation, the capacity and airflow percentage value to operate the equipment progressively increases as the outdoor (duct-zone) becomes warmer (Beaini et al., 2017). Phase 2 testing, described below, assessed the impact of multi-zone capabilities combined with variable speed controls.

Laboratory Setup

The experimental setup used to test the Next-Generation Residential Space Conditioning System in a multi-zone configuration at the UC Davis Western Cooling Efficiency Center Laboratory in Davis, California, including the psychrometric rooms, assembly of the apparatus and instrumentation plan is described in Chapter 2.

To ensure consistency between the results of Phase 1 and Phase 2, dampers were insulated with R-6 insulation, thus restricting the heat loss through them to negligible amounts. Further, the dampers were controlled electronically (through on/off switches located at the entrance of the test chamber) to either fully open or fully shut positions. In other words, none of the zones could be maintained at partially open positions. All duct sections were arranged on shelves to prevent direct thermal contact between ducts, and the ducts were sealed using the Aeroseal® injection system in order to make them airtight.

The exhaust air from the condenser unit was ducted out of the outdoor chamber to prevent recirculation of condenser exhaust into the outdoor chamber. Air flow was maintained through the chamber to minimize the impact of the duct losses on the temperature of the chamber. The effect on duct heat transfer due to this air flow should be minimal since plastic flex ducts with relatively high insulation resistance. In addition, the increased convective impact due to the chamber air velocity should result in a total heat transfer coefficient that is comparable to the combined effect of free convection and radiation from the roof of a real attic.

The instrumentation used is similar to that used in Phase 1 and includes chamber-condition measurements, evaporator measurements, grille measurements and refrigerant measurements as described in Table 18.

Laboratory Tests

A total of 68 steady-state tests were conducted for Phase 2. The average outdoor chamber temperature for these tests were 85°F (29°C), 95°F (35°C) and 115°F (46°C) DB with 17tests at each temperature. The indoor chamber for all the tests was maintained at 75°F (24°C) DB/62.5°F (16.7°C) WB. For the seventeen tests at each outdoor chamber temperature, (a) four were performed at 40 percent capacity and 40 percent airflow rate with only one zone open, (b) three at the same capacity and airflow rates with two zones open, (c) two at 60 percent capacity and air flow rate with two zones open and (d) one at 60 percent capacity and air flow rate with three zones open and e) one at 80 percent capacity and air flow rate with three zones open. For some of these conditions, additional testing was performed by increasing the air flow rate at the same capacity and same number of zones open, thus varying the sensible heat ratio. The minimum number of zones to be used at each capacity tested was determined based on the upper limit of the plenum pressure drop that could be created by the indoor blower to achieve the desired air flow rate. This pressure drop (also referred to as the external static pressure) was calculated by first determining the resistances of the individual supply duct zones and then using the relationship that the duct resistance is the ratio between the square root of the supply plenum pressure and the duct's air flow rate. Clearly, the pressure drop across the fan will be higher when the number of open zones is reduced because the resistance to flow will also be higher. At no-zoning conditions, Zone 1 carried roughly 32 percent of the flow, Zone 2 and 3 carried 21 percent each, and Zone 4 carried 26 percent of the air flow. Hence, the pressure drop across the indoor unit was largest for Zones 2 and 3, followed by Zone 4, and finally Zone 1.

In order to maintain the external static pressure to the values determined using the methodology discussed above and thus simulate the conditions of a fixed ducting network with multiple zones, as the blower speed was reduced to obtain lower airflow rates, an external circulator fan on the indoor chamber conditioning loop was adjusted to maintain the appropriate pressure vs. flow relationship for all tests. Data was recorded at 30 second intervals using LabView as described in Chapter 2. All tests were conducted for a minimum of half hour, with final results calculated based upon the last 30 minutes of operation. Steady state conditions were ensured by adhering to the temperature tolerances set forth in AHRI 210/240 (AHRI, 20018).

Laboratory Results

Table D-1 and Table D-2 (in Appendix D) provide a summary of steady-state tests conducted during Phase 1. These tables present the chamber temperatures during testing, the indoor airflow rate and capacity, the power consumed by the units, and the external static pressure. The calculated parameters presented are the total delivered capacity, the equipment efficiency, delivery effectiveness of the ducts and the overall system efficiency.

Specific enthalpies of the room, grille, return plenum and supply plenum were calculated based on the temperature measurements described in Figure 44. Hence, $h_{room} = f(T_{db,room}, T_{dp,room})$; $h_{rp} = f(T_{db,rp}, T_{dp,room})$; $h_{grille} = f(T_{db,grille}, T_{dp,sp})$; $h_{sp} = f(T_{db,sp}, T_{dp,sp})$. Note that the supply plenum dry bulb temperature is only based on the temperature of the open zones. The total delivered cooling capacity was calculated based on the enthalpy difference between the indoor chamber and the air supplied at the grilles (Equation 1).

$$\dot{Q}_{delivered,cool} = (\sum \dot{m}_{grille})(h_{room} - \overline{h_{grilles}})$$ **Equation 31**

Here, $\overline{\boldsymbol{h_{grilles}}}$ represents a flow-weighted [1-2][09][0-9][0-9]average of the specific enthalpy of all eight grilles,

$$\overline{h_{grilles}} = \frac{\sum(\dot{m}_{grille} h_{grille})}{\sum \dot{m}_{grille}}$$ **Equation 32**

The cooling capacity of the evaporator was based on the enthalpy difference between the return plenum and the supply plenum multiplied by the mass flow rate of air at the evaporator coil.

$$\dot{Q}_{evap,cool} = \dot{m}_{evap}(h_{rp} - h_{sp})$$ **Equation 33**

Note that $\boldsymbol{\dot{m}_{evap}} = \sum \boldsymbol{\dot{m}_{grille}}$ since the duct system was essentially airtight

The equipment efficiency is based on the difference in enthalpies between the return and supply plenums of the equipment, includes the power input to both the condenser and air handler (indoor fan). This efficiency does not account for losses in the return or supply ducts but does include the impact of indoor-fan heat (See Equation 34). The equipment efficiency is generally what is provided in manufacturer performance tables and AHRI test results.

$$COP_{equip} = \frac{\dot{Q}_{evap,cool}}{P_{equip}} = \frac{\dot{m}_{evap}(h_{rp} - h_{sp})}{P_{indoor} + P_{outdoor}}$$ **Equation 34**

The delivery effectiveness at a single grille and that of the whole duct system were defined using Equations 35 and 36. These equations capture the effects of both sensible heat gain of the air in the ducts as well as the changes in latent heat removal provided to the space. *Note that when there is no air leakage throughout the entire air-conditioning system, the latent heat transfer between ducts and the duct-zone (attic) can be ignored and the delivery effectiveness will only be affected by the sensible heat gain of the air in the ducts. Further, it should be noted that only the grilles which are part of open zones will be considered while calculating delivery effectiveness of the system.*

$$\Xi_{del,grille} = \frac{\dot{Q}_{delivered,grille}}{\dot{Q}_{evap,cool}} = \frac{h_{room} - h_{grille}}{h_{rp} - h_{sp}}$$ **Equation 35**

$$\xi_{del,sys} = \frac{\dot{Q}_{delivered,cool}}{\dot{Q}_{evap,cool}} = \frac{h_{room} - \overline{h_{grilles}}}{h_{rp} - h_{sp}}$$ **Equation 36**

Finally, the efficiency of the whole system was calculated simply as the ratio between the delivered cooling capacity to the conditioned space, and the power consumed by the equipment (see Equation 37). This system efficiency can also be written as the product of the equipment efficiency and the delivery effectiveness (see Equation 38).

$$COP_{sys} = \frac{\dot{Q}_{delivered,cool}}{P_{equip}} = \frac{(\sum \dot{m}_{grille})(h_{room} - \overline{h_{grilles}})}{P_{indoor} + P_{outdoor}}$$ **Equation 37**

$$COP_{sys} = COP_{equip} \times \xi_{del}$$ **Equation 38**

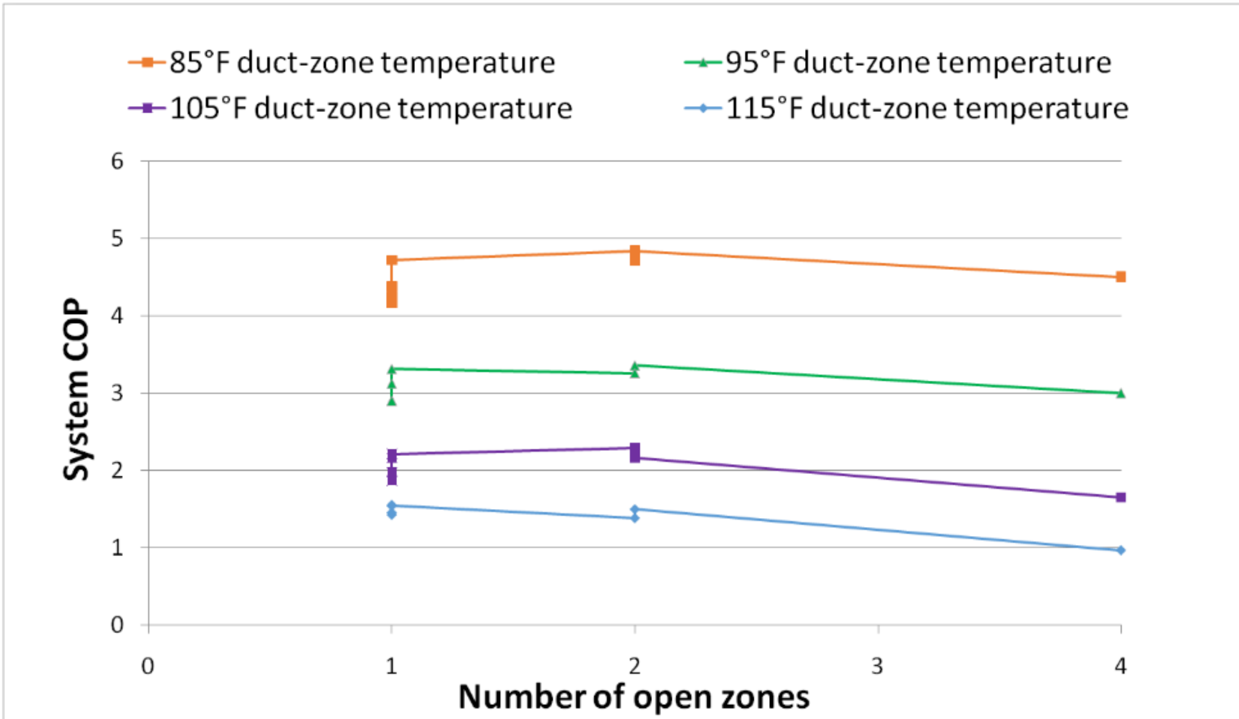
Performance at Synced Capacities and Airflows

It was seen in Phase 1 that reducing the capacity and air flow rate of a variable-capacity/variable-fan-speed heat pump would also reduce the delivery effectiveness of the duct system and therefore have a non-trivial negative impact on the efficiency benefits provided by the air-conditioning equipment. It was also reported based on Phase 1 testing that the reduction in delivery effectiveness is more significant at hotter duct-zone temperatures, resulting in reduction of System COPs at any compressor/fan speeds lower than 100 percent when the ducts are located in a 115°F (46°C) hot attic. The following Sections discusses that zoning can improve delivery effectiveness; followed by a Section that discusses the number of zones that need to be employed for maximum System COP. The subsequent Section discusses the effect of zoning at higher capacities, followed by a Section that discusses the effect of zoning at very low (25 percent) capacities.

Improvement in System COP at Low and Medium Capacities

Figure 106 shows the variation of the System COP plotted against the number of open zones, when the capacity and air flow rate were kept in sync at 40 percent of the nominal values. (Multiple data points represent different zoning configurations.) With the equipment capacity held constant for a given duct-zone temperature, the System COPs are generally higher for less number of open zones than when all four zones are kept open. At an outdoor dry bulb temperature of 85°F (29°C), the System COP at 40 percent capacity and 40 percent air flow rate was 4.50 with all four zones open. The System COP increased to 4.84 when all the capacity was distributed between only two zones and to 4.67 when only one zone was open. At hotter duct-zone temperatures of 95°F (35°C) and 105°F (41°C), not only are the System COPs higher with a zoned system, but the percentage increase is also greater. The System COP increases from 3.00 to 3.35 and from 1.65 to 2.29 for duct-zone temperatures of 95°F (35°C) and 105°F (41°C) respectively when two zones are open instead of all four. At the hottest tested duct-zone temperature of 115°F (46°C), running the compressor and blower at 40 percent of nominal values through only a single zone resulted in a significant increase in System COP from 0.97 to 1.54 (58 percent improvement). These results clearly indicate that zoning can address the performance issues demonstrated in Phase 1 of this project showing a reduction in System COPs (caused by duct heat gain in a hot attic) when operating the compressor and fan at low speeds (Chapter 2).

Figure 106: System COP at 40 Percent Synced Capacity and Air Flow Rate

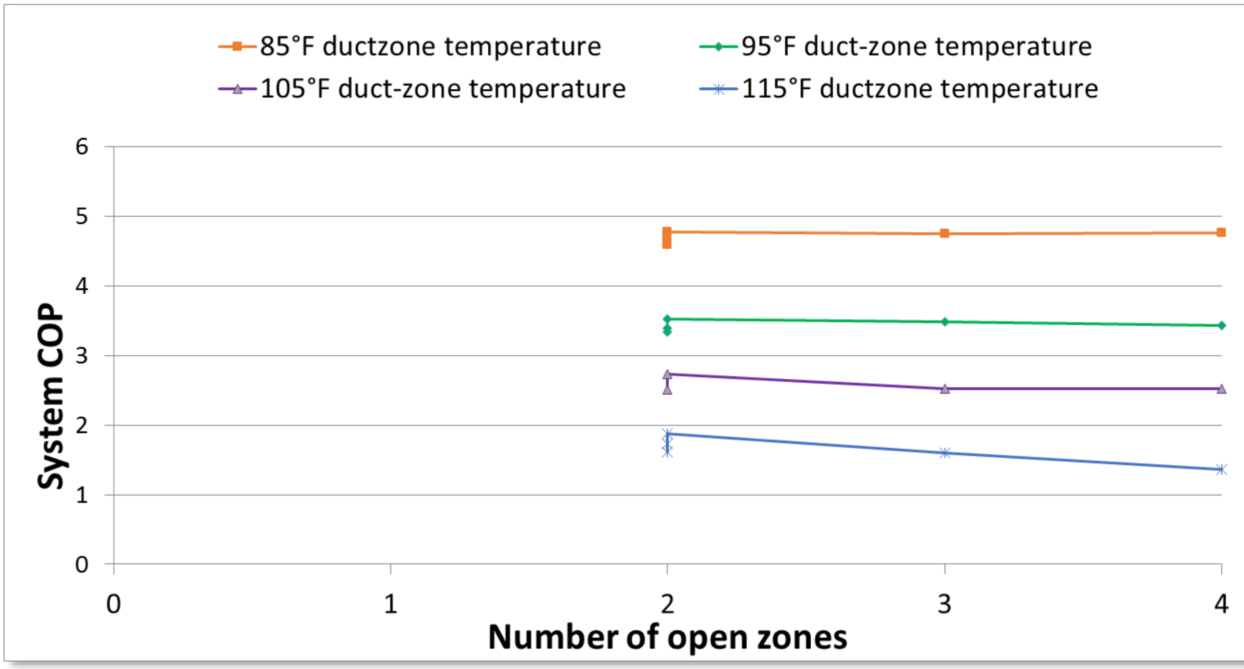


Source: EPRI

It should be noted that based on Phase 1 testing, the highest System COP value without zoning for any combination of capacity and air-flow rate was 1.62 at the most extreme duct-zone temperature tested (115°F [46°C]). This COP, which was reported when the compressor and blower were operating at 100 percent of their nominal operating speeds, is only marginally higher than the COPs obtained at the same temperature conditions at 40 percent capacity and airflow rate when zoned properly. This means that zonal control can be used as a solution in situations where the electric utility calls for a demand-response event to the equipment in a hot-dry climate, by operating the equipment at lower power with minimal impact on system efficiency.

Finally, Figure 107 shows the variations in System COP plotted against the number of zones used for the case of 60 percent capacity and air flow rate. This figure shows that zoning is also effective when operating at 60 percent capacity, although the percentage improvements in COP are slightly lower than the 40 percent case.

Figure 107: System COP at 60 Percent Synced Capacity and Air Flow Rate

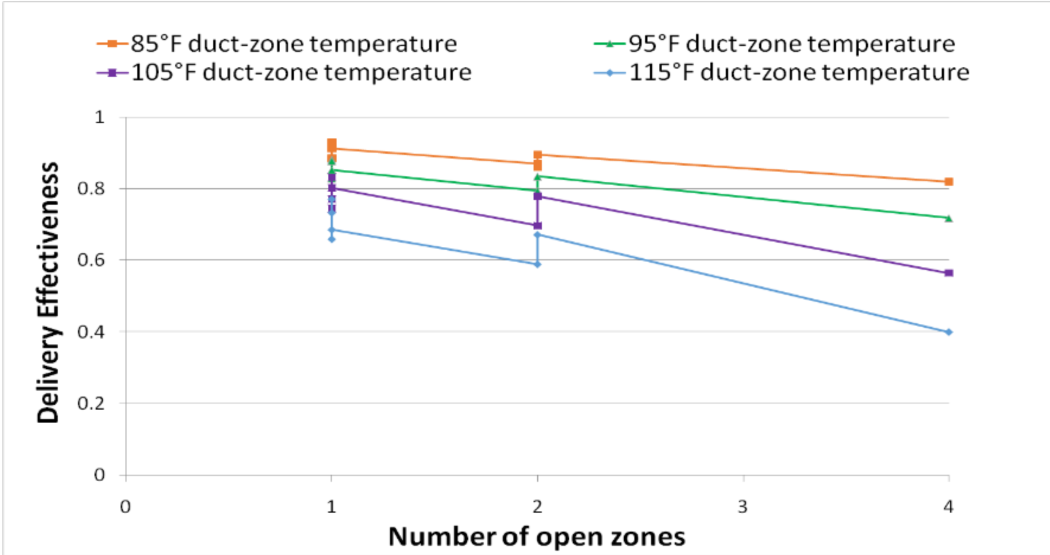


Source: EPRI

Understanding the Optimal Number of Zones

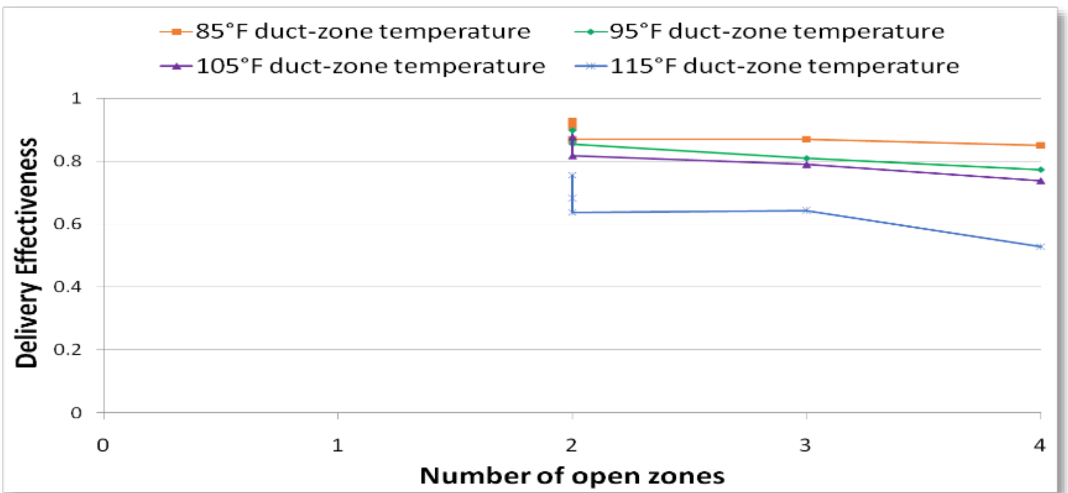
e 106 and e 107 demonstrate that the performance of variable speed equipment at part load can be improved by adding zoning controls. For the case of 60 percent capacity and air-flow rates with a duct-zone temperature of 85°F (29ºC) and 95°F (35ºC), the use of three zones (or two less resistant zones) is optimal while the use of two zones always yields a higher System COP at the hotter duct-zone temperatures of 105°F (41ºC) and 115°F (46ºC). Similarly, at 40 percent capacity and air-flow rate, the use of two zones is more efficient with an 85°F (29ºC) duct-zone temperature while the use of a single zone may be more efficient as the outdoor chamber temperature gets warmer. This can be understood as follows: The delivery effectiveness of the duct system increases as the number of zones delivering the same capacity is reduced (Figure 108, Figure 109, and Figure 110), however the power consumed by the blower increases as the number of active zones is reduced (Table F- in Appendix F) due to a higher static pressure in the supply plenum for the same air flow rate. The increase in fan pressure occurs as a result of the duct resistance added by closing some of the zones. The System COP for zoning depends on the relative changes in the blower power and delivery effectiveness for the given capacity, air flow rate, indoor and duct-zone temperatures. As shown in Figure 108, the delivery effectiveness increases from 0.82 (four zones) to 0.93 (one zone) when only one zone is employed, as opposed to four for duct-zone temperature conditions of 85°F (29ºC) and the capacity and air-flow rates synced with each other at 40 percent; however, this increase is negated by a corresponding increase in total system power of nearly 10 percent due to 50W of additional blower power. By comparison, for similar tests performed at duct-zone temperatures of 95°F (35ºC), 105°F (41ºC) and 115°F (46ºC), zoning improves the delivery effectiveness by much greater amounts (16, 29, and 37 percent respectively) while the blower power increase remains constant at 10% resulting in a higher System COP when operating only a single zone.

Figure 108: Delivery Effectiveness at 40 Percent Synced Capacity and Air Flow Rate



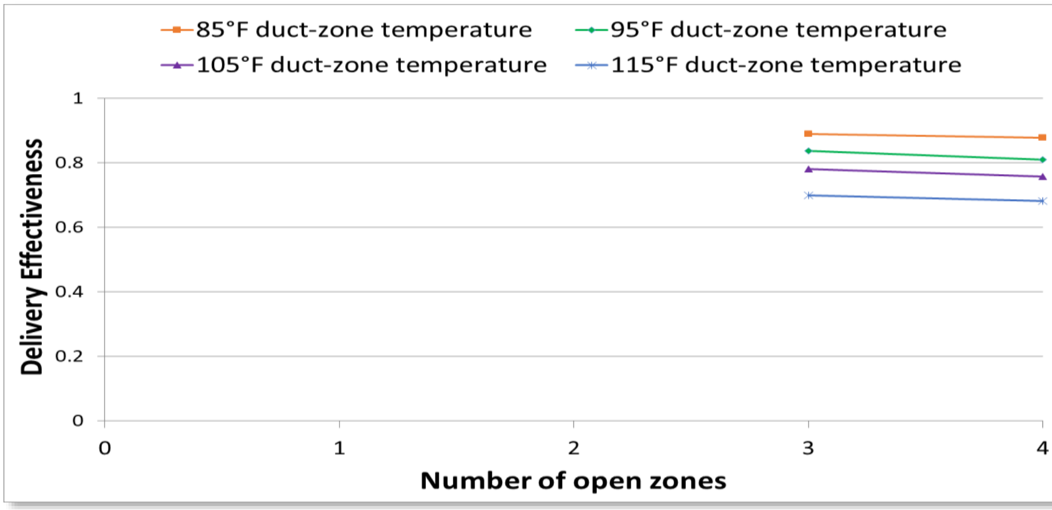
Source: EPRI

Figure 109: Delivery Effectiveness at 60 Percent Synced Capacity and Air Flow Rate



Source: EPRI

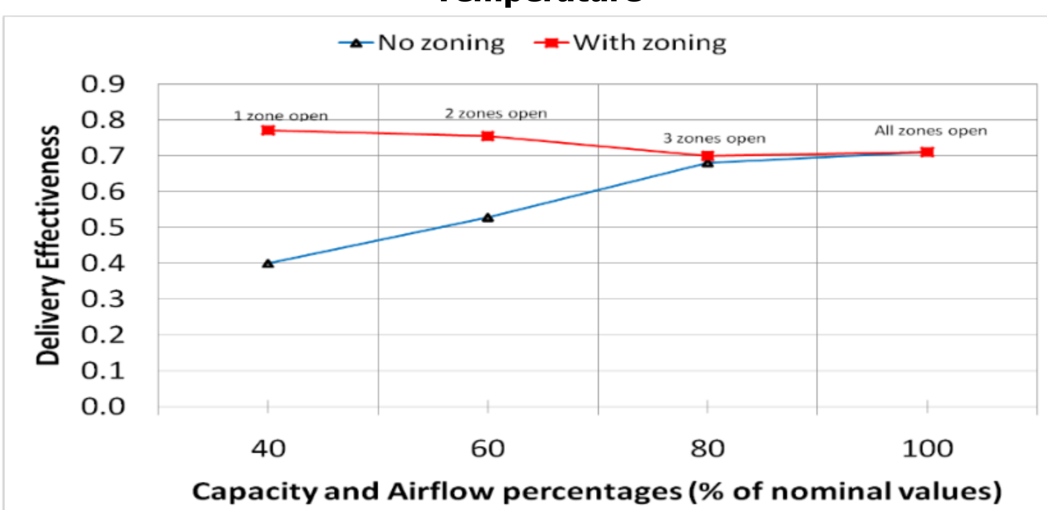
Figure 110: Delivery Effectiveness at 80 Percent Synced Capacity and Air Flow Rate



Source: EPRI

Figure 111 supports the statements made above regarding the increase in delivery effectiveness during zoning operation. This graph shows the variation of delivery effectiveness against capacity and air flow rate (maintained in sync with each other) at 115°F (46°C). Contrary to the no-zoning results from Phase 1, the delivery effectiveness of the duct system during zoning increases as the compressor and fan speeds are reduced. Note that for the case of zoning, the delivery effectiveness values shown here are for the most efficient combination of zones among those tested. However, any combination of zones would yield similar delivery effectiveness values as long as the number of zones employed is fixed, since each of the four zones roughly carries a quarter of the delivered capacity at full speed operation.

Figure 111: Delivery Effectiveness for Synced Operation for 115F Duct-Zone Temperature



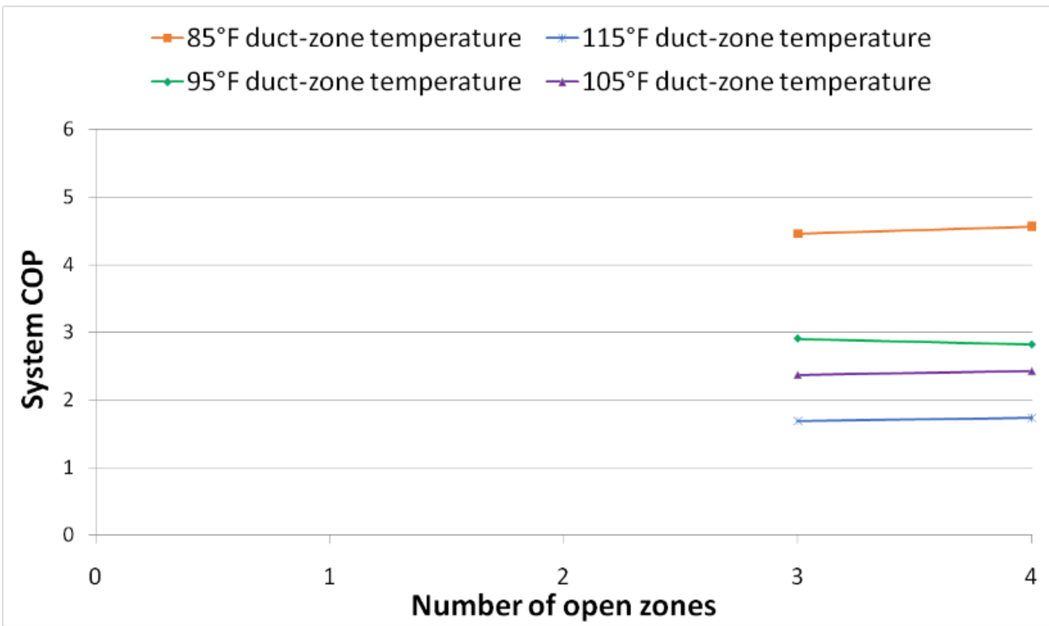
Source: EPRI

Effect of Zoning at Higher Capacities

Additionally, the equipment was tested for zoning performance by increasing the capacity and air flow rate to 80 percent of their nominal values. Zoning was tested at this condition by closing only one zone, since closing more zones would create too much resistance for the blower fan.

Figure 112 shows that at 80 percent capacity and air flow rate, the System COP does not appear to be improved by zoning, and at low attic-zone temperatures the system performance is better with all zones open. This trend, when combined with the observation that the efficiency benefit from zoning is lower at 60 percent capacity than at 40 percent capacity, suggests that zoning becomes less effective at higher capacities when the fan speed and compressor speed are synced. This would also imply that implementation of zonal control is more appropriate for homes using variable-speed air-conditioning equipment rather than with constant-speed equipment.

Figure 112: System COP vs. Number of Open Zones at 80 Percent Capacity and Air Flow Rate

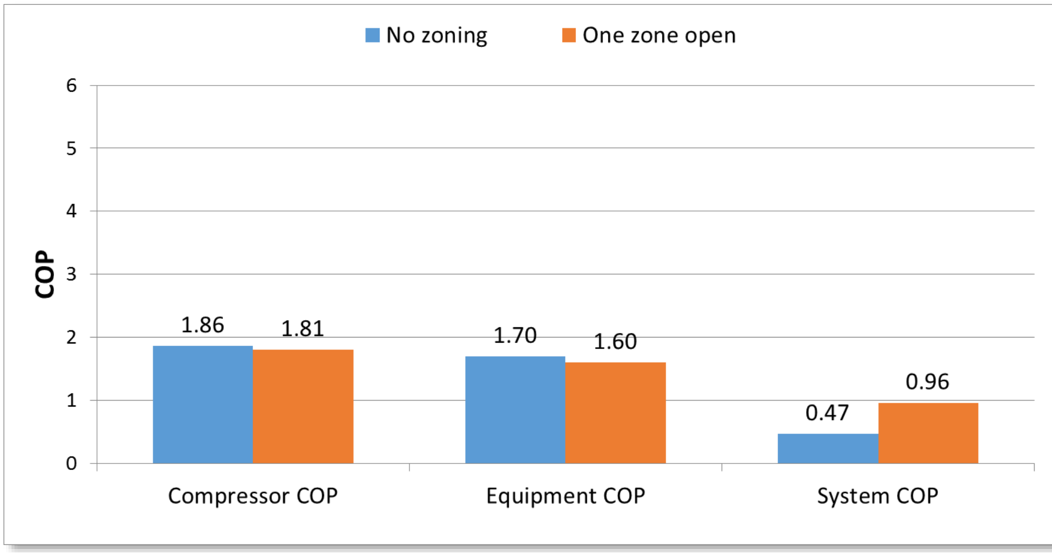


Source: EPRI

Effect of Zoning at Very Low Capacities

The positive effect of zoning on System COP described above becomes much more prominent as the equipment speed is reduced. Figure 113 presents the results for tests at 25 percent capacity and 25 percent airflow under the most extreme duct-zone temperature of 115°F (46°C). At this extreme temperature, the relatively low values of compressor and equipment COP (1.86 and 1.70 respectively) further deteriorate due to the effect of duct heat gain through the attic, reducing the System COP to only 0.47 when operated without zoning. However, when using only one zone at the same capacity, airflow and temperature conditions, the delivery effectiveness increases from 0.27 to 0.60 with a minimal impact on fan energy use. The result is that zoning at this condition increased the system COP by more than 100 percent, from 0.47 to 0.96.

Figure 113: Comparison of COPs at 25 Percent Synced Capacity and Air Flow Rate



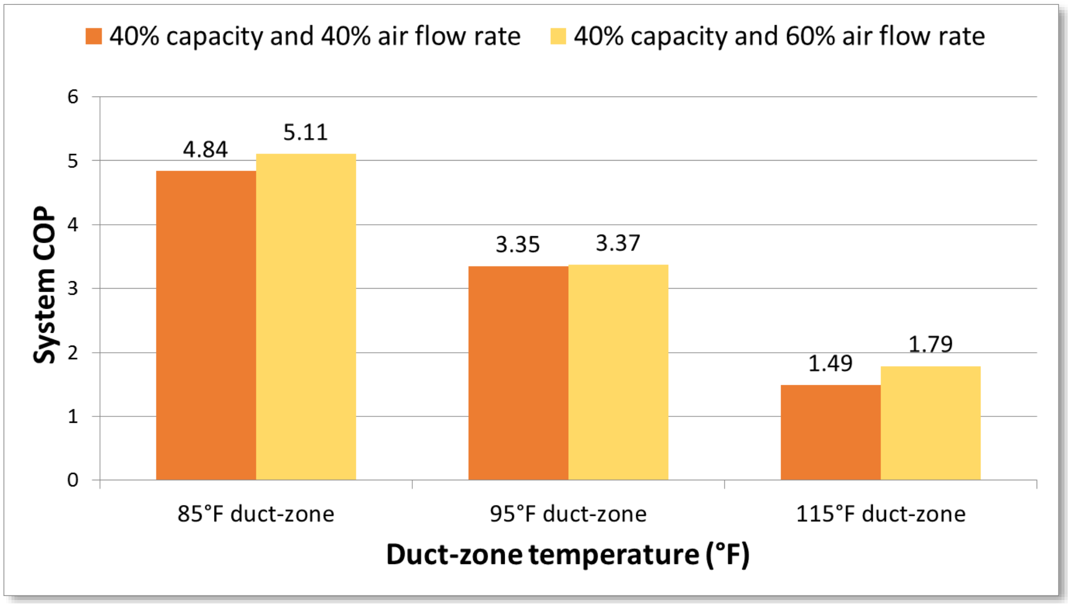
Source: EPRI

Performance at Independent Compressor and Fan Speeds

It was determined in Phase 1 that operating with a higher fraction of indoor air flow than equipment capacity was more efficient for non-zoning operations for compressor capacities lower than 80 percent. This was because the delivery effectiveness and compressor COP increase when the flow rate increases. While these behaviors would not change under zoned operation of the system, the increase in fan power (and therefore the fan heat) would be greater for zoning due to the higher fan pressure. Hence, testing was performed to determine whether higher air flow rates yield better System COP values than those reported in the Section on Performance at Synced Capacities and Airflows above.

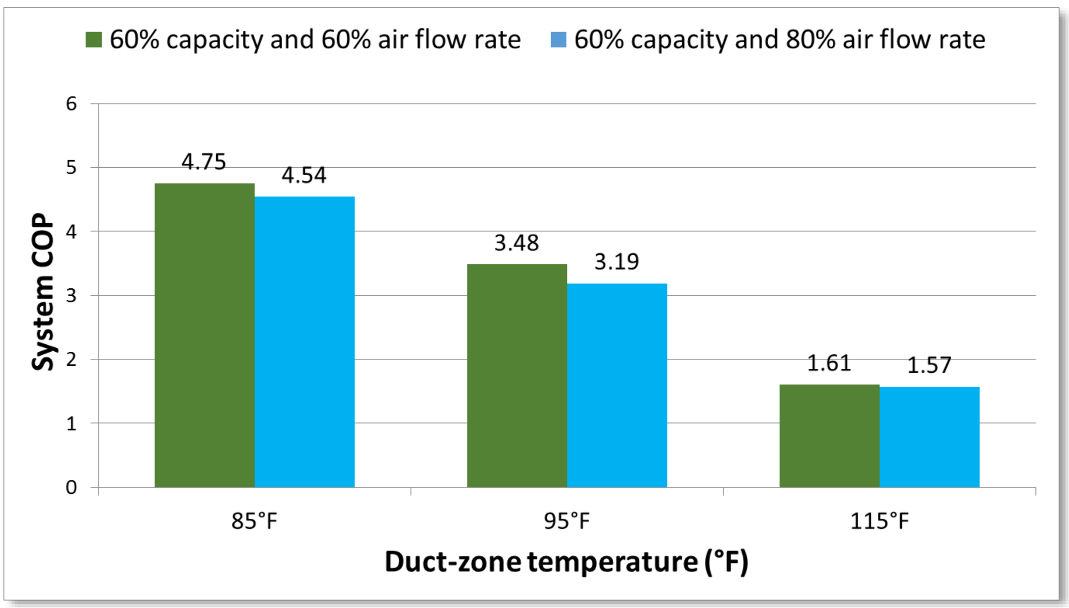
Figure 114, Figure 115, and Figure 116 show the variation of System COP at three different duct-zone temperatures when the air flow rate is maintained at a higher percentage than the compressor speed percentage. When the capacity is maintained at 40 percent and only two zones are open, the System COP is higher when the air flow rate is increased to 60 percent of its full flow rating than when the percentage of air flow is synced with the capacity at 40 percent. However, at higher capacities of 60 percent and 80 percent with three zones open, the System COP drops when the indoor blower is sped up. This indicates that for dwellings with zoning, synchronized operation of the compressor and fan may be more beneficial at higher speeds, while speeding up the fan may be preferred at lower capacities.

Figure 114: System COP vs. Duct-Zone Temperature with Zones 2 and 3 Open



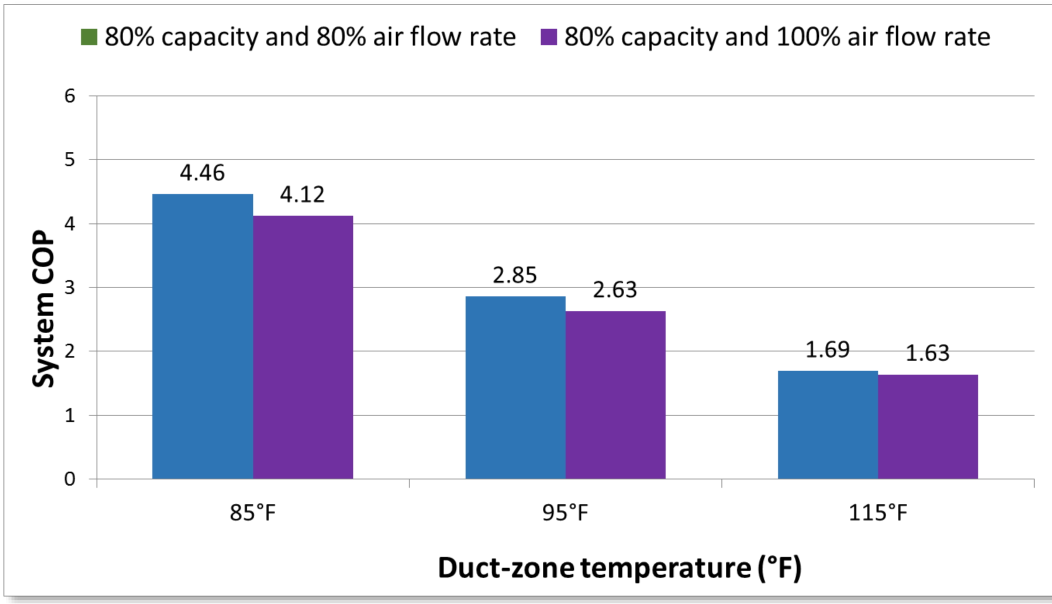
Source: EPRI

Figure 115: System COP vs. Duct-Zone Temperature with Zones 2, 3 and 4 Open



Source: EPRI

Figure 116: System COP vs. Duct-Zone Temperature with Zones 1, 3 and 4 Open



Source: EPRI

Mathematical Model of Delivery Effectiveness for Multi-Zone Applications

A mathematical model was developed with the purpose of providing simulated analyses of the detailed interactions between the variable-capacity/variable-speed air conditioner and the zoned duct system, and the combined impact of these on the efficiency of the system. The objective of the model was to capture the energy losses due to conduction heat transfer and leakage in the ducts during steady-state operation, and then to develop functional relationships to characterize the system according to the outdoor air temperature, humidity level, varying compressor speed, varying airflow rate and number of open zones. The basic assumptions of the model as well as their validity with respect to the experiments are listed below.

Model Assumptions

Duct thermal resistance is dominated by conduction through the insulated duct wall so the impact of convection resistances on the inside and outside of the duct was ignored. The overall heat transfer resistance of 6 British thermal units per hour per square feet per degree Fahrenheit (Btu/h ft^2 °F) was used for validation with the experiments. For comparison, the convection resistance on the inside of the duct is less than 1 percent of the conduction resistance through the wall. The model can be modified to include other thermal resistances if deemed appropriate for other situations.

The temperature throughout the attic is assumed to be uniform.

Thermal regain, or the phenomenon by which heat transfer through the duct walls is split among the various other heat transfer pathways (through roof, ceiling, or ventilation) was ignored.

Under such assumptions, past work has defined the delivery effectiveness of a single duct as the fraction of the sensible energy imparted to the duct system by the cooling equipment that is delivered to the space at the registers (Modera and Treidler, 1995, Francisco et al., 1998):

$$\xi_{del} = \frac{\dot{m}_{grille} C_p (T_{room} - T_{grille})}{\dot{m} C_p (T_{rp} - T_{sp})} \quad \textbf{Equation 39}$$

However, the limitation of this definition is that it doesn't capture the effect of humidity added to the system caused by air leaking into the return duct. It is possible that if the duct-zone air is more humid than the return air, the enthalpy of the air supplied at the grille could be higher than the return air enthalpy at extremely hot duct-zone temperature. To account for this limitation, the delivery effectiveness has been re-defined as:

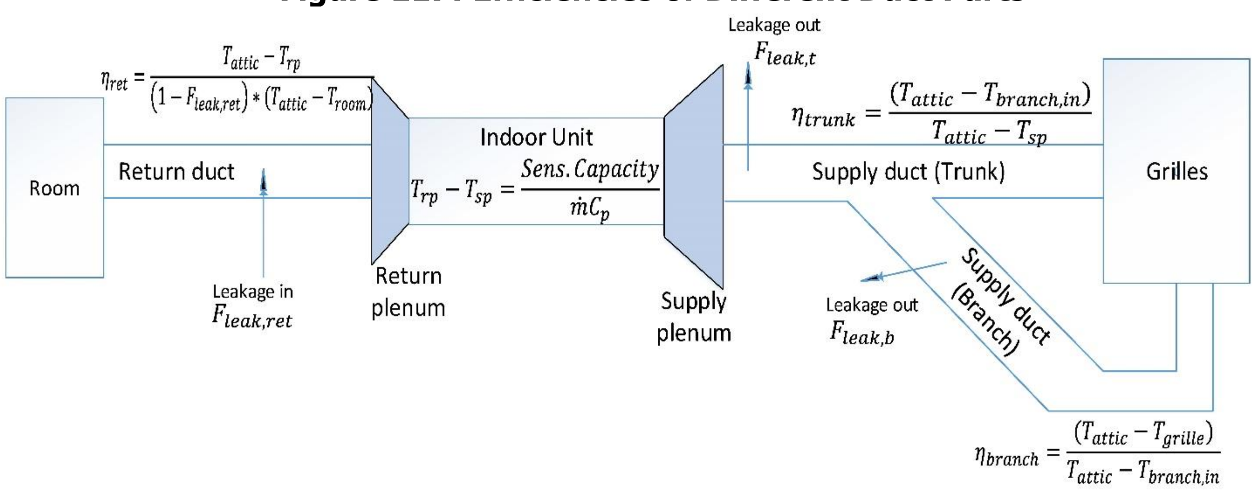
$$\xi_{del} = \frac{\dot{m}_{grille} C_p (h_{room} - h_{grille})}{\dot{m}_{evap} C_p (h_{rp} - h_{sp})} \quad \textbf{Equation 40}$$

Model Development

Consider the duct system shown in Figure 117. The mass-flow rate of the air passing through the indoor unit is represented by the variable $\dot{m}$. This represents the flow taken from the room, $\dot{m}(1 - F_{leakage,ret})$, and combined with the air entering the return duct through leaks, $\dot{m}F_{leakage,ret}$, where $F_{leakage}$ is the mass fraction of air leaking into the return duct. The supply system is split into the supply trunk and the supply branches where the outlet condition of the supply trunk is the inlet condition for the supply branches. Air leakage from the supply trunk, $\dot{m}(F_{leak,t})$, reduces the overall flow entering the supply branches where additional air leakage in the branch system is then considered, $\dot{m}(F_{leakage,br})$. Using heat exchanger theory, the efficiency of each part of the duct system can be separately evaluated using Equation 41.

$$\Xi = e^{\frac{-UA}{\dot{m}C_p}} \quad \textbf{Equation 41}$$

Figure 117: Efficiencies of Different Duct Parts



Source: EPRI

By drawing a control volume around the trunk of the supply duct shown in e 117, the supply trunk delivery effectiveness can be calculated using Equation 42.

$$\Xi_{trunk} = \frac{h_{attic} - h_{br,inlet}}{h_{attic} - h_{sp}}$$ **Equation 42**

Similarly, the efficiencies of other components of the duct system can be written as:

$$\xi_b = \frac{h_{attic} - h_{grille}}{h_{attic} - h_{br,inlet}}$$ **Equation 43**

and

$$\xi_{ret} = \frac{(h_{attic} - h_{rp})(1 - F_{leakage,ret})}{(h_{attic} - h_{room})}$$ **Equation 44**

where the term $(1 - F_{leakage,ret})$ has been included in the numerator to capture the effect of air leaking into the return duct. Note that there is no leakage term in Equation 42 and Equation 43 since the air through the supply duct leaks out rather than in (due to positive pressures), and as a result, there is no change in the temperature of the air in the ducts (negligible due to very negligible pressure change). Finally, by drawing a control volume around the indoor unit:

$$h_{rp} - h_{sp} = \frac{\text{Capacity}}{\dot{m}C_p}$$ **Equation 45**

Equations 14 through 18 allow the delivery effectiveness to be re-written as:

$$\xi_{del} = \left(\xi_{br}\xi_t - \frac{(h_{attic} - h_{room})(1 - \xi_{ret}\xi_{branch}\xi_{trunk})(1 - F_{leakage,ret})}{\frac{\text{Capacity}}{\dot{m}C_p}}\right)(1 - F_{leakage,t})(1 - F_{leakage,br})$$

Equation 46

Equation 46 defines the delivery effectiveness of a duct system in terms of parameters that are generally known about a system; namely attic temperature, room temperature, air flowrate through the equipment, capacity, and the parameters of the duct (length, area, leakage, and duct insulation). The key implications of this equation with respect to the duct efficiency variation at different compressor and fan speeds have already been discussed in Chapter 2, and hence not repeated here for conciseness. However, with respect to the zoning experiments performed in Phase 2, it is noted that the overall delivery effectiveness is obtained by calculating the delivery effectiveness at only those grilles that are part of an open zone using Equation 46 and then obtaining the flow-weighted average of those quantities.

Model Results

Delivery Effectiveness values were calculated using the model developed above, at duct-zone temperatures of 85°F (29°C), 95°F (35°C), 105°F (41°C) and 115°F (46°C) and Western Climate Performance Mapping indoor conditions at different combination of compressor and fan speeds. The variation of delivery effectiveness has been plotted at 25 percent capacity in Figure 118 for 40 percent and 60 percent airflow rates for different zoning mechanisms. It is evident from the figure that the delivery effectiveness

1. increases as the number of open zones is reduced.

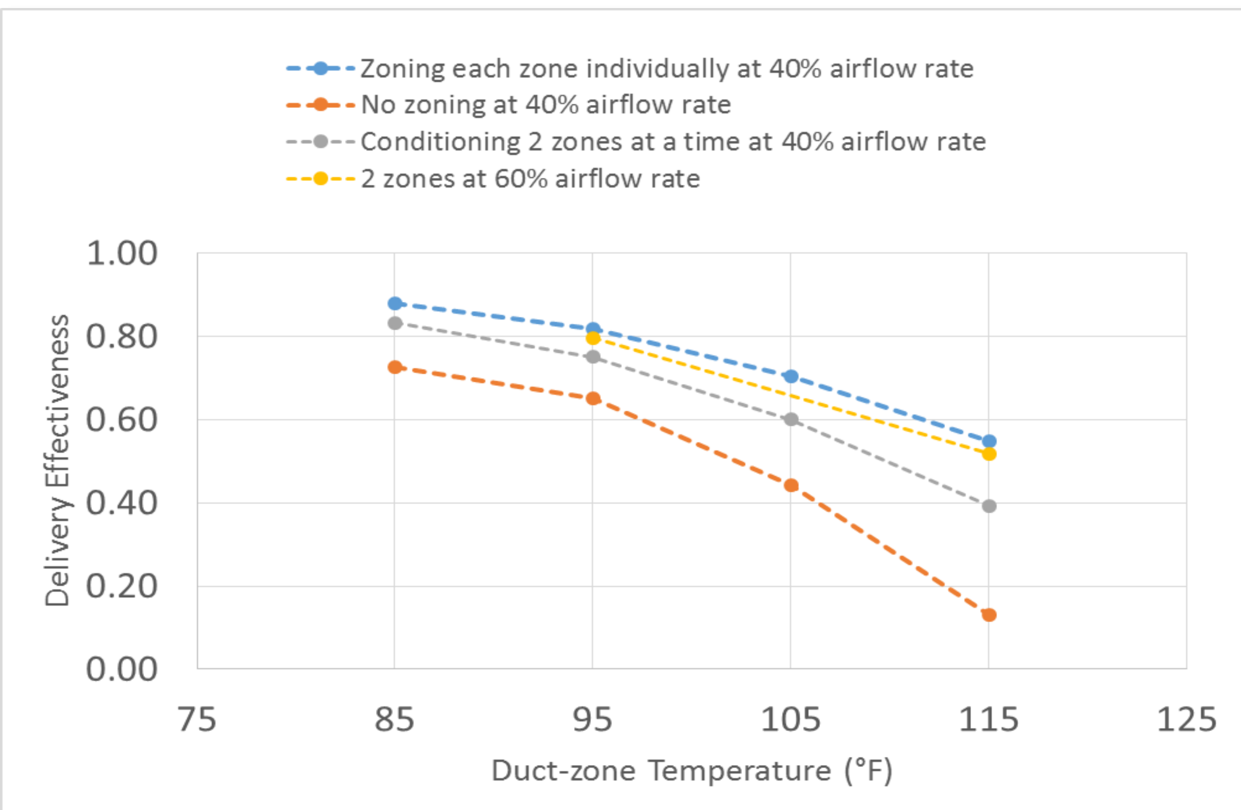
2. increases at the airflow rate is increased with the number of zones fixed
3. increases as the duct-zone gets cooler

The 2nd and 3rd observations above have also been reported in Phase 1 model results. Looking closely at the figure below, conditioning one zone at a lower air flow rate (40 percent) yields higher delivery effectiveness than conditioning two zones at 60 percent air flow rate. This is understandable because, since each zone roughly carries 25 percent of the total flow in case of no-zoning, conditioning two zones at 60 percent air flow rate would mean that each of the two zones carry about 30 percent of the air flow. The duct velocities in such a situation will be lower than when one zone carries 40 percent air flow. However, this doesn't necessarily mean that using one zone is always the best scenario for the system as a whole since the figure below only considers the duct system and does not take into account the additional fan power consumption. A combined equipment efficiency model and the duct model as described here will need to be used to predict optimum zoning conditions for highest System COP.

Figure 118 also predicts that substantial improvements in delivery effectiveness can be obtained when zoning is employed during part load conditions. Delivery effectiveness improves from 16 percent to 57 percent when the number of open zones is reduced from four to one, with a duct-zone temperature of 115°F (46°C), and the capacity and airflow at 25 percent and 40 percent respectively.

Figure 118: Comparison of Delivery Effectiveness at 25 Percent Capacity

75°F DB/62.5°F indoor condition



Source: EPRI

The primary takeaway from Figure 118 above is that delivery effectiveness tends to increase as duct velocity increases. The blue line showing 40 percent flowrate through a single zone has the highest duct velocity followed by two zones operating with 60 percent air flow (30 percent flow through each zone). The gray line shows two zones operating with 40 percent air flow (20 percent flow per zone) and the orange line illustrates the lowest delivery effectiveness when operating with 40 percent air flow through all four zones (10 percent flow through each zone). The trends shown Figure 118 are generally true for any zoned system where each zone is similarly sized.

Phase 2 Summary of Results

Phase 2 laboratory evaluations involved the assessment of the operation and performance of four technology features through a variety of steady-state and dynamic mode tests at three independent laboratories: EPRI, PG&E and WCEC.

The key results are summarized as follows:

1. Testing a Variable Capacity Heat Pump with R-32 at EPRI and PG&E: Both EPRI and PG&E labs evaluated the performance of the variable capacity heat pump (VCHP) using R-32 as a refrigerant. Laboratory findings are provided for R-32 as a refrigerant for residential HVAC systems that can be added to industry literature for use pending legislative approval of this refrigerant. Heating mode and cooling mode testing evaluated the performance of a VCHP, designed for R-410A, but tested with R-32 as a drop-in refrigerant. Three key points should be considered while reviewing both labs' test results:

    a. The VCHP systems tested have been designed for R-410A. R-32 was tested as a drop-in replacement refrigerant. Accordingly, the system was not optimally tuned for the differences in pressure and temperature of the R-32 refrigerant. This, coupled with minor differences in test conditions could be the cause of differing responses in terms of operations and results, as was observed for the two different units of the same system tested. (Repeatable test results would be expected for the operation of a system specifically designed and programmed for R-32.)

    b. Thus, in the Phase 2 testing, both labs evaluated what occurs in an R-410A system when it is replaced with R-32, allowing a general assessment of the system's performance relative to the baseline testing with R-410A in Phase 1. This general assessment serves to inform the next steps for optimizing the system design and characterization testing.

    c. PG&E's testing of the VCHP system with R-32 occurred after the system underwent rigorous Fault Detection and Diagnostic (FDD) testing, while the unit in EPRI's lab was not operated in the same manner/tests prior to the R-32 performance evaluation. During Phase 2, PG&E's unit exhibited discrepancies in repeatability and continuity of operation during testing, which had not previously occurred during Phase 1 for the same testing procedures (such as system shut off during testing, which did not occur during the EPRI testing). Thus, the system strain from the FDD testing may have been a possible cause for the differences in the results between the two labs.

After further evaluation of the results with the manufacturer, the VCHP at the PG&E lab seemed to be over-charged with R-32 based on the nominal charge level, high condenser pressure and high refrigerant subcooling. The manufacturer's manual recommends for R-32 refrigerant charging, to hold to a system subcooling level of 7 to 9°F [–14° to –13°C]with the compressor at high speed for 95°F (35°C) ambient (cooling mode) and 47°F (8°C) ambient (heating mode). While the refrigerant subcooling temperature averaged 9.3°F (–12.8°C) for R-410A testing in Phase 1, it averaged 10.3°F (–12.2°C) for R-32 testing in Phase 2 for 95°F (35°C) ambient with the compressor running at high speed for the PG&E unit.

That said, following are the summary results from each of EPRI and PG&E's testing of Variable Capacity Heat Pump with R-32 as an alternative refrigerant to R-410A.

- EPRI's results: R-32 demonstrated an ability to be an effective, low GWP replacement for R-410A in the variable capacity heat pump from an equipment performance and functionality perspective. The usage of R-32 in HVAC equipment offers a potential mechanism for peak power reduction in the warmest California climates.
  - *In Cooling mode, trends observed of the R-32 variable capacity heat pump are comparable to the trends observed for R-410A as the refrigerant.* At 95°F (35°C) outdoor temperature, where nominal capacity is determined, the minimum output of the R-32 system was 29 percent of the maximum capacity. In R-410A testing of the variable capacity system, the minimum capacity was 30 percent of the maximum capacity at 95°F (35°C). The R-32 variable capacity system demonstrated increased efficiency at part-load operation, and the relative increase in efficiency from maximum to part-load operation increased with decreasing outdoor temperature.
  - The retrofit of R-410A to R-32 resulted in cooling efficiency increases of 6—9 percent, 1—3 percent, and 2—3 percent for maximum, intermediate and minimum operation, respectively.
  - With the implementation of R-32 in the variable capacity heat pump, the peak cooling performance improved by 6.7—8.2 percent. For residential equipment ranging from 2 to 4 tons, the R-410A variable capacity heat pump provides a potential peak reduction of 80—200W over a baseline 14 SEER system. Implementation of R-32 in the variable capacity heat pump provides an additional potential peak reduction of 125—475W depending upon size of the equipment.
  - In heating mode, the COP with R-32 ranged from ~2.4-4.1 for outdoor temperatures between 15 and 65°F (–9 and 18°C). The maximum heating capacity curves of R-32 and R-410A variable capacity heat pump compared the same with 2 improvements of R-32 by 10 percent and 5 percent at 62°F (17°C) and 25°F (-4°C), respectively.

- *PG&E's results*: Swapping out the original charge of R-410A for R-32 did not significantly affect the performance of the heat pump. On average, there was less than a 10 percent overall decrease in performance over the range of performance metrics, which is not a significant loss. Comparing the results from the new tests with R-32 to the same tests with R-410A refrigerant in Phase 1:
    - *In cooling mode:*
        1. Capacity was about 3 percent higher on average but ranged from 34 percent higher to 15 percent lower in specific tests. The cooling capacity measured using the current AHRI Standard rating conditions (0.10" external resistance) produced results 14 percent higher than its rating at 2.28 tons.
        2. Total power consumption was 6 percent higher on average and ranged between 26 percent higher to 5 percent lower.
        3. Because of the higher rise in power compared to the rise in capacity, the cooling EER was 3 percent lower on average; ranging between 27 percent higher to 28 percent lower. At the AHRI Standard rating conditions (0.10" external resistance), the cooling EER was 14.7 Btu/Wh, or 8 percent higher.
        4. The SEER calculated from several of the steady-state tests was 23.3 Btu/Wh at the 0.10" external resistance, which is 3 percent lower than the result with R-410A. At the 0.45" external resistance at maximum blower speed, the SEER was calculated as 19.4 Btu/Wh, which was 12 percent lower than the result from the same test with R-410A.
    - *In heating mode:*
        1. Capacity was about 3 percent lower on average but ranged from 37 percent higher to 97 percent lower in specific tests. (The much lower capacity numbers were from the low speed tests at very cold temperatures when the heat pump would not produce much and is a highly unlikely real operating mode as the system would only be running at its maximum speed or have switched to its backup source under these conditions.) Heating capacity at AHRI Standard rating conditions was 3 percent lower at 24,500 Btu/hr.
        2. Total power consumption was 10 percent higher on average and ranged between 52 percent higher to 3 percent lower.
        3. Because of the higher rise in power compared to the rise in capacity, the heating COP was 12 percent lower on average; ranging between 24 percent higher to 98 percent lower. At the AHRI Standard rating conditions the COP of 3.77 was 8 percent lower than the same tests with R-410A.
        4. The HSPF calculated from several of the steady-state tests for Region IV was 4 percent higher at the 0.45" external resistance, but 4 percent lower at the 0.10" external resistance.

Noting the results above, although the use of R-32 refrigerant shows marginal efficiency and load reduction benefits over R-410A refrigerant, Phase 3 testing will use R-410A refrigerant in the installed unit. This is because R-32 has not been approved by the Environmental Protection Agency (EPA) and California Air Resources Board (ARB) for use in the United States.

2. Integration of Zonal Control and Variable Capacity Space Conditioning: Zonal control and variable capacity offers a potentially effective integration of two technologies for improved efficiency. Understanding the functionality and utility of Zonal Control with a variable capacity heat pump system can provide targeted energy savings. The efficiency impact of zoning is largely dependent upon on the temperature offset for unoccupied zones and the subsequent load reduction on the HVAC system. Laboratory testing demonstrated altered variable capacity performance and functionality with the implementation of zoning. Field evaluations may further assess the performance and functionality of a zoned, variable capacity system.

3. Variable Capacity Space Conditioning connected to a Ductwork System in Multi-Zone Configuration

   The multi-zone operation of a variable-capacity/variable-fan residential air-conditioner utilizing ductwork routed through an attic has been studied experimentally and analytically at the UC Davis Western Cooling Efficiency Center laboratory. The objective was to determine the optimum zoning controls and the optimum operating speeds for both the compressor and indoor fan at the optimum zoning percentages for achieving maximum system efficiency in hot and dry California climates. The data collected describes the performance characteristics of the system operating when—a) varying compressor speed and indoor fan speed together, and b) varying indoor fan speed while holding compressor speed fixed. The results highlight the potential for implementing zonal control in variable-capacity/variable-speed cooling systems using R-410A refrigerant to reduce residential energy use in California and corroborates the proposal to combine these technologies to create an integrated efficiency solution for maximum energy efficiency. The major results of the laboratory can be summarized as follows:

   a. Reducing the number of active zones improves the delivery effectiveness of the duct system (reduces duct losses) and increases fan power

   b. In general, the optimal number of zones for maximizing System COP increases as the capacity/airflow percentage is increased.

   c. It was shown in Phase 1 testing that during very hot outdoor conditions the highest system efficiency occurred at maximum operating speed. Phase 1I testing showed that reducing the number of active zones while operating the equipment at low speed produced similar System COPs as the maximum achieved in Phase 1 tests. This result is especially relevant when considering how to control variable capacity systems during a demand response event.

   Examined in more detail, the multi-zone testing results demonstrated that:

a. Delivery effectiveness of the duct system has an inverse relationship with the number of zones employed for any given capacity/airflow percentage and duct-zone temperature. That is to say, the delivery effectiveness is highest for single-zone operation and progressively decreases as the number of zones increases.

b. System COP values when operating the equipment under zoned conditions are generally higher than the values obtained under non-zoning operation for the same capacity/airflow percentage and duct-zone temperature. This behavior is due to the improved delivery effectiveness of the duct system which increases at higher duct velocities. There is a tradeoff with higher fan power for zoned operation which creates an optimal zoning that does not necessarily coincide with the zoning that achieves the highest delivery effectiveness. In general, the optimal number of zones for maximizing System COP increases as the capacity/airflow percentage is increased.

c. Zoning is more effective at higher duct-zone temperatures. This is because the percentage increase in delivery effectiveness is higher due to zoning when the duct-zone temperatures are hotter, whereas the additional blower power consumption due to zoning is independent of temperature.

d. For very hot duct-zone temperature, the heat pump equipment using R-410A is capable of operating at lower capacities/air flow rates using a zoning mechanism that yields a System COP value comparable to the maximum System COP when operating without zoning. Recalling that lowering the capacity/air-flow rates hurt the System COP for hotter duct-zone conditions when operating without zoning. This result implies that in very hot climates, zoning can be employed in variable-capacity equipment to respond to demand-response events from the utility without compromising on efficiency.

e. When zoning is employed, operating with a higher fraction of indoor air flow than capacity increases system efficiency only at low capacities. When the capacity is 60 percent or more, increasing the blower fan speed relative to the compressor speed reduces the System COP when zoned, due to the higher fan power consumption.

f. More efficient control strategies are needed to optimize the performance of heat pumps connected to ductwork located in an attic for hot and dry California climate zones. The system balance is affected by the duct-zone temperatures, which invites the need for revising ducting standards.

g. A VCHP connected to a multi-zone configuration with ductwork has higher system COP under zoned conditions, compared to non-zoning for the same capacity/airflow percentage and duct-zone temperature.

4. Fault Detection and Diagnostics: Both the heat pump and the furnace have an extended list of faults that are detectable and which can aid in the repair and maintenance of the system. The testing of the fault detection and diagnostic (FDD) capabilities was limited in scope to primarily those faults that were thought to be the most likely to occur during normal usage. Thirteen of the 51 listed fault codes for the heat pump were triggered,

as well as 5 of the 25 codes listed for the furnace. The result of the evaluation was that the FDD system was very good at correctly identifying the cause of a fault condition when something *had* gone very wrong, but it was not as good at alerting the user that something *was* going wrong and should be attended to for optimal performance and preventive maintenance. This capability is thought to be present with the existing components and may just require a more sophisticated software upgrade to make happen. This change should retain some conservatism such that the system will not trigger too many alerts so that the end user stops paying attention and does not take action.

Fault Detection and Diagnostic (FDD) systems can be used to improve HVAC unit performance by alerting users and contractors when degradation or malfunction is taking place. This permits timely correction of an incipient fault that could otherwise result in a system failure or rapid response to a failure that has already occurred.

5. Additional testing on Dual Fuel (Intelligent Heating): Calculations were performed to assess the economics of dual fuel heat pumps in selected locations in California. California energy prices tend to be high relative to National averages and natural gas prices have not kept pace with electric prices. Despite these factors there are still situations, as illustrated in Appendix C, where the Next-Gen RSCS can provide attractive savings.

   The fact that operation of the Next-Gen RSCS can be adjusted as utility prices vary permits the homeowner the option to benefit from future changes in utility prices that might reduce the ratio of electricity to gas prices in the future. The assurance that the homeowner will be able to experience the lowest future heating costs possible, is an important attribute of dual fuel heat pump capability and increases the value of this feature to potential purchasers of the Next-Gen RSCS. Testing confirmed the functionality of the dual fuel heat pump concept in all possible modes of operation.

Project Benefits Based on Phase 1 and Phase 2 Results

Per California's Energy Efficiency Strategic Plan (California's Long Term Energy Efficiency Strategic Plan, 2008), HVAC is the single largest contributor to peak power demand in the state, comprising up to 30 percent of total demand in the hot summer months. The next-generation space conditioning system's combined technologies could significantly reduce peak demand. Variable-capacity systems have the unique attribute of going to a state of higher operating efficiency when the compressor speed is reduced. For a Demand Response event, a reduction in compressor speed provides a reduction in power draw, but with a correspondingly smaller reduction in cooling capacity. Per the Strategic Plan, the CEC estimates that a peak demand reduction of 1,096 MW could be achieved through high-quality HVAC installations by 2020. If next-generation air conditioners, or similar technology, were adopted by California energy codes, the potential energy savings is on the order of 5 times greater, or 3.62 GWh per year with a rough energy cost value of greater than $5 Billion over the equipment lifetime.[6] This will benefit the ratepayers through avoided electric capacity and energy costs, providing

[6] 1 PPEU = 100MW peak generation. Generation = load + 15% T&D losses + 15% reserve margins.

greater reliability, lower costs, and increased reliability for California Investor-Owned Utility (IOU) electricity ratepayers.

Highlights of the benefits of this project based on the Phase 1 laboratory evaluation are:

- Variable capacity heat pump (VCHP) performs at higher system efficiency when operating at lower speed settings instead of rated levels.
- VCHP with demand response capability enables utilities to reduce peak demand.
- More efficient control strategies are needed for heat pumps connected to ductwork located in an attic for hot and dry California climate zones. The system balance is affected by the duct-zone temperatures, which invites the need for revising ducting standards.
- The next-generation VCHP has demonstrated its versatility with intelligent heating capability and integrated ventilation configuration.

Highlights of the benefits of this project based on the Phase 2 laboratory evaluation are:

- Until legislative action is taken to approve of R-32 as a refrigerant for residential HVAC systems, the findings of Phase 2 can be added to the literature, detailing experimental results evaluating a variable capacity heat pump system, designed for R-410A, but tested with R-32 as a drop-in refrigerant, and assessing its performance in both heating and cooling mode.
- Understanding the functionality and utility of Zonal Control with a variable capacity heat pump system can provide targeted energy savings. Recognizing that the variable capacity system performance is altered with zoning, the system efficiency with zoning is largely dependent upon on the temperature offset for unoccupied zones. This can be better evaluated during field demonstration.
- A variable capacity heat pump connected to a ductwork system in a multi-zone configuration has higher system COP when operating under zoned conditions, compared to non-zoning for the same capacity/airflow percentage and duct-zone temperature. Additionally, the benefit of zoning is realized at higher duct-zone temperatures.
- Fault Detection and Diagnostic (FDD) systems are helpful tools that can be used to improve HVAC unit performance. FDD's benefit is to alert users when an issue is taking place that could result in an incipient fault. This can permit the user or contractor to conduct preventive maintenance or take remedial measures to avert the fault condition.

Chapter 5 provides qualitative and quantitative information on the potential California energy, demand and environmental benefits resulting from this project.

CHAPTER 4: Phase 3 Field Evaluation (Task 4)

This chapter outlines the Phase 3 Field Evaluation of the Next-Generation Residential Space Conditioning System (Next-Gen RSCS) technology in three occupied residential households, one in each of the California IOU service territories (PG&E, SCE, SDG&E). The host sites were selected for the CEC's recommended climate zones to provide qualitative and quantitative assessment of the Next-Gen RSCS in both the heating and cooling modes. The field study included retrofitting each home with a new ducted split variable capacity heat pump unit (VCHP) with refrigerant R-410A, provided by Daikin/Goodman. Table 37 shows which key features were studied in each phase. For the field evaluation, each home, depending on its configuration and climate, evaluated a subset of the key features assessed in Phase 1 and Phase 2 lab evaluation, namely:

1. Variable Capacity Compressor
2. Variable Speed Blower
3. Auto Demand Response
4. Integrated Ventilation Control
5. Dual Fuel (intelligent heating)
6. Zonal Control
7. Duct loss assessment for single/multi-zone configurations
8. Fault Detection and Diagnostics (as feasible/appropriate)

Table 37: Technology Features Tested by Phase of the Project

Technology	Phase 1 Lab Test	Phase 2 Lab Test	Phase 3 Field Test
Variable-Capacity Compressor	✓		✓
Variable-Speed Blower	✓		✓
Integrated Ventilation	✓		
Demand Response	✓		✓
Dual Fuel (intelligent heating)	✓		✓
Duct-loss assessment for single-zone	✓		
Alternative Refrigerants		✓	
Fault Detection & Diagnostics		✓	
Zonal Control		✓	✓
Duct-loss assessment for multi-zone		✓	✓

Source: EPRI

Host Sites Specifications

The host sites were selected for the CEC's recommended climate zones to provide qualitative and quantitative assessment of the Next-Generation Residential Space Conditioning System (Next-Gen RSCS) in both the heating and cooling modes. The field study included retrofitting each home with a new ducted split variable capacity heat pump unit with refrigerant R-410A, provided by Daikin/Goodman, similar to the unit/model tested in the laboratory evaluations in Phase 1 and Phase 2.

Table 38 lists the specifications of the three homes that participated in the field evaluation. Daikin/Goodman Certified Professionals were identified for each of the host sites to be the designated HVAC contractors for the system sizing, installation, commissioning and support for the installation of the Measurement and Verification (M&V) instrumentation.

Table 38: Field Testing Host Sites Specifications

Attribute/IOU	PG&E	SCE	SDG&E
City, Zip Code	West Sacramento, 95961	Chino Hills, 91709	San Diego, 92124
Climate Zone	12	10	7
Area (sq.ft.)	2507	1850	1906
Home Vintage	2008	1993	1980
Existing Ducting, Heating System	Ducted AC with Gas Furnace	Ducted AC with Gas Furnace	Ducted AC with Gas Furnace
Location of Ducts	Attic	Attic	Attic
AC Size (tons)	3-ton Condenser 4-ton AHU	4	4
Floors	2	2	1
Number of Residents	4 + 1 pet	4 + 1 pet	1+3 pets

Source: EPRI

Figure 119, Figure 120, and Figure 121 provide photographs from the three houses (all single-family detached homes) used for the field testing, including the ductwork and equipment installed for each home. All three homes are slab on grade construction. A summary of each home's characteristics from the HERS rating test results, pre- and post-installation is provided in Table 38. Included are the specifications of the existing HVAC systems and the Next-Gen RSCS that replaced them.

The ducting in all 3 homes was replaced when the new Next-Gen RSCS was installed. However, R-6 ducting was used for the PG&E home while R-8 ducting was used for both the SCE and SDG&E homes based on the Phase 1 and Phase 2 laboratory evaluation results on duct-loss assessment.

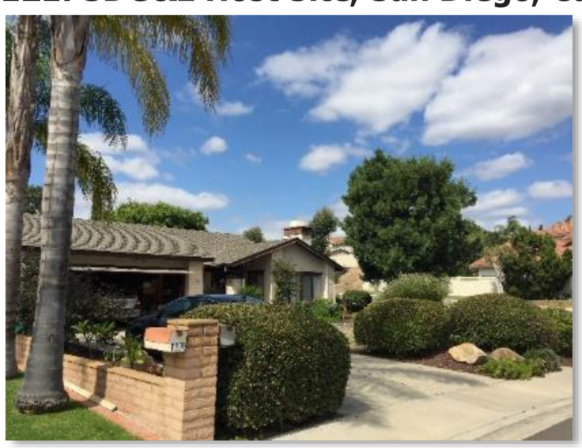
Figure 119: PG&E Host Site, Sacramento, California

Source: EPRI

Figure 120: SCE Host Site, Chino Hills, California

Source: EPRI

Figure 121: SDG&E Host Site, San Diego, California



Source: EPRI

Table 3839 provides descriptions of the HVAC units in place in the three homes prior to the Next Gen units being installed.

Table 39: Summary of HERS Testing Results and Pre- and Post-Installation of Next-Gen RSCS at Three Host Sites

Site	PG&E Host Site	SCE Host Site	SDG&E Host Site
Climate Zone	12	10	7
Conditioned Floor Area (sq.ft.)	2507	1670	1916
Number of Bedrooms	3	4	3
Pre-Installation of Next-Gen RSCS: HVAC System Specifications that Were Replaced			
Manufacturer	Carrier	Goodman	Carrier
Model Number – Cooling	24ABA336A300 (2004 model)	CK49-18	569a048rcue
SEER	14	14	20
Model Number – Heating	58STX090-13116 (2004 model)		Day and Night #394Jaw048100
AFUE	80	80	91
Nominal Cooling Capacity of Condenser (ton)	3	4	4
Heating Capacity (kBtu/h)	71	60	
Duct R-Value	R-4.2	R-6	R-6
Leakage Factor	99 CFM25 (94.14% duct efficiency)	0.15	0.31
Post Installation			
Manufacturer (Heating)	Daikin	Daikin	Daikin
Model Number (Heating)	DM97MC1005CNAA	DM97MC0804CNAA	DM97MC0804CNAA
AFUE	0.97	0.97	0.97
Rated Heating Capacity, Output (Btu/h)	97000	77600	77600
Manufacturer (Cooling)	Daikin	Daikin	Daikin
Model Number (Cooling)	DZ20VC0481AB	DZ20VC0481BB	DZ20VC0481BB
Refrigerant Type	R-410A	R-410A	R-410A
Nominal Cooling Capacity of Condenser (ton)	4	4	4
System Rated Cooling Capacity at Design Conditions (Btu/h)	48000	48000	48000
SEER	20	20	20
Heating Capacity (kBtu/h)	97	77	77.6

Site	PG&E Host Site	SCE Host Site	SDG&E Host Site
Duct R-Value	R-6	R-8	R-8
Leakage Factor	0.15	0.15	0.15
Calculated Target Allowable Duct Leakage (cfm)	315.74	250.64	240
Actual Duct Leakage Rate from Leakage Test Measurement (cfm)	166	246	188
Required Minimum System Airflow Rate (cfm/ton)	300	350	300
Required Minimum System Airflow Target (cfm)	1200	1400	1200
Actual System Airflow Rate Measurement (cfm)	1255	1534	1266
Extension of Existing Duct System (40+ ft.)	R-8	-	-

Source: EPRI

Field Testing Objective and Features

The objective of the field evaluation is to assess the performance of Next-Gen Residential Space Conditioner Systems (Next-Gen RSCS) installed at residences with American-style ducting, and determine the energy efficiency benefits from the individual and collective technology features. The objective will be achieved by providing qualitative and quantitative assessment of the following research questions:

- Assess the functionalities of integrating energy efficiency features with a residential variable capacity heat pump (VCHP) based on customer operation and experience
- Assess the performance of a VCHP system installed with typical American-style ducting to determine the energy efficiency benefits from the individual and/or collective technology features
- Assess the control strategies implemented and their benefits as well as the VCHP interaction from these control strategies
- Determine how the VCHP performance is affected or could be improved by the integration of other control strategies

In the Phase 1 and Phase 2 Laboratory Evaluation, all the technology features were tested. For Phase 3, three of the features are not evaluated after thorough consideration of the testing procedure and how well field testing could provide beneficial insight for the project objectives (Table 37). The three features *not* tested during Phase 3 were:

- Alternative Refrigerants: R-32 was tested in the laboratory evaluation Phase 2 of the project. Since the use of R-32 has not been approved yet by the regulatory bodies, R-32 cannot be tested in residential homes for the field test of the project. Therefore, the lab testing (Phase 2 Interim report) must be relied upon for demonstration of the system performance.

- Fault Detection and Diagnostics (FDD): The lab evaluation of FDD (Phase 2 Interim report) showed that the FDD alerts would occur when the unit was at the verge of breakdown/shut-down. Thus, the manufacturer has taken this information to improve upon their control algorithm for alert notifications. The model available for the field installation would not have any adjustments made from the lab results. The integrity of the new units installed in the homes could have been jeopardized if the FDD feature were field tested.
- Integrated Ventilation: All the approved host sites are pre-2013 construction, meaning they don't have ventilation requirements per Title 24 -2013 standards. Thus, if the Integrated Ventilation feature was evaluated (adding Heat Recovery Ventilator (HRV), there wouldn't be a meaningful baseline comparison. Instead adding a HRV to these homes would increase their energy consumption. Thus, the lab results shown in Phase 1 interim report were used to provide the energy savings for each California climate zone by adding HRV to homes that have standard ventilation requirements (post 2013).

All three residential host sites received the same model 4-ton ducted split variable capacity heat pump system with indoor gas furnace. At the PG&E and SCE homes, the indoor unit and furnace are installed in the attic, in horizontal configuration, while at the SDG&E home, they are installed in the garage in a vertical configuration (Figure 122, Figure 123, Figure 124, Figure 125, Figure 126, and Figure 127).

Figure 122: PG&E Host Site Attic with Horizontal Configuration of Indoor Unit and Furnace

Source: EPRI

Figure 123: EPRI M&V Data Monitoring Box at PG&E Host Site

Source: EPRI

Figure 124: EPRI M&V Data Monitoring Box and EWC Zonal Control Board in Attic at SCE Host Site

Source: EPRI

Figure 125: CTKO4 Thermostat and Wireless Temperature/Relative Humidity Sensor at SCE Host Site

Source: EPRI

Figure 126: Vertical Configuration of Indoor Unit and Furnace in SDG&E Host Site Garage

Source: EPRI

Figure 127: Next-Gen RSCS Outdoor Unit at SDG&E Host Site Backyard

Source: EPRI

Table 40 lists which technology features were installed for testing at each host site. All homes had been equipped with a gas furnace for their heating source. Thus, upgrading their HVAC system to a Dual Fuel (intelligent heating) heat pump (electric powered heat pump with gas back-up) didn't require any major changes to the home's hook-up lines.

For Automated Demand Response (DR) testing, Daikin/Goodman's new DR-enabled thermostat has not been launched in the market yet. Daikin's new smart thermostat One+[7] is scheduled for commercial launch in spring 2019, which is past the project period of performance. Thus, Auto-DR testing was conducted in the field in the same manner it was conducted in the laboratory evaluation, by using Daikin/Goodman's software program, called RAM monitor, installed on a laptop/PC that is connected to the homeowner's Next-Gen RSCS unit. Since the new DR-enabled thermostat was not yet commercially available, the DR testing

[7] 2019, Daikin One+ Smart Thermostat, http://www.daikinone.com/

was conducted in only one home, at the PG&E host site. The PG&E homeowner is also a researcher from WCEC team that has been closely engaged and involved in all phases of this project, as well as conducting the analysis for zonal control in all 3 homes.

Based on the laboratory evaluation of Zonal Control and duct-loss assessment, it was desirable to evaluate the variations in the functionality of Zonal Control with variable capacity heat pump in the field evaluation. Accordingly, the newly installed ducts in each home were setup for different zoning configurations: 2 zones for SDG&E host site, which is a one-story home; 3 zones for SCE host site, which is a two-story home; and 4 zones for the PG&E host site, which is a two-story home.

Table 40: Variable Capacity Heat Pump Technology Features Evaluated at Each Host Site

Technology Attribute	PG&E (2 story)	SCE (2 story)	SDG&E (1 story)
Variable-Capacity Compressor	Yes 4-ton system	Yes 4-ton system	Yes 4-ton system
Variable-Speed Blower			
Dual Fuel (intelligent heating)	Yes	Yes	Yes
Auto Demand Response	Yes (using Ram monitor)	No	No
Zonal Control (# of zones, duct insulation R-value)	Yes 4 zones with R6	Yes 3 zones with R8	Yes 2 zones with R8
Alternative Refrigerants	No	No	No
Fault Detection & Diagnostics	No	No	No
Integrated Ventilation	No	No	No

Source: EPRI

Field Testing Research Questions and Instrumentation

Having defined the scope of the field evaluation as outlined in section 4.2, the team determined the key research questions that would be addressed for each of the Next-Gen RSCS features under evaluation. The following outlines the research questions and methodology of testing for each of the 5 features:

1. Variable Capacity Heat Pump (2 features: variable capacity compressor and variable speed blower):
    a. Research Question: Quantify the field performance of Next-Gen RSCS through its average monthly energy consumption, peak demand and performance curve relative to baseline system.
    b. Baseline System: Single Speed 14 SEER heat pump performance data from EPRI laboratory testing based on field operational data

c. Field Test Metrics: Field testing equipment performance using system efficiency (EER), COP, and power map. Compare field equipment performance data with reference to single-speed lab test data and Next-Gen RSCS lab test data.

2. Dual Fuel (Intelligent Heating)

d. Research Question: What are the savings potential for having a dual-fuel variable capacity heat pump? What temperature setpoints are used for back-up heat to switch on? How do the customers interact with the dual fuel mode? Determine utility rates to calculate breakeven temperature and set this temperature.

e. Baseline Assessment: Gas Furnace only energy consumption (historical data)

f. Field Test Metrics: Quantify monthly gas consumption vs electric consumption with heat pump heating. Evaluate which setpoints are used for each service territory. How does customer interaction impact system operation?

3. Auto Demand Response:

g. Research Question: What is the Next-Gen RSCS power reduction as a function of capacity reduction? How quickly can the demand response signal be sent to the unit and the equipment respond to it?

h. Baseline Assessment: Next-Gen RSCS auto demand response (ADR) lab testing results

i. Field Test Metrics: Test different load reduction options similar to the lab testing. Determine equipment power and capacity change during ADR event and include customer feedback on comfort during demand response.

4. Zonal Control:

j. Research Question: What is the Next-Gen RSCS duct delivery effectiveness for different zonal configurations, especially for lower duct velocities?

k. Baseline Assessment: Laboratory testing for zonal control and duct delivery effectiveness (Phase 1 and Phase 2)

l. Field Test Metrics: Evaluate duct delivery effectiveness under a range of zonal control configurations to understand the impact of lower duct velocities. Different configurations include default settings with the standard zonal controls (EWC control board) and different setting adjustments on the EWC control board or the system thermostat (ComfortNet Controller).

Accordingly, the instrumentation list procured and installed in each of the homes, corresponded to the measurement list below, as part of the measurement and verification (M&V).

Measurement list for each host site covered:

- Whole house power
- Unit power consumption: indoor unit and outdoor unit

- Natural gas consumption (gas flowmeter to furnace unit)
- Outdoor/Ambient air temperature and relative humidity
- Main Supply and Main Return plenums: air temperature and relative humidity
- Attic air temperature and relative humidity
- Indoor air temperature and relative humidity, in each zone (sensor installed near thermostat for designated zone)
- Temperature at each grille (for duct delivery effectiveness assessment)
- Differential pressure (with respect to the house pressure) across each zone control damper, which is used to correlate with air velocity (conducting flow map measurements) to determine airflow

The Next-Gen RSCS was installed between February and March 2018 in each of the 3 homes. The M&V instrumentation was installed on June 20-21, 2018 for the PG&E home and required further troubleshooting for some of the sensors that started collecting data by August 14, 2018. The M&V instrumentation was installed on July 23–24, 2018 for the SCE home and July 25–26, 2018 for the SDG&E home. Accordingly, data analysis for the host sites was officially logged from: August 17th at PG&E site; July 26th for SCE site; July 27th for SDG&E site.

All three host sites had the same instrumentation and sensors installed, with variations in number of sensors that depended on the number of registers available. The specific sensors and configuration of the devices for each host site is provided in the M&V plan schematics in Appendix G, which also includes the instrumentation device product/model/brand.

Each of the homeowners completed an EPRI Customer Consent Form to participate in the research study, allowing the EPRI team to collect data on the installed Next-Gen RSCS units. The homeowners were provided the units' system manual, along with a log book to capture their feedback and any nuances throughout the testing period.

Installed Next-Gen RSCS components are:

- DZ20VC with gas dual fuel
    - Indoor Gas furnaces 120Vac (single phase): DM97MC1005CNAA
    - Outdoor unit 240Vac (single phase): DZ20VC0481AB or DZ20VC0481BB
- Honeywell CTK04 Thermostat: Communicate through ComfortNet, not Wi-Fi.
- EWC control board for zonal control

The following data was requested from each of the homeowners:

- Existing HVAC system specifications (AC and gas furnace): model name, size, any maintenance needs/changes, etc.
- Prior year 2017-2018 utility bills (electric and gas consumption)
- Feedback that captures both energy and non-energy aspects of the Next-Gen RSCS system operation, such as: comfort, ease of operability, drafts, noise, security (as dual fuel system), anomalies in operation, home changes.

Field Evaluation Data Analysis and Results

For each sensor, data is recorded in one-minute intervals, saved on the Acquisuite data acquisition box then transferred via cell modem to the secure EPRI server database. The EPRI team can query and download the data using a variety of programs, namely MATLAB and Tableau. The following summarizes the data scrubbing, analysis, and results for the field evaluation at each of the three host sites, designated as PG&E, SCE, and SDG&E sites.

Data Filtering and Analysis Setup

The data presented in the analysis has been filtered to clearly show the key findings based on the following settings, which is explained below:

1. Quasi-steady state (10 min)
2. Wrong data (NaN and 0)
3. CFM (threshold for readings, < 1800)
4. Temperature (Cooling > 65, Heating < 65)

First, the data is filtered to consider the operation of both the indoor and outdoor unit. A minimum threshold filter, 0.2kW is applied to the power consumption for each of the indoor and outdoor unit. This threshold is roughly the minimum power consumption of the units and eliminates noise from the power consumption data.

The data is also filtered to only consider steady state operation. This is defined as the time when unit has been in operation for over 10 minutes, thus and the first 10 minutes of data when the unit is switched on are not considered. Data with faulty readings (NaN or 0) are also filtered out. Furthermore, the pressure data from pitot tubes show a lot of noise. Thus, a maximum threshold of 1800 CFM per site is used to filter out any readings above that flow rate. This is determined based on the maximum flow rate expected in a typical residential system of 450 CFM/RT. Since all 3 host sites have 4-ton units installed, this translates to 1800 CFM. Lastly, a filter for the outdoor ambient dry bulb temperature is used to separate the heating and cooling season. The heating season is defined as the days where the ambient temperature is below 65^{0}F, while the cooling season data is defined as the days when the ambient temperature is above 65^{0}F.

The air flow rate of the supply plenum is calculated using a flow map which correlates the air flow rate at the return grille to the pressure differential reading at the supply plenum. The return grille flow rate is used in the flow map because it can be measured much easier than the supply plenum and because some supply grilles have very low flow rates that make it difficult to correlate in a flow map. The supply air flow rate is assumed to be the same as the return air flow rate by considering a control volume of the entire indoor space, so the mass into and out of the control volume are balanced.

$$\dot{\mathrm{M}}_{\mathrm{air}}\left[\frac{\mathrm{kg}}{\mathrm{s}}\right] = \dot{\mathrm{m}}_{\mathrm{return,big}}\left[\frac{\mathrm{kg}}{\mathrm{s}}\right] + \dot{\mathrm{m}}_{\mathrm{return,small}}\left[\frac{\mathrm{kg}}{\mathrm{s}}\right] \qquad \textbf{Equation 47}$$

$$\dot{\mathrm{m}}_{\mathrm{return,big}}\left[\frac{\mathrm{kg}}{\mathrm{s}}\right] = 123.17(\Delta\mathrm{P}[\mathrm{kPa}] * 1000\,)^{0.66} * \frac{1.22}{60*35.31}\left[\frac{\mathrm{cfm}}{\frac{\mathrm{kg}}{\mathrm{s}}}\right] \qquad \textbf{Equation 48}$$

[8] $$\dot{m}_{return,small}\left[\frac{kg}{s}\right] = 46(\Delta P[kPa] * 1000\,)^{0.57} * \frac{1.22}{60*35.31}\left[\frac{cfm}{\frac{kg}{s}}\right]$$ **Equation 49**

Experimental Errors and Error Analysis

The accuracy of the data collected is dependent on the accuracy and resolution of the different instrumentation used for measurement and verification. Table 41 summarizes the error source for each measurement and corresponding precision/error. These values are then incorporated into equations that capture the propagation of errors using a quadrature function.[9]

Table 41: Measurement and Instrumentation Error Sources and Values

Measurement	Error source	Sensor/Device Model number	Error
AirFlow	Pressure differential transducer with pitot tube sensor	Dwyer PAFS/616KD-B; 1in W.C. Pitot tube and 1% transducer	For model 616KD-B: ±1% FS[1] FS (full-scale) = 1in w.c.
	Flow Mapping – duct blaster	3% of cfm reading	
Temperature	Monnit Alta Wireless Temp sensor[2] thermistor)	MNS2-9-W2-TS-ST	±1% (1° C or 1.8° F)
Relative Humidity	Monnit Alta Wireless RH sensor[3] (by thermostats) In duct (industrial)[4]	MNS2-9-W2-HU-RH MNS2-9-IN-HU-RH	±3% under normal conditions (10%–90% RH)
Power		W2-M1-mA[5] CT MRS-075	0.1% typical; 0.2% max
Gas flow	Pulse output gas flow meter	Model# BK250 w pulse output 1p/Cf	

1 http://www.dwyer – inst.com/Product/Pressure/DifferentialPressure/Transmitters/Series616KD – specshttp://www.dwyer-inst.com/Product/Pressure/DifferentialPressure/Transmitters/Series616KD#specs

2 https://www.monnit.com/Product/MNS2 – 9 – W2 – TS – SThttps://www.monnit.com/Product/MNS2-9-W2-TS-ST

3 https://www.monnit.com/Product/MNS2 – 9 – W2 – HU – RHhttps://www.monnit.com/Product/MNS2-9-W2-HU-RH

4 https://www.monnit.com/Product/MNS2 – 9 – IN – HU – RHhttps://www.monnit.com/Product/MNS2-9-IN-HU-RH

5 http://www.elkor.net/pdfs/WattsOn – Mark_II_Manual.pdfhttp://www.elkor.net/pdfs/WattsOn-Mark_II_Manual.pdf **[Pg 6, Table 2. Specifications under Accuracy of Apparent Power (VA)]**

Source: EPRI

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[8] The correlation uses two coefficients (a,b) from fitting testing data to a power function (y=a(ΔP)^b). The pressure differential data is given in kPa and is converted into Pa by multiplying 1000. The flow map correlates this pressure reading into air flow rate in CFM, which is then converted into a mass flow rate kg/s by multiplying it with air density and the corresponding dimensional analysis (1.22/(60*35.31)) cfm/(kg/s).

[9] https://reference.wolfram.com/applications/eda/ExperimentalErrorsAndErrorAnalysis.html.

The measurements are used to calculate the unit capacity (Equation 50) and efficiency (as COP (Equation 51). Thus, the corresponding errors are calculated using the following equations and illustrated as error bars in the resulting figures for host site.

$$\mathrm{Error}_{\mathrm{COP}} = \mathrm{COP}\sqrt{\left(\frac{\varepsilon\mathrm{rror}_{\mathrm{Capacity}\ \dot{Q}}}{\mathrm{Capacity}\ \dot{Q}}\right)^2 + \left(\frac{\varepsilon\mathrm{rror}_{\mathrm{P_{HVAC}}}}{\mathrm{P_{HVAC}}}\right)^2} \qquad \textbf{Equation 50}$$

where

$$\varepsilon\mathrm{rror}_{\mathrm{P_{HVAC}}} = 0.1\%\mathrm{P_{Indoor}} + 0.1\%\mathrm{P_{Outdoor}} \qquad \textbf{Equation 51}$$

$$\varepsilon\mathrm{rror}_{\mathrm{Capacity}\ \dot{Q}} = \mathrm{Capacity}\ \dot{Q}\sqrt{\left(\frac{\varepsilon\mathrm{rror}_{\dot{m}}}{\dot{m}}\right)^2 + \left(\frac{\varepsilon\mathrm{rror}_{\mathrm{h_{supply}-h_{return}}}}{\mathrm{h_{supply}-h_{return}}}\right)^2} \qquad \textbf{Equation 52}$$

$$\varepsilon\mathrm{rror}_{\mathrm{h_{supply}-h_{return}}} = 3\%\ (\mathrm{h_{supply}-h_{return}}) \qquad \textbf{Equation 53}$$

$$\varepsilon\mathrm{rror}_{\dot{m}} = \dot{m} * (\text{Exponent as shown in Eqn 48 or 49}) * \frac{\varepsilon\mathrm{rror}_{\Delta\mathrm{P}}}{\Delta\mathrm{P}} \qquad \textbf{Equation 54}$$

Variable Capacity Heat Pump Performance

Cooling Analysis

The cooling season analysis for the 3 sites contains data starting from installation (July/August) in 2018 through Apr 2019. Therefore, the majority of the data obtained for cooling are between August and October in the 3 sites and may not be representative of the summer cooling performance.

While the CEC project ends Apr 30, 2019, EPRI will continue collecting data at the three host sites through Fall 2019 and provide an EPRI Technical Update, augmenting this report with field data analysis for one year. The Technical Update will thus capture the full cooling season more effectively.

The calculations for COP and power consumption are outlined below, where $\dot{Q}_{cooling}$ is the cooling provided by the system in kW, $\dot{m}_{air}$ is the air flow rate in kg/s, h_{return} is the return air enthalpy in kJ/kg, h_{supply} is the supply air enthalpy in kJ/kg, $\dot{m}_{return,big}$ is the air flow rate at the bigger return grille, $\dot{m}_{return,small}$ is the air flow rate at the smaller return grille, ΔP is the pressure differential reading in kPa, $P_{\_HVAC}$ is the total power consumption of the equipment in, kW, which is separated into indoor and outdoor portions. Please note the flow map in Equations 48 and 49, which correlates the pressure differential reading to cfm, which is then converted to kg/s.

$$\dot{Q}_{\mathrm{cooling}}[\mathrm{kW}] = \dot{m}_{\mathrm{air}}\left[\frac{\mathrm{kg}}{\mathrm{s}}\right]\left(\mathrm{h_{return}} - \mathrm{h_{supply}}\right)\left[\frac{\mathrm{kJ}}{\mathrm{kg}}\right] \qquad \textbf{Equation 55}$$

$$\mathrm{COP}\ [-] = \frac{\dot{Q}_{\mathrm{cooling}}[\mathrm{kW}]}{\mathrm{P_{HVAC}}\ [\mathrm{kW}]} \qquad \textbf{Equation 56}$$

$$\mathrm{EER}\ \left[\frac{\mathrm{BTU}}{\mathrm{Wh}}\right] = \mathrm{COP}\frac{3.412[\mathrm{BTU/h}]}{1[\mathrm{W}]} \qquad \textbf{Equation 57}$$

$$\mathrm{P_{HVAC}}[\mathrm{kW}] = \mathrm{P_{outdoor}}[\mathrm{kW}] + \mathrm{P_{indoor}}[\mathrm{kW}] \qquad \textbf{Equation 58}$$

Variable Capacity Heat Pump System Data

Phase 1 and Phase 2 of this project tested a 2-ton Next-Gen RSCS (variable capacity heat pump by Daikin). Since a 4-ton unit is installed for the Phase 3 field testing, the lab results capacity for the 2-ton system are scaled up by a factor of 2 in order to compare the unit's projected performance from lab data to that of the collected field data. The results are shown in Table 42.

Table 42: Variable Capacity Cooling Data Scaled to 4-ton Unit (2-ton variable capacity heat pump)

Outdoor Temp (°F)	Maximum Capacity (Btu/h)	Maximum Power (kW)	Intermediate Capacity (Btu/h)	Intermediate Power (kW)	Minimum Capacity (Btu/h)	Minimum Power (kW)
65	51,889	2.79	28,033	0.91	15,325	0.46
70	50,337	2.99	27,094	1.01	14,878	0.52
75	48,785	3.18	26,156	1.12	14,431	0.58
80	47,232	3.38	25,217	1.22	13,984	0.64
85	45,680	3.58	24,278	1.33	13,538	0.69
90	44,128	3.78	23,339	1.43	13,091	0.75
95	42,575	3.98	22,401	1.53	12,644	0.81
100	41,023	4.18	21,462	1.64	12,197	0.87
105	39,471	4.37	20,523	1.74	11,751	0.93

Source: EPRI

The Next-Gen RSCS field data collected from the variable capacity unit is compared to the scaled lab data for the Next-Gen RSCS. One metric used for comparison is the seasonal energy consumption for cooling. For the field data, this is evaluated by summing the energy used for each 5°F temperature bin. For the scaled lab data, the seasonal energy consumption is estimated analytically in a similar fashion to SEER calculations, where the unit's power consumption for each temperature bin is estimated through the unit's capacity and an assumed load. The assumed load used in this analysis is the field unit's capacity at each of the temperature bins for each host site, since there is no direct measurement on the cooling load of the host sites. The variable speed equipment is able to adjust its capacity to match or approach the cooling load by changing the compressor speed. This is bound by the maximum and minimum stage capacities of the unit in each bin, so if the load is below the unit's minimum capacity, the unit will operate at its minimum capacity, and vice versa for its maximum stage. The variable capacity heat pump's (VCHP's) power consumption is then calculated by interpolating the scaled data using the following equation if the capacity is between maximum and intermediate stage,

$$P_{actual} = \frac{(Q_{actual} - Q_{int})}{(Q_{max} - Q_{int})}(P_{max} - P_{int}) + P_{int} \qquad \textbf{Equation 59}$$

If the capacity is between intermediate and minimum stage, then the equation becomes,

$$P_{actual} = \frac{(Q_{load} - Q_{min})}{(Q_{int} - Q_{min})}(P_{int} - P_{min}) + P_{min} \qquad \textbf{Equation 60}$$

Where P_{actual} is the power consumption corresponding to the load, Q_{actual} is the cooling capacity of the field unit, and the subscripts max, int, and min, corresponds to the scaled VCHP performance data.

Based on the assumed load and the unit's capacity, a cyclic factor is calculated using

$$\gamma = 1 - 0.12(1 - \frac{Q_{load}}{Q_{actual}}) \quad \textbf{Equation 61}$$

Where γ is the cyclic factor[10] with a maximum value of 1, and Q_{load} is the assumed load (equal to the capacity of the field unit).

The cyclic factor is used to calculate an adjusted capacity, which is in turn used to calculate the operating time of the unit. The operating time reflects the duty cycle of the unit and captures the cycling losses of the unit.

$$Q_{adjusted} = \gamma Q_{actual} \quad \textbf{Equation 62}$$

$$t_{op} = \frac{Q_{load}}{Q_{adjusted}} \quad \textbf{Equation 63}$$

The energy consumption in each temperature bin is then calculated using the total operating time of the field variable capacity unit in each temperature bin,

$$E = P_{actual} t_{op} h \quad \textbf{Equation 64}$$

Thus, the seasonal energy consumption is simply the sum of the energy consumption in each temperature bin.

The cooling seasonal energy consumption analysis is shown for the PG&E and SDG&E site (Figure 128). The SCE host site is omitted from this analysis due to an equipment issue that was identified by EPRI and the designated Daikin Certified Professional (DCP) subcontractor using the systems' FDD messages, thus presenting an anomaly to the 2018 SCE host site cooling season data[11].

Figure 128 compares the Next-Gen RSCS of the variable speed heat pump lab scaled data (from 2-ton to 4-ton system) with the field data (4-ton system). The comparison between the variable speed lab data and field data demonstrates how the equipment performs in a well-controlled environment (lab testing), and how its performance changes when implemented in a real-world operating environment (field evaluation). Since a field evaluation of a single speed system is beyond the scope of this work, there is no comparison between the field data with a single speed system. Future work can investigate the performance of single and variable speed equipment in the field to conduct a more conclusive comparison.

The field equipment outperformed the lab model at the PG&E site by 12 percent, while the opposite occurred at the SDG&E site by 18 percent. Typically, lab-based testing outperforms

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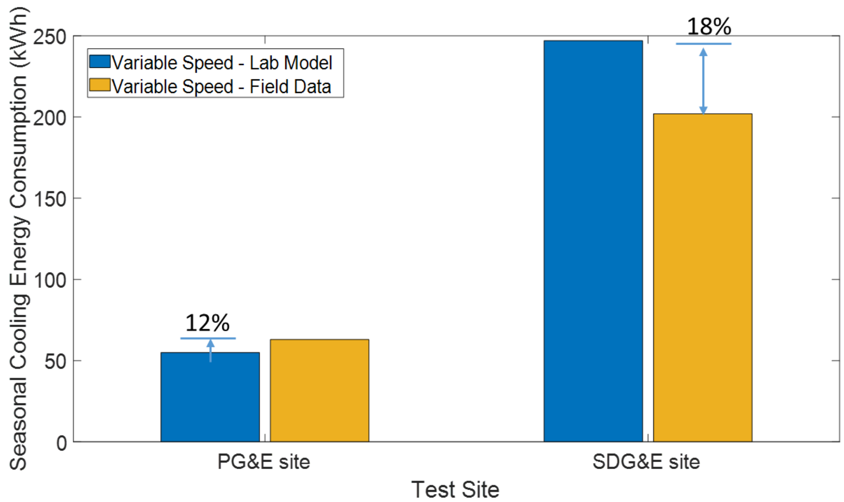
[10] For cyclic degradation factor equation, 0.12 is specified based on AHRI 210/240.

[11] Since the SCE host site unit troubleshooting has been completed, the upcoming EPRI Technical Update will include the updated cooling season 2019 data for all 3 sites.

field evaluation since the lab tests simulate a more controlled environment and operating conditions than field tested equipment. The discrepancy at the SDG&E site can be caused by different indoor thermostat setpoints, as well as sources of error (as summarized in the Error Analysis section of 4.4.1). A 1°F change in thermostat setpoint can cause up to 7 percent change in the system's power consumption, therefore, if the performance discrepancies can be caused by differences between the indoor temperature setpoint during Phase 1 and 2 lab testing and the thermostat setpoints at the host sites.

Figure 128: Cooling Results Comparison (Aug – Oct 2018) at Two Host Sites for Variable Speed Field vs Scaled Lab Data



Source: EPRI

Additional plots of the field data, illustrating the equipment performance at the PG&E and SDG&E host sites are provided in Appendix G (see Figure G-1 through Source: EPRI

Figure G-12). The figures capture the equipment's capacity, performance, EER and COP, as function of the outdoor temperature.

The variable capacity unit's operation can be grouped into three modes, high, intermediate, and low stage. These are defined similar to Phase 1 lab testing: high stage is above 50 percent of unit capacity, intermediate is between 50 percent and 30 percent, and low stage is below 30 percent. The unit capacity is calculated as a percent of total capacity using the correlation shown below, where x is the percent capacity of the system, and T_{amb} is the outdoor ambient temperature in °F.

$$x = 0.08 + 0.86P_{HVAC} - 0.01T_{amb} - 0.13P_{HVAC}^2 + 0.001P_{HVAC}T_{amb} + 3x10^{-5}T_{amb}^2$$

Equation 65

The cooling capacity provided to the house is calculated using humid air properties and the flow rate of the supply plenum. This is then used to calculate the COP and EER of the system, and the results are shown with respect to the temperatures bin and operation modes.

The unit's power consumption shows a clear trend as the ambient temperature changes. As the ambient temperature increases, the system has to condense at a higher temperature which increases the amount of work done by the unit. Both capacity and power consumption data from the field evaluation agree with the scaled-up trends from the lab testing. The COP increases as the unit stages down because the capacity is decreasing while the heat exchanger sizes stay the same. With relatively larger heat exchangers, the heat transfer effectiveness is higher, thus COP is also higher.

Heating Analysis

The heating season analysis was performed in a similar fashion as the cooling season. While the system has dual fuel heating capabilities, this section only presents the analysis on the heat pump heating performance while the furnace analysis will be included in the dual fuel section. The heating capacity of the unit is calculated using Equation 66.

$$\dot{Q}_{heating}[kW] = \dot{m}_{air}\left[\frac{kg}{s}\right]\left(h_{supply} - h_{return}\right)\left[\frac{kJ}{kg}\right] \quad \textbf{Equation 66}$$

The performance of the 2-ton Next-Gen RSCS from lab testing is also scaled up and presented below in Table 43.

Table 43: Scaled Heating Data of 2-Ton to 4-Ton Lab Tested Single Speed Heat Pump (8.2 HSPF)

Outdoor Temp (F)	Capacity (Btu/h)	Power (kW)
30	30,949	3.39
35	34,033	3.44
40	37,117	3.50
45	40,201	3.56
50	43,285	3.62
55	46,369	3.68
60	49,453	3.73
65	49,453	3.73

Source: EPRI

Table 44: Scaled Heating Data from 4-Ton to 2-Ton Lab Tested Variable Capacity Heat Pump (Next-Gen RSCS)

	Maximum		Intermediate		Minimum	
Outdoor Temp (F)	Capacity (Btu/h)	Power (kW)	Capacity (Btu/h)	Power (kW)	Capacity (Btu/h)	Power (kW)
30	35,920	3.72	17,338	1.48	7,248	0.96
35	39,402	3.80	19,705	1.53	8,475	0.90
40	42,883	3.87	22,073	1.58	9,702	0.86
45	46,365	3.93	24,441	1.62	10,930	0.83
50	49,847	3.98	26,808	1.66	12,157	0.81
55	53,328	4.03	29,176	1.69	13,384	0.79
60	56,810	4.07	31,544	1.72	14,611	0.78

	Maximum		Intermediate		Minimum	
65	60,291	4.14	33,911	1.74	15,839	0.77

Source: EPRI

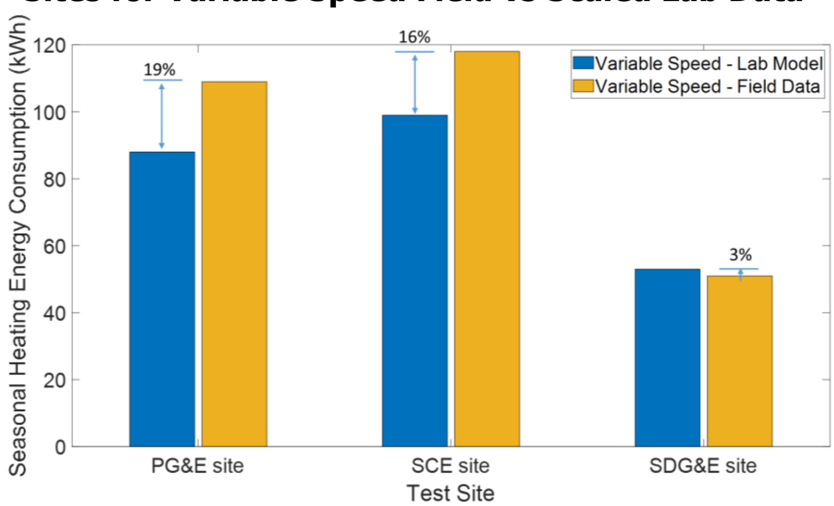
The heating performance of both the lab and field tested Next-Gen RSCS is estimated using the same methodology as the cooling season analysis. Figure 129 summarizes the variable speed lab-based model versus field test data for heating mode at the 3 host sites. For the heating season, the SCE host site data is included in the analysis because the equipment issue did not affect the system's performance as much as the cooling analysis.

The comparison between variable capacity lab model and field equipment (Figure 129) shows a similar trend to the cooling season analysis. Both the PG&E and SCE host site show a 15—20 percent reduction in energy consumption from the field equipment to the lab model. Once again, the SDG&E host site show lower energy consumption in the field equipment than the lab model, which is likely due to the same setpoint factors as described before.

Additional plots of the heating season field data, illustrating the equipment performance at the PG&E and SDG&E host sites, are provided in Appendix G (see Figure G-1 through Source: EPRI

Figure G-12). The figures capture the equipment's capacity, performance, and COP, as function of the outdoor ambient temperature bins.

Figure 129: Comparison of Heating Results (Nov 2018 – Feb 2019) at Two Host Sites for Variable Speed Field vs Scaled Lab Data



Source: EPRI

Dual Fuel (Intelligent Heating) Performance

The cost comparison between heat pump and furnace operation was evaluated in a normalized way (cents/therm) for simple comparisons. The furnace operation cost is assumed constant with outdoor temperature while the heat pump operation cost is a function of the outdoor temperature. The analysis assumed a furnace AFUE of 0.97, which is the same rating for the furnace installed at the host sites. The heat pump operation cost is calculated using

$\mathrm{C_{Furnace}}\left[\frac{¢}{\mathrm{therm}}\right] = \frac{\mathrm{R_{NG}}[\frac{¢}{\mathrm{therm}}]}{\mathrm{AFUE}}$ **Equation 67** to equation $\gamma[-] = \frac{Q_{heating}}{Q_{HP}}$ **Equation 70.**

$$\mathrm{C_{Furnace}}\left[\frac{¢}{\mathrm{therm}}\right] = \frac{\mathrm{R_{NG}}[\frac{¢}{\mathrm{therm}}]}{\mathrm{AFUE}}$$ **Equation 67**

$$\mathrm{C_{HP}}\left[\frac{¢}{\mathrm{therm}}\right] = \frac{\mathrm{R_e}[\frac{¢}{\mathrm{therm}}]}{\mathrm{COP}[-]}$$ **Equation 68**

$$COP[-] = -2.28 + 8.25\gamma + 0.12T - 4.31\gamma^2 - 0.08\gamma T$$ **Equation 69**

$$\gamma[-] = \frac{Q_{heating}}{Q_{HP}}$$ **Equation 70**

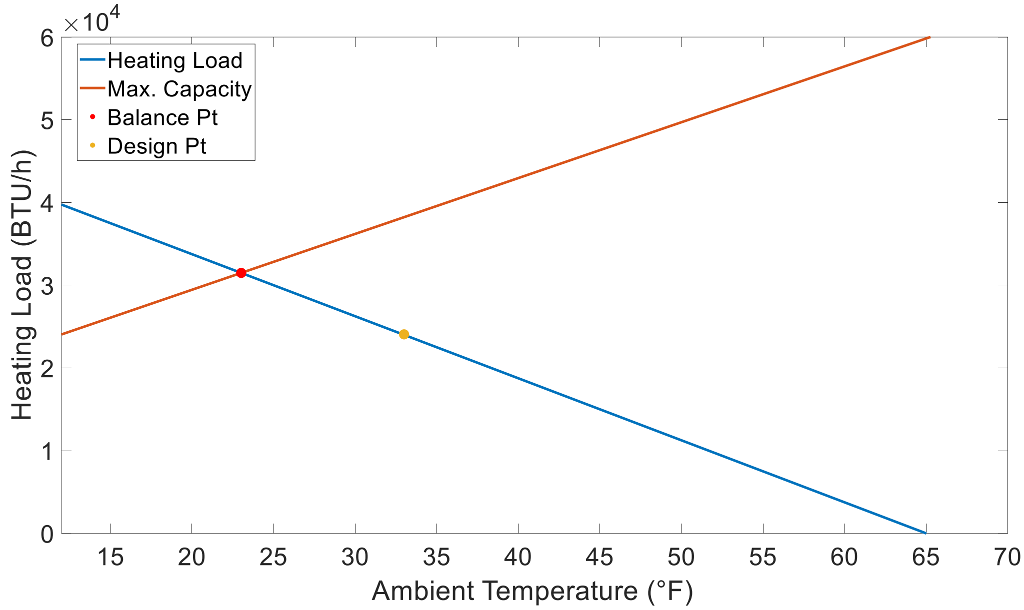
Where C is the normalized cost (cents/therm), R is the cost of natural gas or electricity cents/therm), γ is the ratio of the heating load $Q_{heating}$ to the heat pump maximum capacity Q_{HP}, and T is the ambient dry bulb temperature (°F). Lab testing data for the 2-ton Next-Gen RSCS unit is scaled up to develop the COP correlation in Equation 64, using the heating performance data shown in Table 42.

Figure 130 shows the balance point calculation for the 3 host sites. The heating load design point was 33°F (1°C) for all 3 sites from a load calculation performed at the PG&E host site and using the industry standard of no heating requirements at 65°F (18°C). By assuming a linear relationship between outdoor ambient temperature and the heating requirement of the space, a heating load line is calculated. The intersection between this heating load line and the maximum capacity of the unit is the balance point of the unit, at 23°F (–5°C), which refers to the lowest temperature at which the heat pump can meet all heating load of the space without another electric or natural gas backup heating.

While the Next-Gen RSCS can satisfy over 90 percent of the heating loads for the majority of California's 16 climate zones (as evaluated in Phase 1 of the project (see Chapter 2), using the electric heat pump alone the dual fuel functionality enables greater heating capacity range for the extreme cases/geographies. Technically, California residents would only need a back-up to their heat pump for a small fraction of the time. Utility rates can greatly impact the choice of fuel for residential heating. The dual source is not just for backup; the furnace can displace the electric heat pump when it is economical to do so.

The breakeven temperature is a common metric used to compare the economics of operating a heat pump and furnace. While the heat pump operating cost is driven by the ambient temperature and the price of electricity, the furnace operating cost is driven by the price of natural gas.

Figure 130: Heat Pump Balance Point Calculation Based on Scaled Lab Testing Data



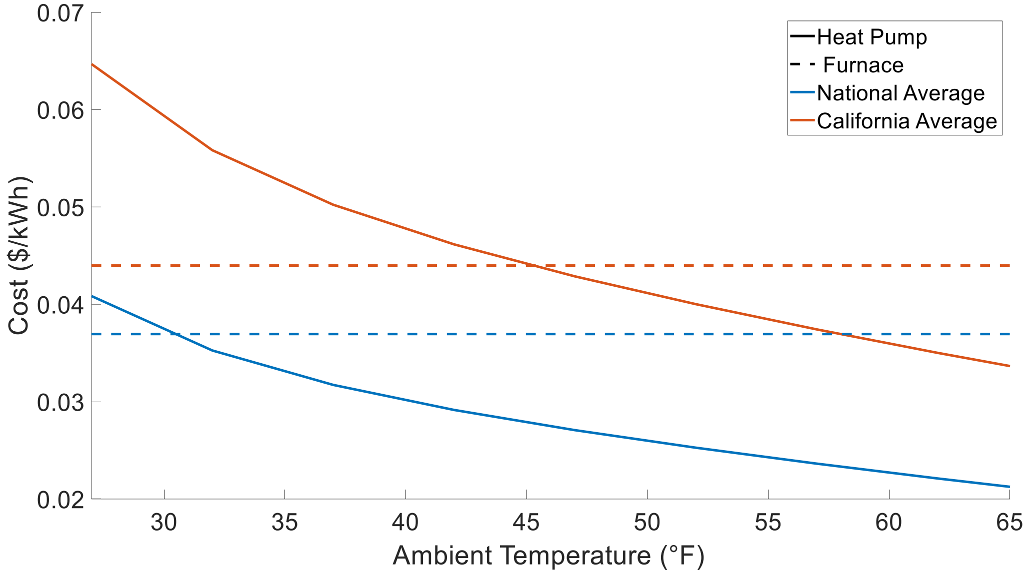
Source: EPRI

Figure 131 illustrates the cost analysis for electric variable capacity heat pump (Next-Gen RSCS) and furnace operation based on the average national electric and gas rates[12] and more expensive and cheaper prices in the United States (Table 45) to determine the breakeven temperature for different combinations of electricity and natural gas prices. The solid lines are the variable capacity heat pump cost and the dotted lines are the furnace costs.

For example, average prices in California is $0.19/kWh for electricity and $1.25/therm for natural gas, meaning the price of electricity is well above national average, while the natural gas prices are close to the national average. The breakeven temperature for national average prices is roughly 31°F (–1°C), which is lower when compared to California average which is roughly 45°F (7°C). For California, a significant portion of the heating season experiences temperatures between 45°F (7°C) and 65°F (18°C) (no heating load), so heat pump will be the more economic choice for those hours. For residential customers, the operating cost will be the main influence in choosing the heating source. Thus, utility rates are primary drivers for incentivizing the usage of heat pumps.

Figure 131: Cost Analysis for Heat Pump and Furnace Operation Based on National Average Prices and Variations for Electricity and Gas

[12] Sources: https://www.eia.gov/electricity/monthly/epm_table_grapher.php?t=epmt_5_6_a; https://www.eia.gov/dnav/ng/ng_pri_sum_dcu_nus_m.htm.



Heat Pump
Furnace
National Average
California Average
Cost ($/kWh)
0.07
0.06
0.05
0.04
0.03
0.02
30
35
40
45
50
55
60
65
Ambient Temperature (°F)

Source: EPRI

Table 45. National Average Electricity and Gas Prices*

	National Average Price*	California Average Price
Electricity ($/kWh)	0.12	0.19
Gas ($/therm)	1.05	1.25

***Hawaii not included**

Source: EPRI

Auto Demand Response Performance

The demand response (DR) field test required the use of the manufacturer's software program to adjust the outdoor unit's input load. The manufacturer's smart thermostat that is DR-enabled will be commercially launched in 2019, which is beyond the timeline of this project's execution and report deadline. Thus, the DR field test was only conducted at the PG&E host site since the homeowner has been part of the laboratory evaluation and familiar with the product's operation. Additionally, the PG&E host site was selected for conducting further field experiments for the zonal control.

The following guidelines were provided to conduct field demand response (DR) test:

a. Plan for 10min DR event
b. For a specific DR event, input "payload value" – meaning an upper limit for the unit operation between 0 percent and 100 percent of the power load of the system (not operation level). [Using RAM monitor software program from manufacturer]
c. Specify DR start time and end time and payload value. Conduct the following payload value tests with a 30min interval in between tests to allow for steady-state.
    - 30 percent (reduce power down to or close to 30 percent of system power)
d. 50 percent (reduce power down to or close to 50 percent of system power)

Testing procedure:

a. Set thermostats in all zones to the same setpoint (ex. 60^{o}F). Please monitor the return temperature before and after each DR event.
b. After 30min, begin DR test 1. Set payload value to 30 percent. Run DR event 1 for 10min.
c. Wait 30min to begin DR test 2 if running tests on the same day. Note that waiting time in between tests depends on return temperature value. Ensure return temperature is steady-state and close to the return temperature at the start of testing, before DR test 1. Set payload value to 50 percent. Run DR event 2 for 10min.

The demand response field tests were both conducted in the heating mode due to scheduling. Additional demand response tests will be conducted in the cooling mode in the upcoming months, the results of which will be provided in the forthcoming Technical Update, capturing the remainder of the field-testing data.

Demand Response Test Results

On January 23, 2019, a DR test was conducted at one of the sites to reduce power consumption to 30%. The event occurred from 11:30am through 11:44am. Table 46 shows the measured power used and its system capacity at intervals during the event. Table 47 shows the summaries of events that occurred during the event. The total power consumption of the indoor and outdoor units leading up to, during, and after the DR event is illustrated in Figure 132. Figure 133 shows the power consumption of the indoor and outdoor units and their affected output capacity during the DR event, and the total power consumption is shown in Figure 134.

Table 46: Summary Results — 30 Percent Demand Test (Jan 23, 2019)

Reduce from 100% Demand to Close 30% Demand	Power Measurement (Total System Power = IDU + ODU)	Capacity Calculation through Temperature and Pressure Measurements
11:10am	3.26 kW	8.743 kW
11:15am	4.17 kW	14.371 kW
Event Start: 11:30am	4.18 kW	14.393 kW
11:34am	1.655 kW	7.662 kW
Event End: 11:44am	0.906 kW	3.828 kW
Percent Change (from 11:30am to 11:44am)	78.3%	73.4%

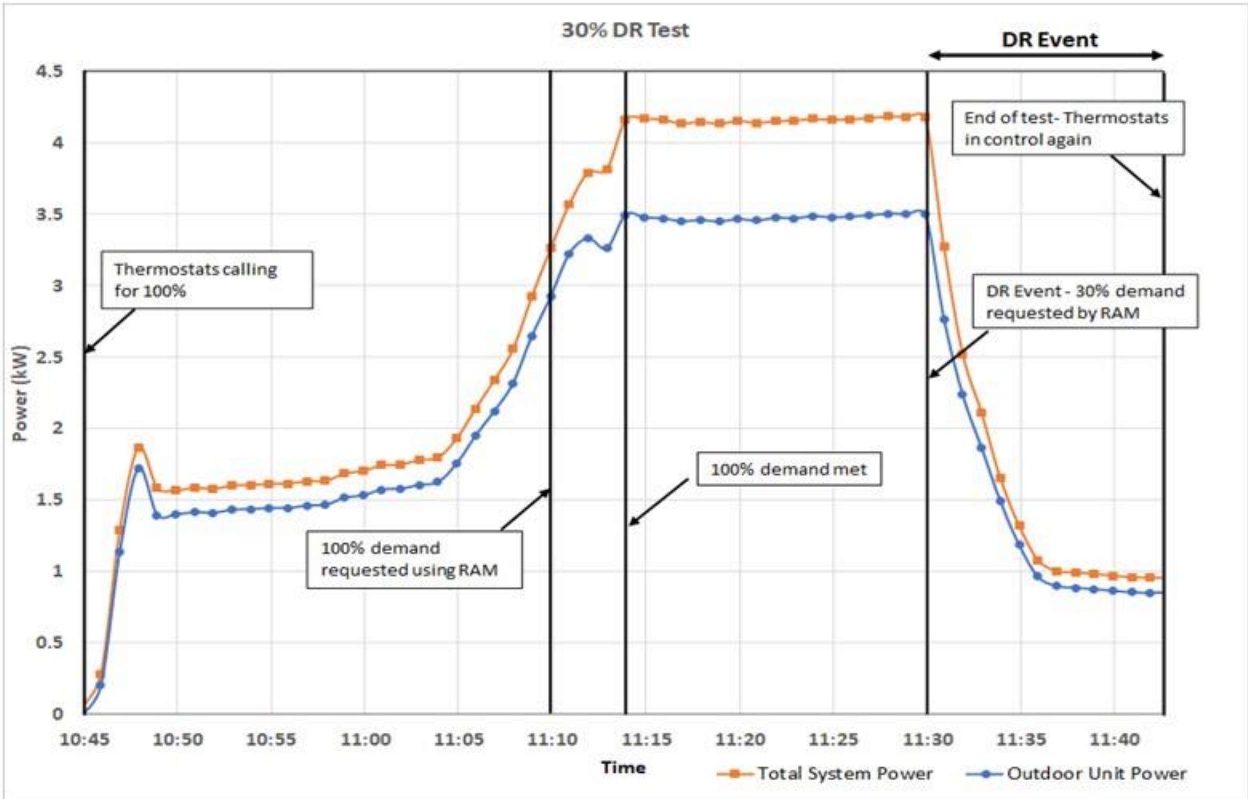
Source: EPRI

Table 47: Test Run-Time and Summary (from Homeowner's Notes)

Timestamp Jan. 23, 2019	Activity	Notes
10:45–10:50am	Turned on all thermostats to start test	CTK04 says request is 100% but actual demand is around 50%
11:10am	Changed demand in ram monitor to 200 (100%)	
11:30am	Changed to 60 (30%) demand	System seemed to respond slowly to 30% demand change
11:34am	System eventually dropped to 30% demand	
11:44am	Ended test	Down back thermostat is calling for 57% demand and ram says system is 33.5%. Other zones should have closed.
11:48am	Reached 57% demand	

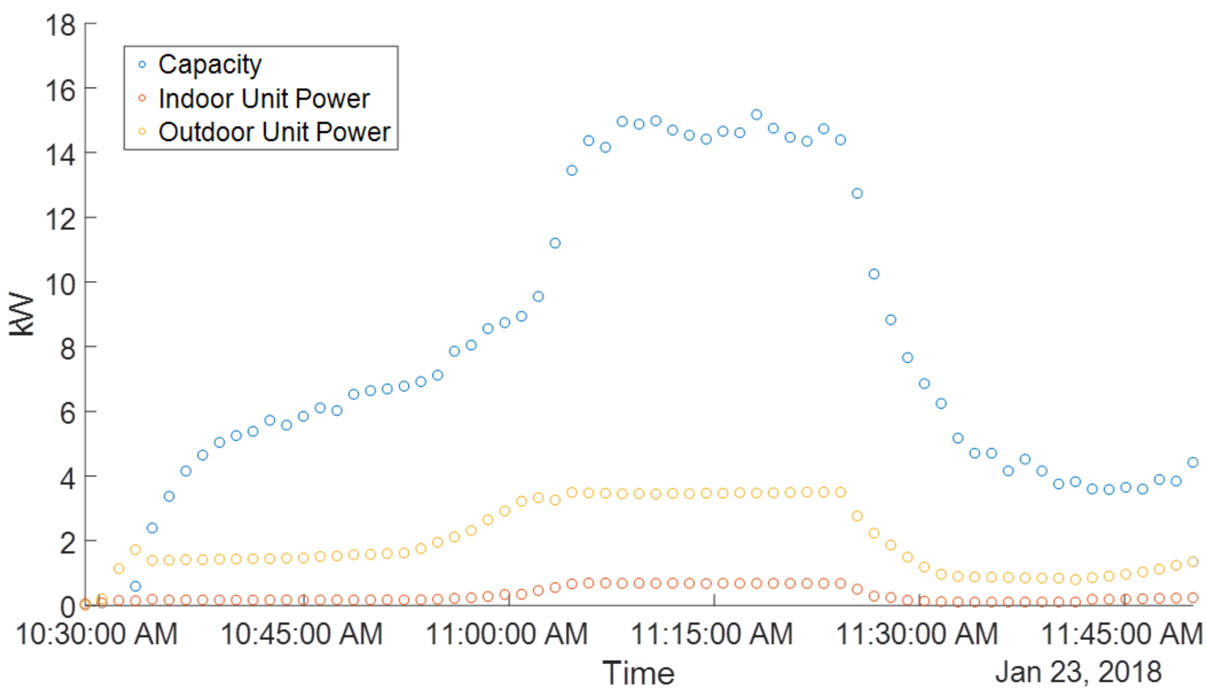
Source: EPRI

Figure 132: Total Outdoor and Indoor Unit Power Consumption with Time



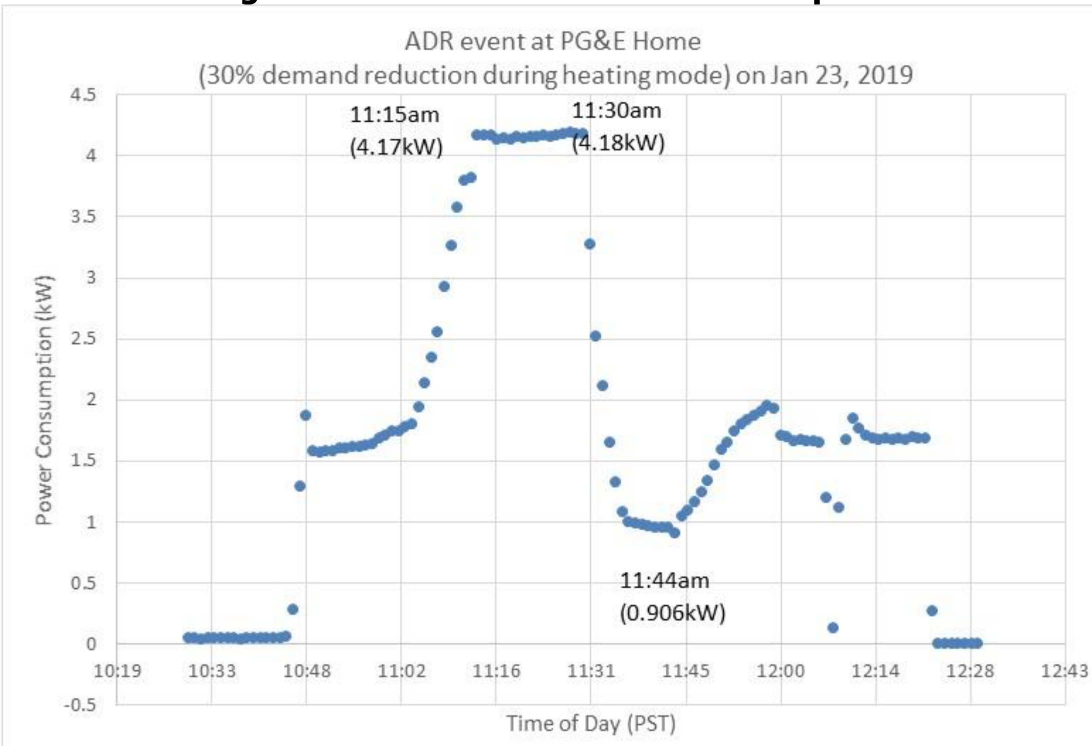
Source: EPRI

Figure 133: Unit Power Consumption and Capacity for Close 30 Percent Load Demand Response Event



Source: EPRI

Figure 134: Total Unit Power Consumption



Source: EPRI

On February 8, 2019, another DR test was conducted in the same house, this time to reduce power consumption by 50%. The event occurred from 11:56am through 12:15pm. Table 48 shows the measured power used and its system capacity at intervals during the event. Table 49 shows the summaries of events that occurred during the event. Figure 135 shows the total power consumption during the DR event.

Table 48: Summary Results: Approach 50 Percent Demand Test (Feb 8, 2019)

Reduce from 100% Demand to 50% Demand	Power Measurement (Total System Power = IDU + ODU)	Capacity Calculation through Temperature and Pressure Measurements
11:17am		3.864
11:38am		9.877
11:41am	4.071 kW	13.056
Event Start: 11:56am	4.162 kW	14.362
11:59am	2.126 kW	8.843
12:00 pm	1.451 kW	7.342
12:05pm	1.349 kW	6.198
Event End: 12:15pm	1.34 kW	5.335
12:18pm	1.663 kW	5.603
% Change (from 11:56am to 12:15pm)	67.8%	62.9%

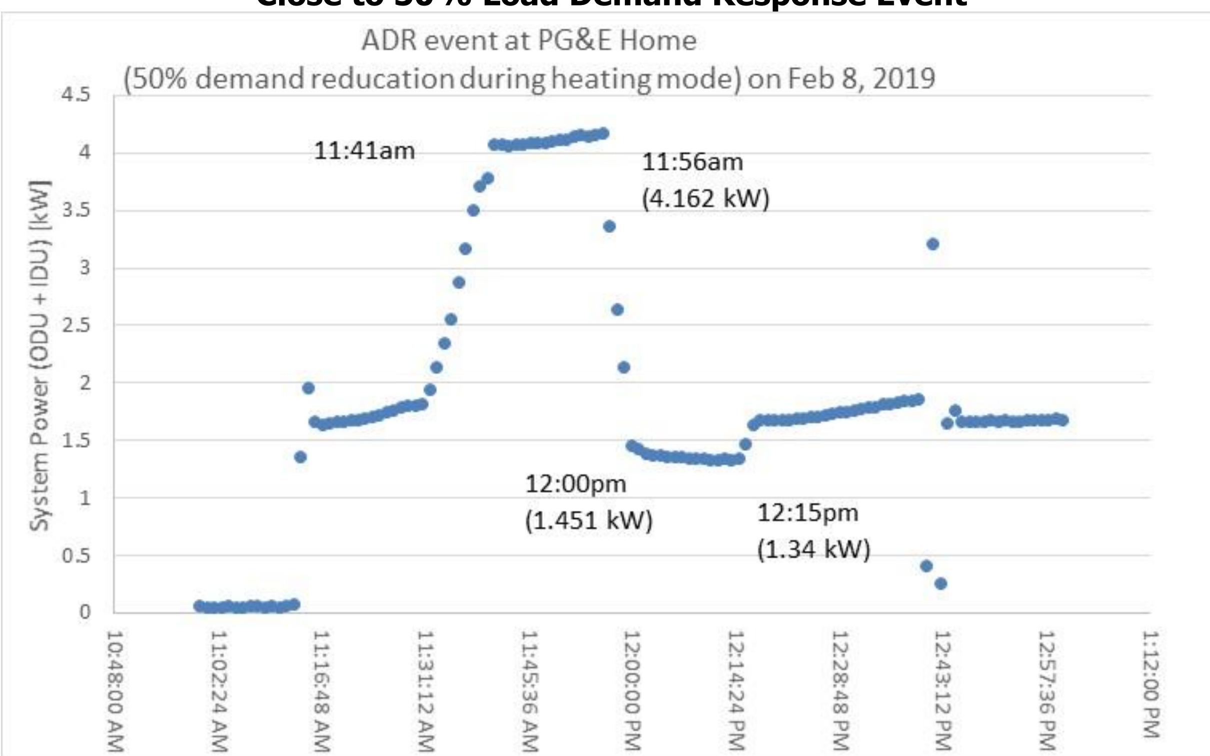
Source: EPRI

Table 49: Test Run-Time and Summary (from Homeowner's Notes)

Timestamp Feb. 8, 2019	Activity	Notes
11:15am	Turned on all thermostats	Outdoor Unit Ambient Temp = 50F
11:17am	Changed demand in ram monitor to 200 (100%)	
11:20am	Still running at 55% demand but request shows 100%	11:25am Tstats say house is 64, 64, 65F. 11:31am still at 55% demand 11:33am at 59% demand. Seems to ramp up slowly 11:34am 67% 11:35am 70% 11:36am 75.5%
11:38am	Demand at 100%	
11:56am	DR event start – reduced demand in RAM monitor to 50%	11:57 83.5% demand 11:58 70% demand
11:59am	50% demand reached	

Source: EPRI

Figure 135: Total Unit Power Consumption During Close to 50% Load Demand Response Event



Source: EPRI

As summarized in Table 50, both the field DR events, tested on different dates, resulted in 4.9 percent variation in the reductions for the unit capacity and unit power in heating mode. The reduction in system capacity was less than the reduction in unit power, which is consistent with the DR lab testing results. However, the variation in the reductions for unit capacity and unit power in cooling mode for the lab testing was larger, ranging between 8.3 percent and 12.3 percent. In other words, for both lab and field ADR testing, the unit capacity reduction is less than the unit power reduction, thus the customer comfort is not as compromised while energy savings can be realized during a utility-led demand response event.

Table 50: Summary of Field Demand Response Testing

Demand Response Event	Field Test Unit Power	Field Test Unit Capacity	Lab Test Unit Power	Lab Test Unit Capacity
Baseline for 30% load test	4,180 W	14,393 W	1,866 W	17,000 Btu/h
Reduction to 30% load	906 W	3,823 W	558 W	6,500 Btu/h
Percent Change from Baseline	***78.3%***	***73.4%***	***70.1%***	***61.8%***
Variation between Power and Capacity Change	*4.9% variation*		*8.3% variation*	
Baseline for 50% Load Test	4,162 W	14,362 W	1,866 W	17,000 Btu/h
Reduction to 50% Load	1,340 W	5,335 W	928 W	10,500 Btu/h
Percent Change from Baseline	***67.8%***	***62.9%***	***50.3%***	***38.2%***
Variation between Power and Capacity change	*4.9% variation*		*12.3% variation*	

Source: EPRI

Zonal Control and Duct Delivery Effectiveness Performance

One of the primary objectives of the field testing of variable-capacity systems was to study the impact of duct systems located in unconditioned space on overall system performance. Laboratory testing in Phase 1 and 2 of this project identified considerable advantages to using air-side zoning to reduce duct losses when reducing speed (capacity and air flow) of a variable-capacity system. The equipment installed in the field sites for this project all included zoning dampers and controls.

Airflow was monitored by measuring a representative pressure for each duct-zone in the system and mapping the airflows measured using a powered flow hood to the duct-zone pressures. A flow map was generated for each grille. There were some issues generating the flow maps for each grille in the SCE home (damper position appeared to shift during the measurements), so only the total flow was monitored for that site. The duct analysis relies on individual grille flow, so the duct performance measurements described below only include the PG&E and SDG&E host sites.

The PG&E home initially had four similarly sized zones installed to align with the four-zone system tested in the laboratory. Due to challenges with air balancing at the test home, the ducting arrangement was modified to three operable zones (Zones 2, 3, and 4) and one “open” zone (Zone 1). This means that Zone 1 is active whenever any of the other zones are active. The minimum opening for the Zone 1 damper was adjusted during commissioning and set to a constant for the duration of the study. This results in lower flow rate for the grilles in Zone 1 compared to if the damper was left fully open.

Table 51 shows the zoning details for each of the three test sites. The SDG&E and SCE test site had two and three zones, respectively.

Table 51: Zoning Arrangement for Each Test House

	PG&E Host Site	SCE Host Site		SDG&E Host Site	
Zone 1	Entry	Zone 1	Entry	Zone 1	Master Bed
	Dining		Living Room		Master Bath
Zone 2	Kitchen		Family Room		Bedroom 2
	Family Room		Dining		Office
	Guest Room	Zone 2	Upstairs Bath	Zone 2	Hall Bath
	Guest Bath*		Office		Living Room
	Laundry		Bedroom 2		Kitchen
Zone 3	Master Bed		Bedroom 3		Dining Room
	Office*	Zone 3	Master Bed		Entry
	Master Bath*		Master Bath		Family Room
	Master Toilet		Master Toilet		
Zone 4	Bedroom 2	*No grille temperature data*			
	Bedroom 3				
	Upstairs Bath*				
	Upstairs Toilet*				
	Loft				

Source: EPRI

Data Filtering Process and Settings

The duct analysis was performed by comparing the delivered cooling/heating to the home at the individual grilles to the cooling/heating produced by the equipment. As Phase 2 testing showed, the thermal losses through ducts located in unconditioned space are impacted by the compressor/fan speed and the duct-zone temperature. The ducts were located in the attic for all homes monitored so the attic temperature represents the duct-zone temperature.

Several filters were used to avoid analyzing transient behavior of the system during start up and shut down. Individual grille performance was analyzed only when that particular grille was active for at least five minutes and then up to five minutes prior to shutting down. A zone was considered active whenever airflow was moving through the duct and the compressor was operating.

Thermal duct losses only affect the sensible capacity delivered through the duct system. In the cooling season, there is a latent component of the capacity due to drying of the air that is not impacted by conduction losses. In the heating season, only sensible heating is performed by the heat pump so the sensible capacity is considered the total capacity. The impact of duct leakage was not considered for this analysis. HERS testing of the duct system in each home showed a 15 percent leakage rate.

The PG&E home had several grille temperatures that were not recorded during the study due to a communication failure with the temperature probe. In those cases, the appropriate zone-average temperature was used as the temperature for those grilles. The zone-average temperature was calculated based on the average grille temperature in the other grilles in the same zone. In some cases, when a similar grille existed within the same zone (same duct sizing and length), the temperature data for the grille with data was used for the grille that was missing data as a representative grille temperature.

Simplified Characterization of Duct Thermal Losses

Delivery Effectiveness for a particular grille is the ratio of the sensible (or total) cooling or heating delivered at a grille, to the cooling or heating delivered by the equipment to that grille. This effectiveness can be analyzed mathematically as a heat exchanger represented by the duct and its surroundings (the attic). The goal is to have the duct be the least effective heat exchanger possible with respect to the unconditioned space to reduce heat transfer with the attic. The standard equations for heat exchanger performance are:

$$q = \varepsilon \dot{m}_{duct} c_p (T_{in,duct} - T_{attic}) \qquad \textbf{Equation 71}$$

where the heat exchanger effectiveness is defined as:

$$\varepsilon = \left(1 - e^{-\frac{UA}{\dot{m}_{duct} c_p}}\right) = \left(1 - e^{-NTU}\right) \qquad \textbf{Equation 72}$$

where

A is the duct surface area [ft^2]

c_p is specific heat of air [Btu/lb-^{o}F]

$\dot{m}_{duct}$ is the mass flowrate of air through the duct [lb/h]

NTU is number of transfer units (standard heat exchanger parameter) [-]

U is the normalized conductance of the duct wall [Btu/h-ft^2-^{o}F]

$T_{in,duct}$ is the temperature of the air entering the duct [^{o}F]

The temperature entering the duct can be expressed as:

$$T_{in,duct} = T_{ret-plen} + \frac{\dot{Q}_{equip}}{\dot{m}_{duct} c_p} \qquad \textbf{Equation 73}$$

So, substituting Equation 68 into Equation 66, and rearranging:

$$q = \varepsilon \dot{m}_{duct} C_p \left(T_{retplen} + \frac{\dot{Q}_{equip}}{\dot{m}_{duct} c_p} - T_{attic}\right) = \varepsilon \dot{m}_{duct} c_p \left((T_{retplen} - T_{attic}) + \frac{\dot{Q}_{equip}}{\dot{m}_{duct} c_p}\right)$$

$$= \varepsilon \dot{m}_{duct} C_p \left(\Delta T_{retplen-attic} + \frac{\dot{Q}_{equip}}{\dot{m}_{duct} c_p}\right) \qquad \textbf{Equation 74}$$

The temperature difference between the duct and the attic can thus be split into two components, one component that is independent of equipment operation, and another that does depend on equipment operation.

This heat loss can be used to calculate the delivery effectiveness of the supply ductwork on a fractional basis:

$$\eta_{\mathrm{duct}} = \frac{\dot{Q}_{\mathrm{equip}} - q}{\dot{Q}_{\mathrm{equip}}} \qquad \textbf{Equation 75}$$

In the case of sensible heating,

$$\begin{aligned}\dot{Q}_{\mathrm{equip}} &= \dot{m}_{\mathrm{duct}} c_p (T_{\mathrm{sup-plen}} - T_{\mathrm{ret-plen}}) \\ &= \dot{m}_{\mathrm{duct}} c_p (T_{\mathrm{in,duct}} - T_{\mathrm{ret-plen}}) = \dot{m}_{\mathrm{duct}} c_p (\Delta T_{\mathrm{equip}})\end{aligned} \qquad \textbf{Equation 76}$$

Substituting Equation 71 and 69 into Equation 70,

$$\eta_{\mathrm{duct}} = \frac{\dot{m}_{\mathrm{duct}} c_p (\Delta T_{\mathrm{equip}}) - \varepsilon \dot{m}_{\mathrm{duct}} c_p (\Delta T_{\mathrm{retplen-attic}} + \Delta T_{\mathrm{equip}})}{\dot{m}_{\mathrm{duct}} c_p (\Delta T_{\mathrm{equip}})} = \frac{(\Delta T_{\mathrm{equip}}) - \varepsilon (\Delta T_{\mathrm{retplen-attic}} + \Delta T_{\mathrm{equip}})}{(\Delta T_{\mathrm{equip}})}$$

Equation 77

$$\eta_{\mathrm{duct}} = 1 - \varepsilon\left(\frac{\Delta T_{\mathrm{retplen-attic}}}{\Delta T_{\mathrm{equip}}} + 1\right) \textbf{ or } \eta_{\mathrm{duct}} = 1 - \left(1 - e^{-\mathrm{NTU}}\right)\left(\frac{\Delta T_{\mathrm{retplen-attic}}}{\Delta T_{\mathrm{equip}}} + 1\right) \qquad \textbf{Equation 78}$$

Performing the multiplication

$$\begin{aligned}\eta_{\mathrm{duct}} &= 1 - 1 + e^{-\mathrm{NTU}} - \frac{\Delta T_{\mathrm{retplen-attic}}}{\Delta T_{\mathrm{equip}}} + e^{-\mathrm{NTU}} \frac{\Delta T_{\mathrm{retplen-attic}}}{\Delta T_{\mathrm{equip}}} \\ &= e^{-\mathrm{NTU}}\left(1 + \frac{\Delta T_{\mathrm{retplen-attic}}}{\Delta T_{\mathrm{equip}}}\right) - \frac{\Delta T_{\mathrm{retplen-attic}}}{\Delta T_{\mathrm{equip}}} \\ &= e^{-NTU} + \frac{\Delta T_{retplen-attic}}{\Delta T_{equip}}(e^{-NTU} - 1)\end{aligned} \qquad \textbf{Equation 79}$$

Looking at $= e^{-NTU} + \frac{\Delta T_{retplen-attic}}{\Delta T_{equip}}(e^{-NTU} - 1)$ **Equation 79**, it should be noted that duct efficiency always ranges between 0 and 1, which means that the second term in the last version of $= e^{-NTU} + \frac{\Delta T_{retplen-attic}}{\Delta T_{equip}}(e^{-NTU} - 1)$ **Equation 79** will always be negative, will increase in absolute value as the attic gets colder in the winter or warmer in the summer, and will increase in absolute value as the temperature differential across the equipment decreases.

Going back to Equation 67, for a round duct, UA = (1/R)πDL and $\dot{m}c_p = \dot{V}\rho c_p = VA\rho c_p = V\pi D^2 \rho c_p / 4$, so

$$\mathbf{NTU} = \mathbf{UA}/\, \dot{m}c_p = 4L/RVD\rho c_p \qquad \textbf{Equation 80}$$

where:

D	is the duct diameter [ft]
R	is the R-value of the duct wall [h-ft^2-^{o}F/Btu]
V	is the duct velocity [ft/h]

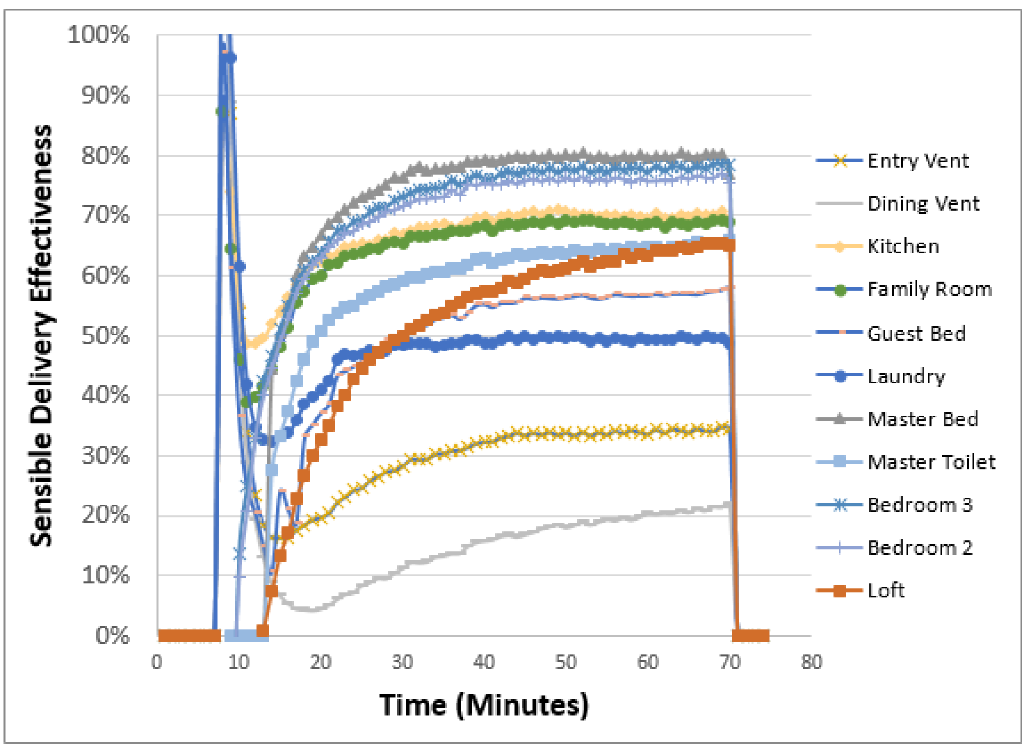
ρ is air density [lb/ft^3]

In the field, the air leaving any particular grille has typically passed through multiple ducts on the way to that grille, which makes the calculation of the NTU for a particular grille considerably more complex. To manage that complexity, a simplified methodology was employed. That methodology utilizes a length-weighted-average value for the parameters impacting performance, which translates to a length-weighted average 1/VD. This represents a way to compare the performance of different grilles in the same house, comparing the NTU associated with each duct run.

Impact of Duct Velocity on Delivery Effectiveness

Figure 136 shows the sensible delivery effectiveness for each grille at the PG&E test site during a period of operation when the unit was running at full speed and all zones were in operation. The attic temperature during this period of operation was 101–106°F (38–41°C). This figure clearly shows that not all ducts experience the same thermal losses as they pass through the attic. The range in sensible delivery effectiveness was between 20 percent and 80 percent. The primary reason for the low delivery effectiveness of the Entry Vent and the Dining Vent was due to the low air velocity in the duct leading up to these grilles which was about a third of the average velocity through the ducts leading to other grills.

Figure 136: Sensible Delivery Effectiveness for Each Grille at PG&E Field Site During Full-Speed Operation

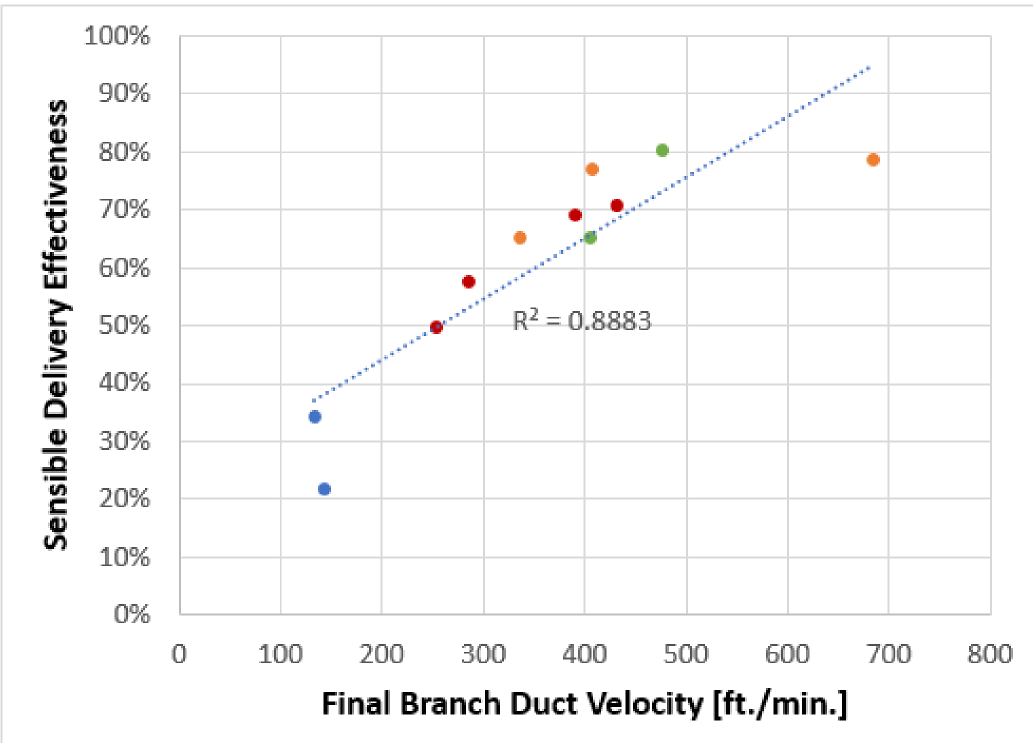


Source: EPRI

Figure 137 shows the sensible delivery effectiveness for each grille at the PG&E test site compared to the final branch duct velocity (the final branch duct velocity is calculated using the diameter of the final duct-run leading to the grille). There is a clear trend showing that increased duct velocity results in increased sensible delivery effectiveness. This result agrees with laboratory measurements in Phases 1 and 2 of this project and follows the trend calculated by the duct models developed therein. The total thermal losses for each grille depend on the entire duct system design; however, using the final branch velocity seems to be a good indicator of the relative performance of each branch in the system. The different colors indicate the different grills of the house, each of which should experience losses through the attic.

Typical maximum velocity in the final branch of a residential duct system is about 600 feet per minute (fpm).[13] Based on the results in Figure 137, designing a duct system to achieve 600 fpm on each branch would result in a sensible delivery effectiveness of more than 80 percent with R-6 ducts located in the attic; however, when this same duct system is utilized with variable-capacity equipment that can reduce flow to less than half of the maximum used for duct design, the sensible delivery effectiveness can drop below 50 percent. *This results in a 30 percent reduction in sensible capacity through a duct system located in the attic when running the system at low capacity with a fully operational duct system.*

Figure 137: Sensible delivery Effectiveness vs Final Branch Duct Velocity for Each Grille, PG&E Field Site



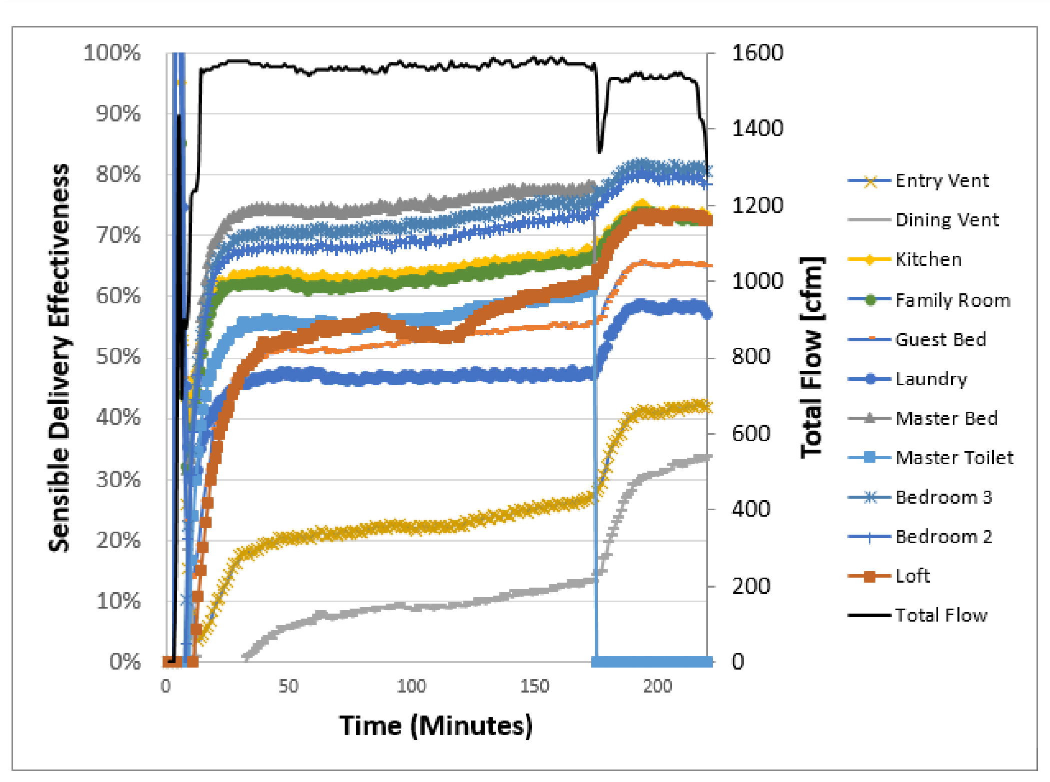
Source: EPRI

Figure 138 shows the sensible delivery effectiveness for each grille with temperature data at the PG&E test house during a period when the system started with all zones open at full speed and abruptly changed as Zone 3 (Laundry and Master Bed) was closed off (at ~175 minutes). Once the zone closed down there was a corresponding increase in air velocity through each of

13 Lindeberg, M.R., Mechanical Engineering Reference Manual for the PE Exam. 12th ed.

the other zones that were still active. The plotted airflow shows that the total flow did not change significantly once the zone was closed off. The average air velocity the other zones increased by about 25 percent after the Zone 3 closed causing an immediate increase in sensible delivery effectiveness of more than 10 percentage points.

Figure 138: Impact of Increased Duct Velocity During Zonal

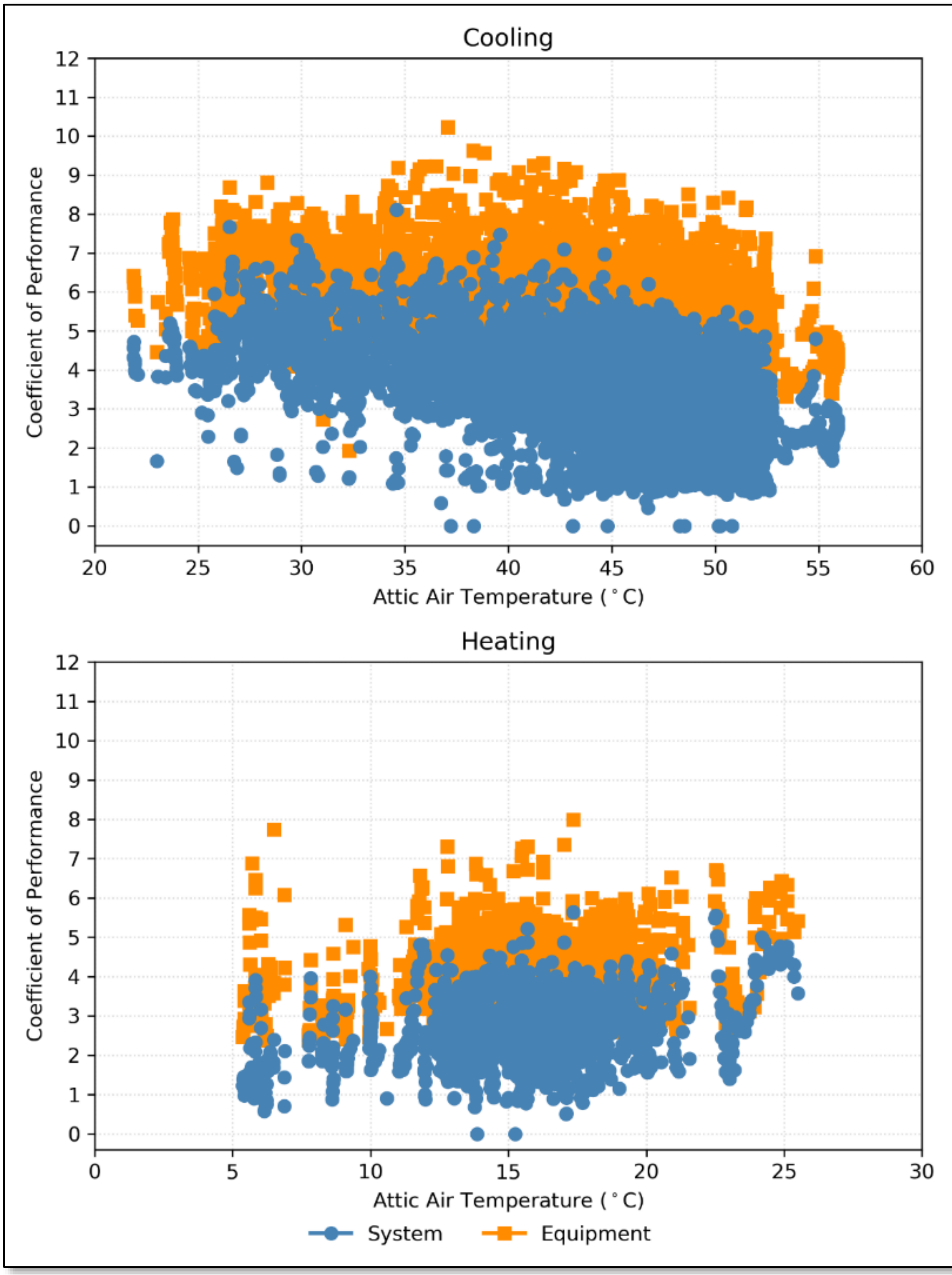


Source: EPRI

SDG&E Field Results

Figure 139 shows the system and equipment efficiency measured for the SDG&E test site during the field evaluations. The efficiencies are plotted against the attic temperature, which is one of the fundamental drivers of delivery effectiveness. The cooling data shows a clear trend downward as attic temperatures increase. The average equipment COP for cooling was 5.2, while the system COP, which accounts for duct losses, was about 3.4. This means that the overall average delivery effectiveness was 65 percent during the monitoring period. The average equipment COP in heating mode was 3.8 and the average system COP was 2.6 which results in an average delivery effectiveness of 68 percent.

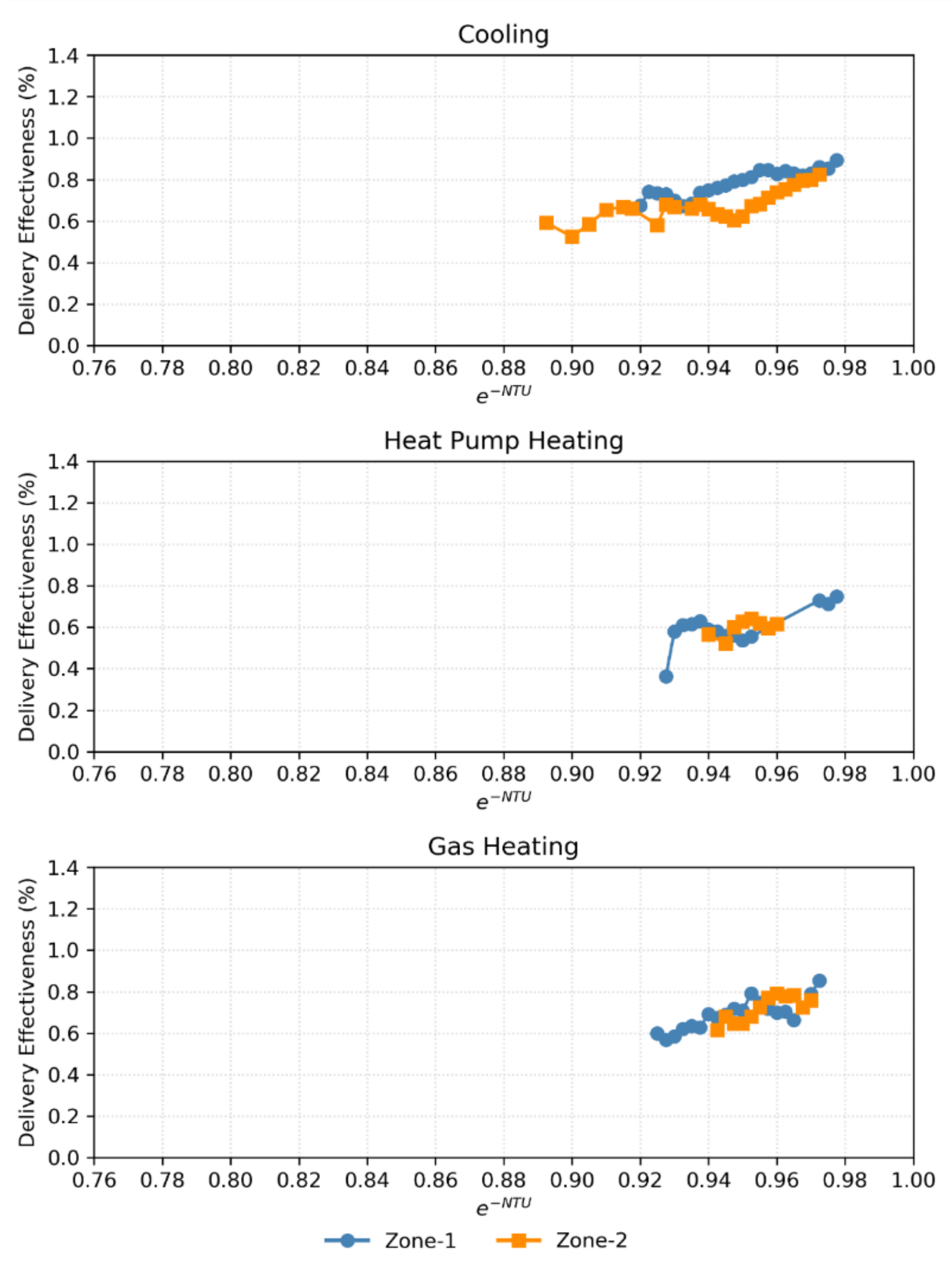
Figure 139: System and Equipment Efficiency vs Attic Temperature for Cooling and Heating, SDG&E Test Site



Source: EPRI

Figure 140 shows the delivery effectiveness for the SDG&E test site plotted against an estimate of e^{-NTU} for the duct system which is the mathematical value from Equation 79 that describes the effectiveness of a heat exchanger. The results for cooling, heat pump heating, and gas heating are provided. The plot shows the average delivery effectiveness for each bin of e^{-NTU} values. The NTU for the duct system is a function of the duct R-value, air velocity, and duct surface area. In the case of a duct system, a lower NTU number is desirable as it reduces the amount of heat transferred between the conditioned air in the duct and the attic. Delivery effectiveness is shown generally increasing as the e^{-NTU} of the duct system increases. A typical single-speed duct system does not have a changing NTU number, but with variable speed equipment, the NTU value for the duct system goes up (resulting in lower e^{-NTU} values) as the indoor fan slows down. Reducing fan speed for a duct system results in higher NTU values, due to air's longer residence time, increasing heat losses to the attic, whereas reducing surface area of the duct system in operation (through zoning) results in lower NTU values.

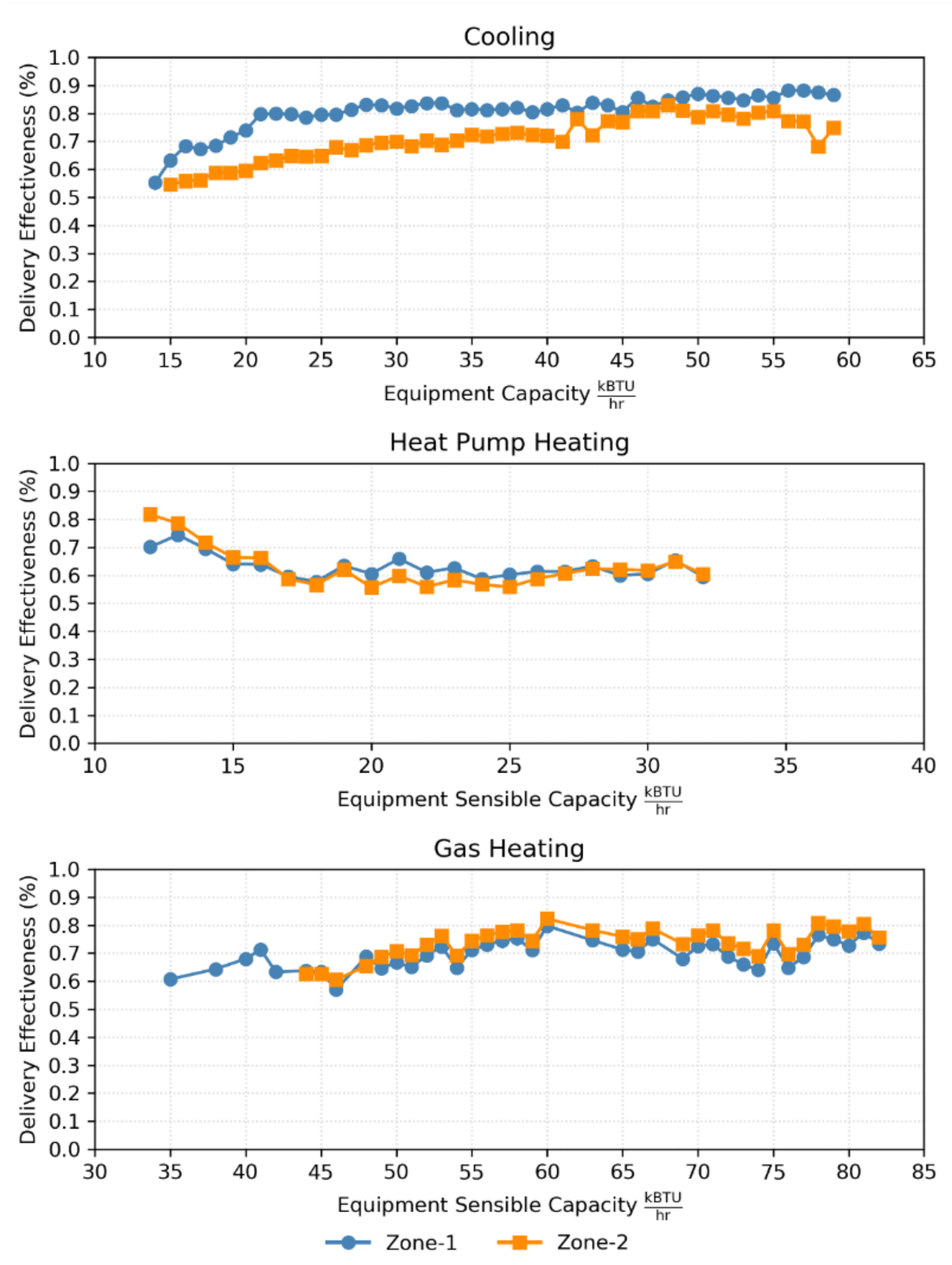
Figure 140: Delivery Effectiveness vs e-^{NTU} for Duct System, SDG&E Test Site



Source: EPRI

Figure 141 shows the delivery effectiveness plotted against bins of equipment capacity for cooling, heat pump heating, and gas heating. The equations for calculating delivery effectiveness also depend on equipment capacity. Increasing equipment capacity tends to increase delivery effectiveness which is demonstrated in Figure 141 for cooling and gas heat. However, for heat pump heating, this was not the case, showing an increase in delivery effectiveness at low capacities.

Figure 141: Delivery Effectiveness vs Equipment Capacity for Duct System, SDG&E Test Site

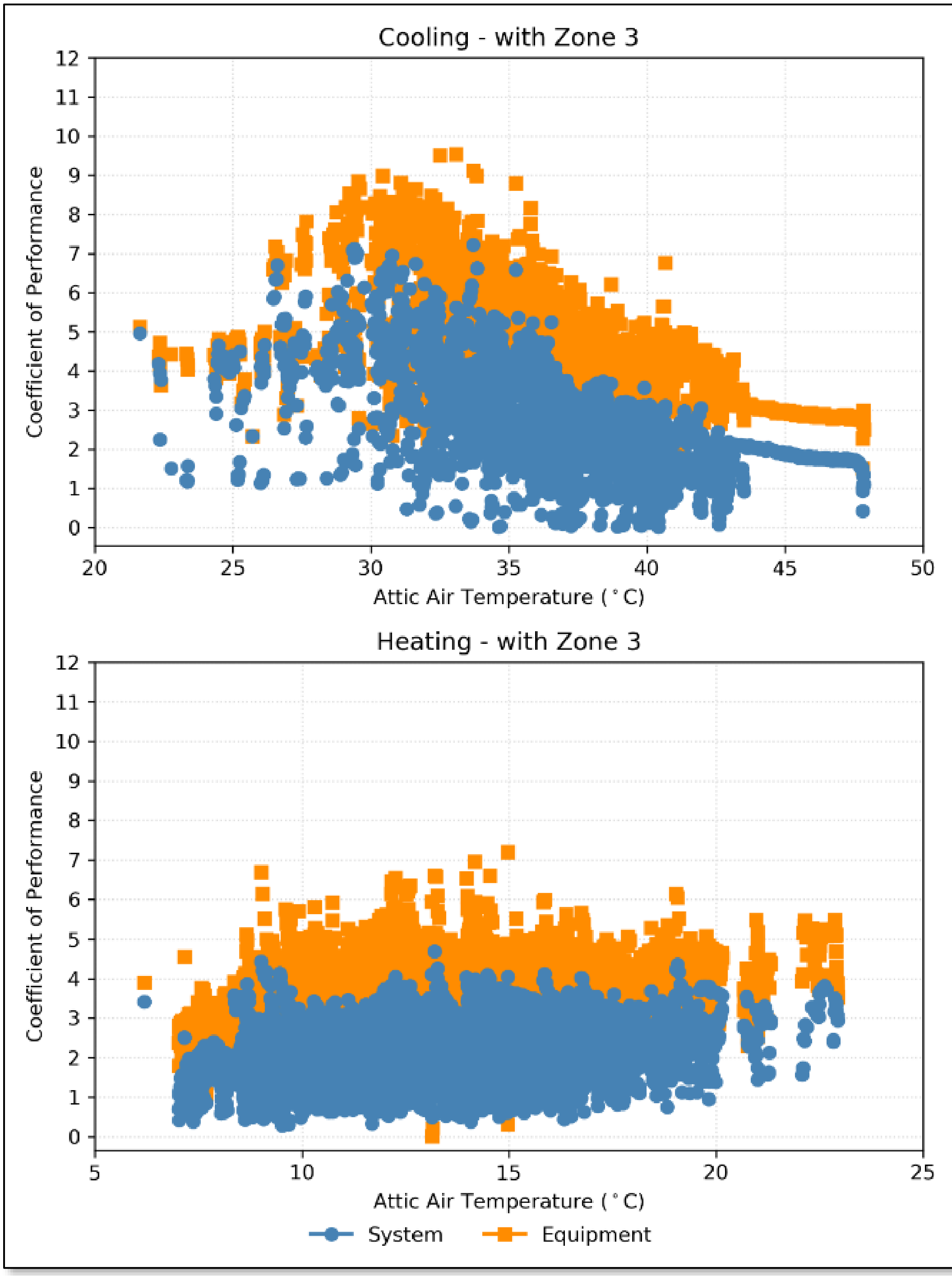


Source: EPRI

PG&E Field Results

Figure 142 shows the equipment and system efficiency plotted against attic air temperature for the PG&E test house.

Figure 142: System and Equipment Efficiency vs Attic Temperature for Cooling and Heating, PG&E Host Site



Source: EPRI

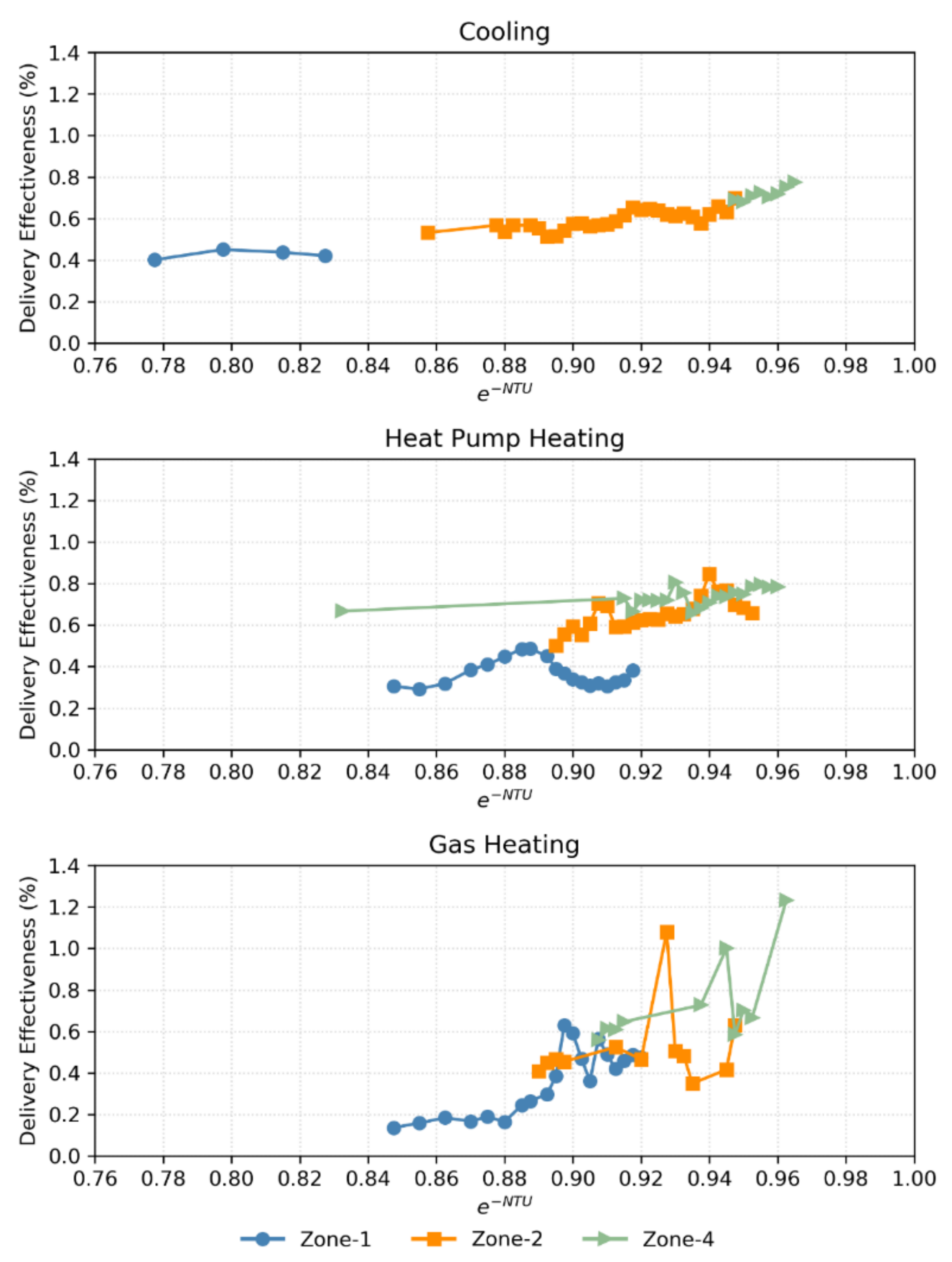
The average equipment and system COP during the cooling operations for this site was 4.2 and 2.6, respectively. This results in an average delivery effectiveness of 62 percent. There also appears to be several observations that show system COP close to zero suggesting no cooling was accomplished even after filtering the data for periods when the outdoor unit has been operating for at least five minutes. The heating results show an average equipment COP of 3.1 and system COP of 2.5. This results in an average delivery effectiveness of 81 percent which is significantly higher than the effectiveness calculated for the cooling season.

The delivery effectiveness was slightly lower during the cooling season and higher during the heating season for the PG&E site compared to the SDG&E site. The two primary differences between the systems installed at the two sites were the duct R-value that was installed and the control strategy. The SDG&E location had R-8 ducts installed whereas the PG&E home had R-6 ducts installed. R-value of of the duct system is inversely related to the overall NTU value calculated for a duct system, with R-6 resulting in a 25 percent reduction in the NTU value for the duct system compared to R-8.

The PG&E site had a different control scheme implemented at the end of the cooling season in an attempt to test the optimization logic demonstrated in the Phase 2 lab testing. The control scheme attempted to force the demand for a given zone to be fixed based on the size of that zone. The goal was to maintain higher duct velocities by not allowing the capacity/airflow delivered to a particular zone to modulate down (the unit would still be allowed to modulate based on the number of zones requesting heating or cooling). With so many variable affecting delivery effectiveness, there is not a clear explanation for the differences observed in delivery effectivess but one possiblity is that the change in control method resulted in an increased delivery effectiveness during the heating season for the PG&E home.

Figure 143 shows the delivery effectiveness for the PG&E test site plotted against an estimate of e^{-NTU} for the duct system. This is a binned analysis that averages the results from all observations for a range of e^{-NTU} values for each mode of operation of the system. Note that Zone 3 is not shown in the delivery effectiveness plots because several grills did not have temperature data and there were no other grills that were considered representative of the grills without temperature data. The same general trend as shown in Figure 140 for the SDG&E home is observed in Figure 143 with gradually increasing delivery effectiveness as e^{-NTU} increases. This normalization of the duct system based on e^{-NTU} allows each zone to be compared more directly since this value accounts for differences in duct air flow, duct surface area, and duct R-Value. As seen in Figure 143, each zone (although displaying very different characteristics) follows the same general trend.

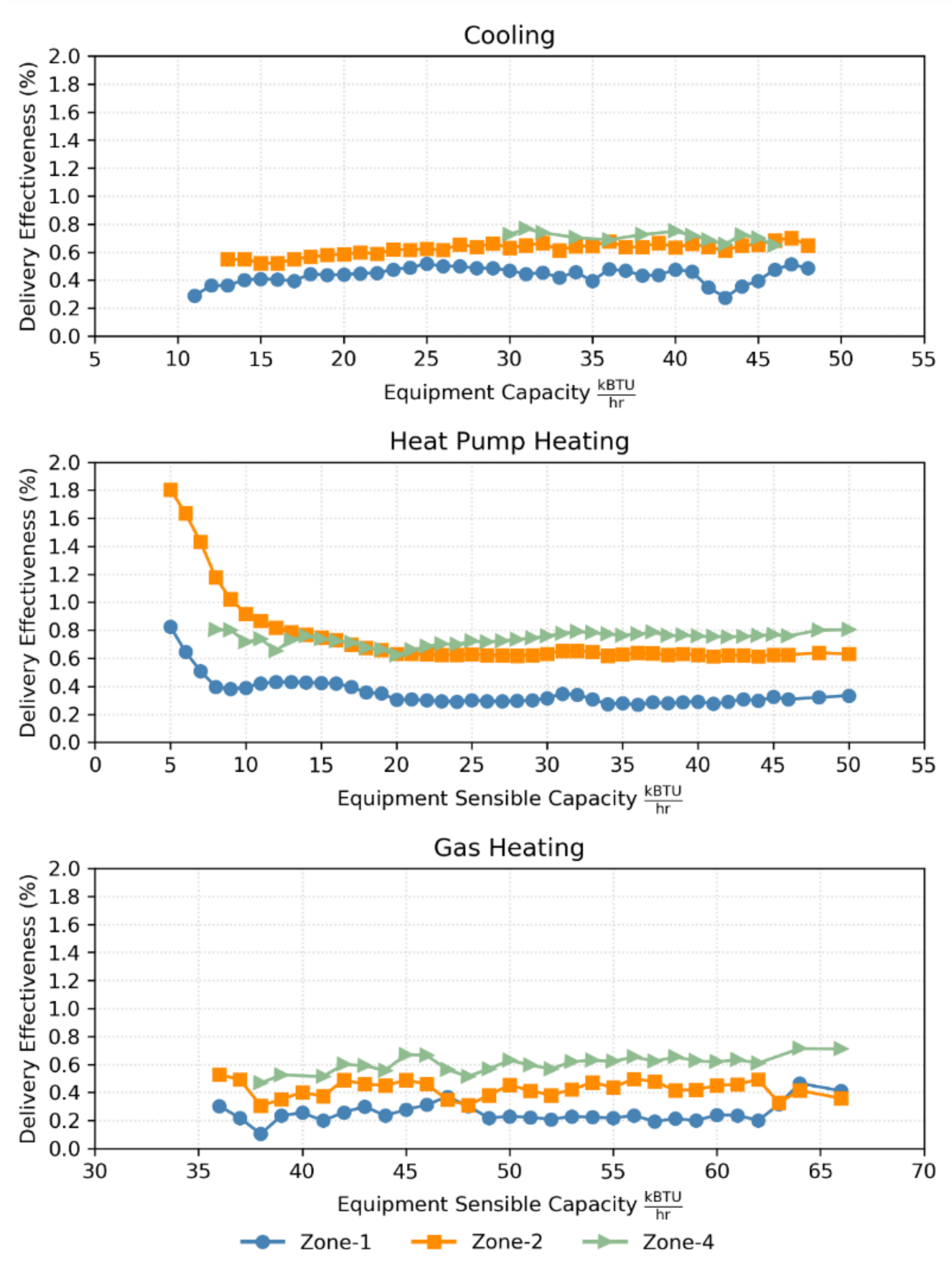
Figure 143: Delivery Effectiveness vs e^{-NTU} for Duct System, PG&E Host Site



Source: EPRI

Figure 144 shows the delivery effectiveness plotted against bins of equipment capacity for cooling, heat pump heating, and gas heating. As noted, the equations for calculating delivery effectiveness also depend on equipment capacity. The same unusual trend was observed at the PG&E home with a sharp increase in delivery effectiveness at low capacities. This trend may be investigated further to understand the reasons for the increase.

Figure 144: Delivery Effectiveness vs Equipment Capacity for Duct System, PG&E Host Site



Source: EPRI

Feedback from Homeowners

On June 8, 2018, the EPRI team hosted a call with the homeowners and the technology provider to exchange feedback, experiences, and address any questions, challenges, concerns. Attendees included the 3 host site home owners, Daikin/Goodman representatives, EPRI and WCEC team members. The three homeowners shared their experiences with the Next-Gen RSCS thus far. Each homeowner's testimony is summarized below.

SDG&E host site owner said that the installation process went smoothly, and the few glitches she discovered were fixed quickly. Her house was previously not zoned, and she was surprised by the amount of work that needed to be done to install the zonal control system. The outdoor unit of the old system was adjacent to her bedroom, and it caused the bedroom walls to vibrate. Since the new unit is bigger than her previous single speed unit, it was placed in the backyard because there was not enough space to put it in the same location as the older unit. Unfortunately, the new unit is also outside of her bedroom wall, but the noise is not as bothersome.

Because of San Diego's mild climate, she has not needed to use the system much since its installation. The fan circulation works well. She expressed that she would like to be able to control the units remotely from her phone, but the thermostats are not WIFI-capable. She was able to add a WiFi control feature to one zone (master) by adding another piece to the thermostat but could not successfully connect both thermostats.

SCE host site owner appreciated the HVAC contractors' professionalism and responsiveness both during installation and afterward when parts needed to be fixed or replaced. He has three thermostats in his house and keeps the air handler in the attic. The house now has a three-zone system, as he created a new zone for the master bedroom. He has had difficulty with zonal control (excessive flow in some zones versus others based on setting) and thinks this aspect of the system might require more assistance. He has the Honeywell red link connected router coupled with his smart device, but he does not find the remote control to be as intuitive as what he had previously. He finds the outdoor unit to be extremely quiet (he can barely hear it turn on), but the air flow unit in the master bedroom is noisy because there is more flow than necessary. He suggests balancing the airflow and making adjustments to the main fan because there are limitations as to how low the flow can go based on the zones. He expressed interest in having a way to know when the furnace turns on versus the heat pump because he would prefer not to burn fossil fuels if the heat pump can perform the same function.

Before inspection from the city, the unit was not providing cooling and the contractor came to look at it and found refrigerant leakage. The inspection was successful after the leakage was repaired. The previous Saturday, all the thermostats were off, and the control board was not receiving power. He was unable to operate the system as there was no way to bypass the control board. He left a message for the contractor and they came to fix the problem on Monday, so he was further pleased by their responsiveness. He would like to see whether the fan speed can be controlled to a lower setting.

PG&E host site owner upgraded his home from two zones (one per floor) to four zones (two per floor) and kept the air handler in the attic as well. He agreed with the other homeowners that installation went smoothly. He has a very quiet unit and thinks that the fan circulation is a nice feature. He found some limitations to the EWC control board with respect to the testing he wanted to do, but he replaced the boards with a new version and he had no further concerns about control board limitations. Like the SCE homeowner, he found there to be too much airflow in just one zone, so he made fan adjustments by working with the EWC controls and contractor menus on the thermostat. When he was still unable to adjust the airflow in one zone to a reasonable level, he switched to a three-zone system. Additionally, he changed the dip switches (furnace control electric switches) on the air handler to change the airflow setting of the unit because he did not think the contractor properly adjusted those settings. These changes improved the airflow somewhat, but he does not think it would have helped if he had not already removed one of the zones. This is because the EWC cannot control airflow directly, so the changes must be made using controls on the outdoor unit. He therefore suggests the three-zone setup due to the limitations of allowable airflow setpoints. He does not have the system connected to WiFi but thinks it is a nice feature for the system to have.

Further Follow-Up with Homeowners

The EPRI team has had regular follow-ups with the homeowners, which included notification of contractor visits to address unit maintenance and site visit in Jan 2019 to carry out flow mapping of the ducting systems.

In summary, the homeowner's provided the following feedback on their experience with the Next-Gen RSCS:

- They appreciate how much quieter the Next-Gen RSCS operates compared to their previous single speed AC unit.
- They like how quickly it cools or heats the space.
- They like having an app-based controller for the Thermostats to turn on and set the temperature of individual zones.
- They like the ability to control temperatures in individual spaces (an advantage and nice attribute of zonal control is to control zones independently).
- While zonal control is a convenient feature, it added complexity to the VCHP system use for more than 2 zones. Airflow was too forceful in certain zones, thus making it noisy in certain rooms. Once the weighting of the zones was adjusted with the Zonal Control board, this mitigated the effect of the airflow imbalance.

Stakeholders Webinar Summary

A key deliverable for the project is a workshop to the stakeholders to share the project results and recommendations. CEC scheduled the workshop as a public webinar on March 27, 2019, from 10am – noon pacific time. The webinar notice was posted on the CEC's Research Notices website: https://www.energy.ca.gov/research/notices/, http://innovation.energy.ca.gov/SearchResultProject.aspx?p=30005&tks=636041876838279811.

The announcement titled, "Project Results and Technology Recommendations", included the following summary and agenda:

> The objective of this webinar is to share the results of the project's laboratory and field evaluation results and technology recommendations to California stakeholders and manufacturers, for the California Next-Generation Residential Space Conditioning

Systems. Additionally, the webinar will provide technology transfer activities, lessons learned and next steps. The Next-Generation Residential Space Conditioning System consists of a ducted split-system variable capacity heat pump unit with energy-efficiency technology features incorporated into it and optimized for the California climate. The features are:

- Variable capacity compressors and blowers/fans to improve part load cooling and heating performance.
- Integrated ventilation control to provide heat recovery heating and cooling
- Zonal control using variable capacity capabilities
- Automatic demand response strategies using variable speed technology to reduce residential peak demand while maximizing user comfort.
- Advanced fault detection and diagnosis (FDD) to optimize maintenance, improve reliability and postpone or avoid major equipment failures
- Intelligent heating in the form of a dual fuel heat pump that selects whether to use a heat pump or gas furnace to provide heating comfort at the lowest energy cost possible
- Use of alternative refrigerants to reduce global warming potential and possibly improve device efficiency

Topic	**Presenter**
Welcome & Background	Jackson Thach, CEC Ammi Amarnath, EPRI
Project Overview: Scope, Features Tested, Summary Results	Ammi Amarnath, EPRI
Project Methodology	Sara Beaini, EPRI
Next-Generation Residential Space Conditioning System Evaluation Results	Sara Beaini, Aaron Tam, EPRI Curtis Harrington, WCEC
Recommendations and Lessons Learned	Sara Beaini, EPRI Curtis Harrington, WCEC
Technology Transfer	Sara Beaini, EPRI
Questions & Discussion	All

Source: EPRI

The webinar agenda, slides and recording have been posted on CEC's webpage: https://www.energy.ca.gov/research/notices/#03272019.

More than 100 stakeholders attended the webinar and actively participated in the interactive discussions at the end of the presentation. About 40 percent of webinar attendees included CEC personnel as well as California utility members. The questions and answers discussion are summarized in Appendix G.

Field Evaluation Results and Discussion

The objective of the field evaluation is to assess the functionality of the Next-Gen RSCS features in residential homes and evaluate the performance with respect to the customer experience. Field evaluation data began in August 2018 and will continue through fall 2019. Given the timeline of the project, this report contains the data analysis through April 2019. A Technical Update will follow that captures the remainder of the field evaluation data results.

Thus far, the key results from the five features assessed are:

a. Variable Capacity Heat Pump (two features: variable capacity compressor and variable speed blower):
    - The cooling and heating seasonal analysis both show energy consumption within 20 percent of the lab data model. Some discrepancies could be caused by the difference between the indoor chamber setpoint during lab testing, and the thermostat setpoints at the host sites.
    - The cooling performance of the units show a range of EER between 10 and 25, which agrees with the range of EER shown in the lab testing in Phase 1.
    - The cooling performance of the units show a range of COP between 2-5, which is significantly higher than the efficiency of the fossil fuel heating options (maximum COP of 1).

b. Dual Fuel (Intelligent Heating):
    - Once the breakeven temperature was set for each of the three host sites, the Next-Gen RSCS performed as expected in heat pump mode and gas furnace mode throughout the duration of the heating season. The Next-Gen RSCS dual fuel capability provides an important choice for the residential customer to adjust the unit settings based on economic factors (utility rates), efficiency factors (heat pump vs gas furnace), or environmental factors (reduction of carbon footprint).
    - Having a controller that automatically receives pricing information and sets the breakeven temperature accordingly would minimize the need for customer interaction (unless they chose to do so.) Future work is needed to develop a versatile heating controller that can receive a signal based on utility prices and customer choice/preference (price, efficiency, fuel mix distribution), which would optimize the Next-Gen RSCS's performance in the heating mode.

c. Auto Demand Response (ADR):
    - For both lab and field ADR testing, the unit capacity reduction is less than the unit power reduction, thus the customer comfort is not as compromised while energy savings can be realized during a utility-led demand response event.
    - Field ADR testing was conducted in the heating mode and will be conducted in cooling mode during the upcoming cooling season. Updated results will be provided in the Technical Update to follow in late 2019.

d. Zonal Control and Duct Delivery Effectiveness:
    - The field evaluations all included zoning equipment based on the learnings from the lab test results that showed significant efficiency gains when zoning variable-capacity equipment. There were some challenges implementing the zoning that

led to too much airflow through smaller zones in the home during some periods of operation. For example, during defrost cycles, the indoor fan would ramp up to high speed without consideration of the number of open zones. The project team was able to adjust the zoning controls to minimize the issues around over-flow in small zones, but there should be better coordination between the zone controller and the heat pump that allows the zone controller to adjust dampers based on feedback from the heat pump about the mode of operation or fan speed.

- This project demonstrates a clear need for zoning when installing variable capacity equipment with ductwork located in unconditioned space. A minimum of two zones is recommended since there are challenges with adding additional zones. Two zone systems with proper controls would achieve much of the benefit of the optimized approach demonstrated in Phase 2.

e. Customer Feedback:

- Appreciate how much quieter it operates compared to previous single speed AC unit.
- Like how quickly it cools/heats the space
- Like having app-based controller with Thermostat to turn on specific zones
- Zonal control added complexity to system use while providing convenience of controlling zones independently, if more than 2 zones.

CHAPTER 5: Evaluation of Project Results (Task 5)

Technologies Evaluated

The Next-Generation Residential Space Conditioning System consists of a ducted split-system variable capacity heat pump unit with energy-efficiency technology features incorporated into it and optimized for the California climate.

The following technologies were evaluated in this Next-Gen RSCS project to reduce heating and cooling energy use:

- Variable capacity compressors and blowers/fans to improve part load cooling and heating performance.
- Integrated ventilation control to provide heat recovery heating and cooling
- Zonal control using variable capacity capabilities
- Automatic demand response strategies using variable speed technology to reduce residential peak demand while maximizing user comfort.
- Advanced fault detection and diagnosis (FDD) to optimize maintenance, improve reliability and postpone or avoid major equipment failures
- Intelligent heating in the form of a dual fuel heat pump that selects whether to use a heat pump or gas furnace to provide heating comfort at the lowest energy cost possible
- Use of alternative refrigerants to reduce global warming potential and possibly improve device efficiency

All technologies tested are scalable and can be expeditiously commercialized by the manufacturing partner or similar organizations. Most of the technologies tested are already commercially available and some, such as FDD, may need modest refinement or enhancement. The technology furthest from commercialization is likely to be the use of an alternative refrigerant such as R-32. The time frame for adoption of R-32 is dependent, in large part, upon regulatory action regarding its use as a refrigerant and its subsequent optimization for residential applications.

Barriers to Market Penetration

The main barriers are likely to be increases in first cost, user unfamiliarity with the technologies and their benefits. The research is approaching the first cost barrier by providing information on operating cost savings, and other features such as improvement in reliability, and comfort that will offset this increase in first cost. Information will be delivered in the most expeditious manner using existing channels and organizations, as outlined in the technology transfer plan (For more details see Chapter 6).

The technology transfer plan will assure that results of the project are widely disseminated to key market participants including specifiers, distributors, manufacturers and code setting officials in order to overcome these barriers. EPRI is working directly with key manufacturers including Daikin to assure that the technology elements that are deemed to be cost effective and beneficial to California residential HVAC users and California ratepayers as a whole are accessible in the residential market and understood by that market.

Comparing the Next-Gen RSCS to Existing Systems

Cost (non-energy)

The technologies deployed in the Next-Generation Residential Space Conditioning System will generally have higher first costs than the technologies they are replacing, but these costs will be offset by lower energy costs and other benefits including improved reliability and greater comfort.

Reliability

The use of FDD will increase reliability of the units, anticipating failures and conducting timely maintenance and repair to reduce system down time. Variable capacity should provide smoother operation, reducing equipment stops and starts, thus reducing stress on components, resulting in increased equipment life.

Safety

The anticipated higher reliability of the Next-Generation Residential Space Conditioning System should provide greater occupant safety when comfort is a safety issue (as in extreme climate situations). If R-32 is used as the refrigerant in future models of the Next-Generation Residential Space Conditioning System then flammability issues must be addressed in the equipment design, placement and operation to assure that safety is equivalent to or better than the equipment that it is replacing.

Consumer Appeal

In addition to the cost, reliability and safety issues just identified, the following Next-Generation Residential Space Conditioning System advantages should appeal to consumers:

The Next-Generation Residential Space Conditioning System will provide greater comfort during normal operation due to use of variable capacity to permit load following, closer temperature control and less on-off cycling.

The use of FDD will increase reliability of the units, reducing down time, repair bills and loss of space conditioning capability.

The use of a furnace and a heat pump as heating options affords the user with the opportunity to take advantage of the lowest price source of heating. This should appeal to cost conscious consumers.

User Behavior

The energy savings determined by project testing will largely not require changes in user behavior. In this regard, it is assumed that thermostat controls will be employed to achieve optimal zoning benefits. Also, electricity and gas prices will be needed to be automatically input into intelligent heating (dual fuel) controls (thermostat settings for determining the breakeven temperature). If either of these is not the case, then direct user intervention will be required to achieve zoning and/or intelligent heating benefits.

FDD systems will require the user or a contractor to respond to a signal/notification indicating that action is needed to address potential system malfunctions. If the user delegates this function (as well as access to the unit) then no user action will be required.

Demand response efforts will likely require the user to accept modest increases in indoor temperature to enable the demand reductions to take place. These reductions in comfort will be less however than those required when employing demand response with conventional residential space conditioning technology

Benefits to Ratepayers

Energy Efficiency and Environmental Benefits

Based on the standard emission factor (0.331 kg/kWh saved) and energy savings of 475 GWh/yr or 475 million kWh/yr, the Next-Gen RSCS could provide a reduction of 157 million kg of CO_2 emission per year in California. EPRI's own research with the California Energy Commission[14,15] and other research has shown that variable-speed space-conditioning systems can save from 20 to 40 percent of energy expended annually over conventional fixed –speed systems in California or comparable climates. Testing performed in this Next-Gen RSCS project confirmed these savings, demonstrating 22 percent to 32 percent savings in laboratory cooling mode testing, compared to a baseline 14 SEER air-conditioner for residential applications across the California Climate Zones (for more details see Chapter 2).

The research also shows that energy cost savings of 50 percent are technologically achievable from employing the Next-Gen RSCS compared to commonly used residential HVAC systems. If it were to achieve a 20 percent market penetration, the Next-Gen RSCS is estimated to have the technical potential of saving $475 GWh/yr, or approximately $83 million annually. Assuming an equipment life of 15 years, the Next-Gen RSCS could save California ratepayers more than $1 billion over the lifetime of the equipment.

- Based on laboratory test data, modeling showed that heat recovery ventilators could provide 1 percent to 4 percent seasonal cooling energy savings and up to 1 percent seasonal heating energy savings when compared to a baseline variable capacity heat pump system (for more details see Chapter 2).
- Zonal control, or limiting space conditioning in unoccupied spaces, has the potential to save the 15.9 percent of energy used to condition unoccupied spaces.[16] Testing and modeling work showed that the variable speed capability of the Next-Gen RSCS system could provide savings that were greater than the amount of load reduction afforded by not conditioning unoccupied spaces. A 10 percent load reduction would likely

[14] B. Fortenbery et al. ((Electric Power Research Institute (EPRI). Consumer Electronics and Motorized Appliances, California Energy Commission Contract 500-10-022, 2014

[15] W. Hunt, R. Domitrovic, and A. Amarnath. Cooling Efficiency Comparison between Residential Variable Capacity and Single Speed Heat Pump. ASHRAE 2013 Annual Conference, Denver, CO.

[16] Meyers, R.J., Williams, E.D., Matthews, H.S. Scoping the potential of monitoring and control technologies to reduce energy use in homes, Energy and Buildings, 42(5): 563–569, 2010.

correspond with a 12.8 percent HVAC power reduction, and a 20 percent load reduction would correspond to 25.7 percent HVAC power reduction, based on the laboratory evaluation. Efficiency effect depends on the temperature offsets in the unoccupied zones that are driving the load reductions (for more details see Chapter 3).

- Automated demand response (ADR) coupled with variable speed compressor and fan operation can provide greater cooling (or heating) output for the same power as a fixed speed system when operating at part-load conditions. Test results for simulated ADR events showed that a 50 percent power reduction resulted in only a 38.2 percent capacity reduction, and a 70 percent power reduction resulted in only a 61.8 percent capacity reduction. This incremental capacity permits the residential customer to achieve a greater level of comfort when responding to an ADR event. Since the room temperature will have to recover less after the event, energy efficiency will also be improved (for more details see Chapter 2).
- Intelligent heating (dual fuel heat pumps) use an electric heat pump for cooling and heating and a gas-fired furnace as an alternative heating means and also for heating backup. Modeling results for the 16 California climate zones and selected residential electric and gas prices and calculations run for all 16 California climate zones using average California residential electricity (15.54¢/kWh) and gas prices (129¢/therm) showed energy cost savings ranging from 2.5 percent to 28.5 percent when comparing the dual fuel heat pump to a 97 percent efficiency gas furnace. For average electric and gas prices, the highest savings were about $200 per heating season for Oakland in climate zone 3 (22 percent energy cost savings) and Arcata in climate zone 1 (15 percent energy cost savings). For more details see Chapter 2.
- Use of an advanced refrigerant such as R-32 could contribute to a reduction in global warming. Laboratory testing of R-32 by other investigators[17] showed comparable or improved cooling and heating efficiencies and capacities compared to R-410A. R-32, while mildly flammable, has a global warming potential (GWP) of 675, compared to a GWP of 1800 for R-410A. Assuming typical leakage rates, an R-32 system will emit only 33 percent as much CO_2-equivalent direct emissions as the equivalent R-410A system over its lifetime. This value may be even lower in reality because systems using R-32 will have a lower refrigerant charge than R-410A systems. These global warming savings, while substantially less than the equivalent CO_2 emissions savings due to lower energy use, are still considerable. Laboratory testing in the current study found that use of R-32 in lieu of R-410A within the Next-Gen RSCS provided an annual HVAC cooling efficiency improvement of 1.2 percent to 3.0 percent and a peak HVAC cooling improvement of 6.7 percent to 8.2 percent. Cooling performance improvement with R-32 in the variable capacity system was generally highest in the warmest California climate zones. This improved cooling performance with R-32 translates to peak cooling demand reduction of 6.7 percent, 7.0 percent, and 8.2 percent compared to R-410A at

[17] For example, Li, Hao, and By, Bob (2012), Soft-optimized System Test of Refrigerant R-32 in 3-ton Split System Heat Pump, Goodman Manufacturing for AHRI Low-GWP AREP. http://www.ahrinet.org/App_Content/ahri/files/RESEARCH/AREP_Final_Reports/AHRI_Low-GWP_AREP-Rpt-005.pdf. R32 capacity is 4% higher than R410A at A condition and 2% higher in B condition, EER/COP is 1 to 2% higher for A/B/H1 conditions.

outdoor temperatures of 95°F, 105°F and 115°F, respectively. In the heating mode, capacity was improved by 5 percent to 10 percent using R-32 but the COP was reduced by 2 percent to 4 percent with R-32. R-32 provided better performance at outdoor temperatures above 35°F (2°C), and R-410A generally provided better performance at outdoor temperatures below 35°F (2°C). R-32 reduced system charge by 29 percent compared to R-410A (for more details see Chapter 3).

- Fault detection and diagnostic (FDD) systems can provide substantial energy savings. Faults in the installation and operation of space conditioning equipment can reduce efficiency by 10 percent to 55 percent.[18] These estimates included 3 percent to 12 percent for low air flow, 4 percent to 7 percent for evaporator blockage, 7 percent to 36 percent for duct losses, and 4 percent to 56 percent for refrigerant mischarging. The premise is that these could be avoided, in part, with early detection and correction. Other investigators reported a range of achievable savings when deploying FDD systems.[19] Fault detection and diagnostic system testing in the Next-Gen RSCS project provided some guidance for improving the FDD algorithms in future versions of the Next-Gen RSCS that could enable achieving some of the savings asserted in the literature (for more details see Chapter 3).

Utility Integration and Demand Benefits

HVAC is the single largest contributor to peak power demand in the state, comprising up to 30 percent of total demand in the hot summer months. Next-Gen RSCS technologies can greatly reduce peak demand by employing automated demand response capabilities coupled with variable speed compressors and air-moving equipment.

Intelligent, variable-speed space-conditioning equipment has been demonstrated to reduce power by 60 percent during peak events, while continuing to provide cooling to the conditioned space.[20] Variable-capacity systems have the unique attribute of going to a state of higher operating efficiency when the compressor speed is reduced. This Next-Gen RSCS project confirmed that power reductions would exceed capacity reductions when operating the units below maximum speed. This provides demand savings and increased cooling capacity compared to equivalent single speed units. According to its strategic plan, the CEC estimates that a peak demand reduction of 1,096 MW could be achieved through high-quality HVAC installations by 2020. Assuming market penetration of 20 percent and the ability of the system to reduce peak demand by 60 percent, the cumulative peak demand reduction for California would be approximately 1.5 GW. Thus a 1 to 1.5 GW in demand reduction due to deployment

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[18] R. Mowris, E. Jones, and R. Eshom. Laboratory Measurements of HVAC Installation and Maintenance Faults, ASHRAE Transactions. 2012.

[19] Springer, David, (2016), Expert Meeting Report: HVAC Fault Detection, Diagnosis, and Repair/Replacement, Alliance for Residential Building Innovation (ARBI). http://apps1.eere.energy.gov/buildings/publications/pdfs/building_america/emr-hvac-fault-detection-diagnosis-repair.pdf.

[20] Variable-Speed Heat Pumps for Energy Efficiency and Demand Response: Field Testing High-Efficiency Systems in a Simulated-Occupancy Home in Knoxville, Tennessee. EPRI, Palo Alto, CA 3002003925, 2014

of units similar to the Next-Gen RSCS appears to be an aggressive but potentially achievable goal.

CHAPTER 6: Technology/Knowledge Transfer Activities (Task 6)

Technology Transfer Plan

This chapter outlines the objectives of the technology transfer effort. Relying on conferences, symposia, meetings, publications, and websites already existing in the industry and including trade associations, professional associations, and other stakeholder organizations, the plan identifies transfer activities the project team has already accomplished and activities yet to be accomplished through presentations, articles and white papers, and specialized efforts.

Objectives and Summary

The objective of the technology transfer effort was to improve the market focus of the project's technology products and thereby increase the public benefits of EPIC's investment in the program.

Key market participants are identified and their functions in the market are described along with issues affecting their product preferences and suggested actions/stimuli for addressing these issues. Features that are important to decision makers are compared to the features expected to be embodied in the Next-Generation Residential Space Conditioning System, providing a focus for emphasizing those attributes that could result in product selection.

Codes and standard issues, actions needed to address these issues and organizations involved in setting and monitoring codes and standards are tabulated.

A suggested action plan lists the information products to be developed, venues for disseminating the products and delivery dates

Product Features

The following paragraphs describe the features being studied and their potential advantages (and liabilities) as obtained from a review of the literature. The potential benefits and limitations are compared to features sought by market participants in selecting residential space conditioning equipment.

Variable Capacity Compressors and Indoor Blowers and Fans

Devices that must meet a range of loads and output requirements have to be modulated. This modulation can be accomplished with on-off operation, throttling with control valves, unloading, or other means. Variable capacity equipment using adjustable speed drives (ASDs) typically control the speed of motors and therefore the output capacity by varying their frequency. Adjustable speed drives are typically 92 percent to 95 percent efficient[21] and provide more efficient operation than the alternative means of modulation.

Adjustable speed drives provide opportunities to continuously control motor driven equipment. Soft starting, ramping, and braking result in smoother equipment operation and therefore should reduce system maintenance[22] and improve system reliability. In addition to smooth operation, adjusting speed to meet the load affords the opportunity to eliminate start and stop transients under off-design, part load operation.

Also, speed and capacity modulation will likely result in longer operating times and less compressor on-off cycles, affording fewer abrupt changes in the sound levels coming from the units that could potentially disturb occupants. The ability to control capacity permits closer space temperature control than with on-off devices, thus resulting in greater comfort by ramping temperature up or down more gradually.

Low speed compressor operation is equivalent to having a smaller compressor and correspondingly oversized heat exchangers, resulting in lower system temperature differences and correspondingly higher equipment efficiencies. A previous EPRI study for the CEC showed 36 percent savings for residential HVAC equipment.[23] Other work reported up to 35 percent savings for HVAC compressor operation.[24]

Disadvantages of adjustable speed drives are higher initial cost, harmonics that could cause facility interference, motor heating at low speeds, and the typical temperature and contamination constraints of electronic equipment.[25]

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[21] March 2007, *Variable Speed Drives*, CTG006, Carbon Trust. http://www.energylab.es/fotos/081105155611_5gf9.pdf.

[22] September 21, 2008, Stark, G., AC *Adjustable Speed Drives*, Process Automation Control, http://www.pacontrol.com/download/Adjustable-Speed-Drives-Tutorial.pdf. The ASD does not cycle motors on and off, as commonly seen with certain processes. By eliminating the cycling of these motors, the variable frequency drive eliminates the amount of in-rush and the torque pulsations felt throughout the system. Smoother operation reduces stress produced by start and stop transients and consequent wear on components.

[23] March 14, 2014, Lawrence, Roger and Blatt, Morton, *Adjustable Speed Drives in Residential Appliances*, White Paper, EPRI for CEC/PIER 500-10-022, Testing of ASDs in central air conditioners, showed 36% or 533 kWh/yr. energy savings for the equipment using ASDs compared to baseline single speed equipment.

[24] December 9, 2015, Li, Yunhua, *Variable Frequency Drive Applications in HVAC Applications*, in "New Applications of Electric Drives", book edited by Miroslav Chomat, ISBN 978-953-51-2233-3, http://www.intechopen.com/books/new-applications-of-electric-drives/variable-frequency-drive-applications-in-hvac-systems.

[25] March 14, 2014, Lawrence, Roger and Blatt, Morton, *Adjustable Speed Drives in Residential Appliances*, White Paper, EPRI for CEC/PIER 500-10-022. Costs of ASD controlled central HVAC systems are approximately $1000 to $2000 per ton compared to $500 to $700 per ton for single speed equipment.

It is essential to align the benefits of ASDs in residential HVAC equipment with the features and benefits sought by users and other market participants.[26] Potential ASD benefits translate to lower energy bills, better controllability, closer temperature control, minimal changes in noise level, overall smoother operation, and greater equipment reliability. All these attributes align with desirable characteristics sought by residential consumers.

Alternative Refrigerants

The refrigerant typically used in heat pumps in the past, R-22, is being phased out because of its ozone depletion potential (ODP). The most common alternative to R-22 in residential space conditioning applications is R-410A, a blend of R-32 and R-125. While R-410A has zero ODP, it actually has a higher global warming potential (GWP) than R-22. R-410A has a GWP of about 2,100 and R-22 has a GWP of about 1,800.[27] Replacements are therefore being sought for R-410A, with one of the most promising replacement options being R-32 with a GWP of 675, albeit with mild flammability. R-32 also offers potential performance improvements compared to R-410A. The current study will investigate the use of R-32 for improving performance and environmental effect of the Next-Gen RSCS.

Work at Daikin showed that R-32 improved COP by 5 percent at 95°F (35°C) and 8 percent at 131°F (55°C).[28]

Work organized by Oak Ridge National Laboratory, under the Air-Conditioning, Heating, and Refrigeration Institute Alternative Refrigerants Evaluation Program (AHRI AREP), tested R-410A alternatives. R-32 appeared to be the most promising.

- Testing at Lennox[29] on a unit optimized for operation with R-410A showed comparable SEERs for R-32 and R-410A with higher HSPF, similar heating capacity, slightly lower cooling efficiency and higher cooling capacity for R-32.

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[26] December 13, 2013, Blatt, Morton, Using Advanced Electronics to Save Energy in Consumer Electronics and Motorized Appliances, Technology Transfer, Presentation at Second PAC Meeting, Sacramento, CA.

[27] June 27, 2016, Alternative Refrigerants for Heat Pumps, EPRI/Daikin.

[28] June 27, 2016, Alternative Refrigerants for Heat Pumps, EPRI/Daikin. R-32 refrigerant is the most "ready for market" option. Items need to be addressed by the industry: Space Conditioners equipped with R-32 have higher compressor discharge temperature than R-410A. Manufacturers need to design the equipment to handle this. This could be a challenge for "drop-in" type applications; it is not addressed in this CEC project. R-32 is an A2L refrigerant, meaning it has mild flammability. This is one of the items that will have to be addressed by the industry, with increased awareness and training of contractors. It is important to note that there are very few refrigerant options in the <~750 GWP range that are not A2L. Therefore an A2L refrigerant may be an inevitable solution. Also, A2L flammability issues are currently being addressed actively in other regions (Europe and Japan, for instance).

[29] April 24, 2013, Crawford, Crawford and Uselton, Dutch, Test Report # 4, System Drop-in Test of Refrigerant R-32 in Split System Heat Pump (with Addendum), Lennox Industries for AHRI Low-GWP AREP. http://www.ahrinet.org/App_Content/ahri/files/RESEARCH/AREP_Final_Reports/AHRI%20Low-GWP%20AREP-Rpt-004_with%20addendum.pdf.

- Testing at Goodman,[30] following AHRI 210-240 and ASHRAE 37, showed a 1.3 percent improvement for R-32 compared to R-410A, for Cooling Test A, 1.2 percent improvement for Cooling Test B, and 2.2 percent improvement for Heating Test H1.

Earlier testing work at the University of Maryland[31] found that R-32 improved cooling capacity between 3.4 percent and 9.7 percent, and COP was improved between 2.0 percent and 9.0 percent compared to an identical conventional vapor compression cycle using R-410A. R-32 was deemed to be an excellent alternative to R-410A that could be further enhanced by component optimization.

In summary, using R-32 as an alternative to R-410A appears to offer some performance improvement potential and clear environmental benefits with lower GWP. The biggest downside of using R-32 is its flammability. Consumer concerns and pending global actions with environmental issues may be important enough to warrant favoring R-32 compared to R-410A. Performance improvements are also attributes that could enhance residential customer acceptance. These positives could be balanced by the fear of flammability and corresponding safety and insurance issues.

Dual Fuel

Dual fuel heat pumps use an electric heat pump for cooling and heating and a gas-fired furnace as an alternative heating means and also for heating backup. The use of a furnace and a heat pump as alternative means of heating gives the user the opportunity to select the option with the lowest heating cost. During the heating season, with a typical dual fuel heat pump, the heat pump will be the primary means of space heating, with the gas furnace providing supplementary heat when the heat pump alone cannot satisfy the building heating load. At very low outdoor temperatures it is likely that the furnace will be the most economical heating means.

Calculations can be performed to determine the balance point temperature, that is, the temperature at which the heating load exceeds the heat pump capacity. Above the balance point the heat pump can handle the entire load and no backup is required. Below the balance point backup is required and the furnace must supply the shortfall in capacity. Calculations can also be performed to determine the breakeven temperature. Above the breakeven temperature it is more economical to run the heat pump and below the breakeven temperature it is more economical to run the furnace.

Two situations are possible:

- If the breakeven temperature is above the balance point, the heat pump will run above the breakeven temperature and the furnace will run below the breakeven temperature.

[30] December 19, 2012, Li, Hao, and By, Bob Soft-optimized System Test of Refrigerant R-32 in 3-ton Split System Heat Pump, Goodman Manufacturing for AHRI Low-GWP AREP. http://www.ahrinet.org/App_Content/ahri/files/RESEARCH/AREP_Final_Reports/AHRI_Low-GWP_AREP-Rpt-005.pdf

[31] July 2012, Xu, Xing, Hwang, Yunho, and Rademacher, Reinhard, Performance Measurement of R32 in Vapor Injection Heat Pump System, University of Maryland, International Refrigeration and Air Conditioning Conference at Purdue, July 16-19, 2012. http://docs.lib.purdue.edu/iracc/1261/.

- If the breakeven temperature is below the balance point, the heat pump will run exclusively above the balance point; the furnace will provide back-up heat between the balance point and the breakeven temperature; and the furnace will operate exclusively below the breakeven temperature.

Although equipment costs are likely to be higher with dual fuel heat pumps than with equivalent all-electric heat pumps or gas furnace/air conditioner combinations,[32] they afford the consumer the opportunity to select the fuel for lowest heating costs at all outdoor temperatures. If gas or electric prices change, the unit can be reset to maintain this optimal operation. Although California gas prices are currently near their historical lows compared to electric prices, more "normal" prices should result in the dual fuel heat pump providing substantial savings and attractive payback periods compared to all-electric heat pumps or gas furnace/air conditioners. An additional benefit of the dual heat pump for winter peaking utilities is the ability to shed load when the outdoor temperature is low.

Fault Detection and Diagnostics (FDD)

Fault detection and diagnostics (FDD) attempt to monitor HVAC system operation to detect degradation in performance, triggering maintenance or repair diagnostics leading the technician/service personnel to

Fault detection the issue to be remedied. Methods vary on what points to measure and how to interpret the measurements. Typically, methods include direct measurement of temperatures, pressures, and electric power, to identify faults relating to air flow; refrigerant charge; sensor malfunctions; fan, blower and compressor degradation; and other malfunctioning components. Early detection of degradation that triggers maintenance alerts and corresponding responses can prevent system malfunctions and failures. This minimizes wasted energy and loss of functionality and assures the occupant that system reliability and performance are optimized.

Units typically have an interface for the homeowner and for the contractor; both can log in and see how their system has been running. Emails can provide notice of issues/faults with instructions for resolving them as well as the availability of monthly reports and instructions on how to access the reports to determine the efficiency score of the unit. To save time, model number and problem information can be sent to the servicing technician before the visit, permitting him or her to bring the parts likely to be needed for the service call.

Opinions differ on the efficacy of FDD technology. A report issued in May of 2016 cites presentations and discussions on the advantages and disadvantages of HVAC fault detection, diagnosis, and repair/replacement.[33]

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[32] Approximately 10 to 15% more than an all-electric heat pump and 20 to 30% more than a gas-electric unit.

[33] May 2016, Springer, David, *Expert Meeting Report: HVAC Fault Detection, Diagnosis, and Repair/Replacement*, Alliance for Residential Building Innovation (ARBI). http://apps1.eere.energy.gov/buildings/publications/pdfs/building_america/emr-hvac-fault-detection-diagnosis-repair.pdf.

Mowris[34] cited common installation and maintenance conditions that can reduce EER by 10 percent to 55 percent. Some of these problems and degradation found in the study were low air flow (EER reduced by 3 percent to 12 percent), evaporator blockage (EER reduced by 4 percent to 7 percent), duct losses (EER reduced by 7 percent to 36 percent), refrigerant overcharge (EER reduced by 4 percent to 17 percent), and refrigerant undercharge (EER reduced by 4 percent to 56 percent). Maintenance practices are cited and the need for service and equipment improvements noted.

Proctor[35] looked at the effects of refrigerant charge level, failure to remove nitrogen from the refrigerant line, airflow restrictions, and duct leakage as major causes of system performance degradation. A 50 percent reduction in airflow reduced EER by 25 percent. A refrigerant charge that is 70 percent of the recommended charge reduced EER by about 45 percent.

In the meeting cited in the May 2016 report, others indicated that field evaluation and repair of refrigerant charge can be detrimental to system performance. Iain Walker (Springer 2016) suggested that it may be wiser to ensure that systems are correctly charged initially and leave them alone. Field studies have shown that a sample of systems that were routinely serviced had more refrigerant charge defects and problems with non-condensables than a sample that had no maintenance. Robert Mowris stated that "Every tech introduces non-condensables," and their presence is difficult to diagnose or measure. Multiple studies have evaluated potential and realized energy savings, many of which are referenced in a document prepared to support California Title 24 standards for refrigerant charge testing.[36] Beyond this information, there is little data on the cost or cost-effectiveness of FDD and maintenance programs besides what has been determined through utility program evaluations, which so far have been uncertain or mostly unfavorable.[37] New programs such as PG&E's Quality Care may provide better information on the relationship between maintenance cost and savings on a large scale, which is badly needed.

Purdue developed and exercised a method for testing several FDD protocols used in public utility sponsored efficiency programs.[38] Six results were possible in the testing: No Response,

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[34] Mowris, Robert, et al., Laboratory Measurements of Residential HVAC Installation and Maintenance Faults, http://apps1.eere.energy.gov/buildings/publications/pdfs/building_america/hvac_expert_mtg_mowris.pdf.

[35] Proctor, John, What is at Stake? And What Does CheckMe!® Do?, http://apps1.eere.energy.gov/buildings/publications/pdfs/building_america/hvac_expert_mtg_proctor2.pdf.

[36] December 2011"Measure Information Template – Residential Refrigerant Charge Testing and Related Issues." Prepared for the 2013 California Building Energy Efficiency Standards under the California Utilities Statewide Codes and Standards Program, California Energy Commission http://www.energy.ca.gov/title24/2013standards/prerulemaking/documents/current/Reports/Residential/HVAC/2013_CASE_R_Refrigerant_Charge_Testing_Dec_2011.pdf.

[37] December 29, 2010, Hunt, M., K. Heinemeyer, and M. Hoeschele, HVAC Energy Efficiency Maintenance Study, Project report prepared by Davis Energy Group for Southern California Edison. http://www.calmac.org/publications/HVAC_EE_Maintenance_Final.pdf .

[38] February 18, 2014, Braun, James and Yuill, David, Evaluation of the Effectiveness of Currently Utilized Diagnostic Protocols, Herrick Laboratories, Prepared for Portland Energy Conservation,Inc. http://www.

Correct, False, Alarm Misdiagnosis, Missed Detection, and No Diagnosis. Results were surprisingly poor. Protocols suffered from very high False Alarm rates (60 percent to 100 percent overall, with most categories more than 95 percent), high Misdiagnosis rates, and high No Diagnosis rates. The Missed Detection rates were low, suggesting that the protocols may be too sensitive. FDD provides no benefits if faults are not addressed correctly. Handheld FDD is a tool intended to help maintenance personnel perform better service than they could with other methods. If they experience and identify False Alarms, Missed Detections, Misdiagnoses, and No Diagnosis cases, it seems probable that they'll soon abandon diagnostics, or ignore them if FDD use is mandated.

Mowris[39] presented laboratory test results of a new 3-ton split-system 13-SEER air conditioner using R-22 refrigerant. The combination of multiple faults such as low airflow, undercharge, duct leakage, and condenser coil blockage reduced EER by 54 percent and SEER by 67 percent.

From these studies it is clear that the potential benefits of timely detection and correction of faults can provide substantial energy savings and improvements in functionality. It remains to be seen whether the FDD devices and algorithms currently on the market and the procedures used to correct detected faults provide possible benefits.

Reduced Duct Losses

Duct losses in an HVAC delivery system are attributable to two major sources: duct leakage and heat transfer between the conditioned air in the duct and the unconditioned space surrounding the duct. Placing ducts in the attic, for example, increases heat transfer losses, while placing ducts in the conditioned space minimizes these losses. This project focused on analysis of heat transfer losses.

With variable speed, duct losses can reduce some of the benefit of improved system efficiency at part load. If the airflow is reduced during part load variable speed operation the stay time of the air is increased compared to meeting the load with full capacity and flow modulating using on-off operation. This could result in increased heat transfer between the conditioned air and unconditioned space.

A short U.S. DOE article on mini-split heat pumps[40] claims that duct losses can account for more than 30 percent of energy consumption for space conditioning, especially if the ducts are

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performancealliance.org/Portals/4/Documents/HVAC%20Research/EffectivenessOfFDDProtocols-Purdue-2014-02.pdf.

[39] June 2012, Mowris, Robert; Jones, Ean; Eshom, Robert, Laboratory Measurements of HVAC Installation and Maintenance Faults, ASHRAE Transactions, 2012 Vol 118 Issue 2, pp. 165-172. http://web.a.ebscohost.com/abstract?direct=true&profile=ehost&scope=site&authtype=crawler&jrnl=00012505&AN=83754907&h=xbebG19L7KqwFwTQ1DEqauODPWFFmjkCmcEnLTmQr5bz%2flw1noDKZ3mF7QnpamAB110%2bgs6BjNKzKeOfpICHxg%3d%3d&crl=c&resultNs=AdminWebAuth&resultLocal=ErrCrlNotAuth&crlhashurl=login.aspx%3fdirect%3dtrue%26profile%3dehost%26scope%3dsite%26authtype%3dcrawler%26jrnl%3d00012505%26AN%3d83754907.

[40] Downloaded July 13, 2016, Ductless, Mini-Split Heat Pumps, U.S. Department of Energy. http://energy.gov/energysaver/ductless-mini-split-heat-pumps.

in an unconditioned space such as an attic. A U.S. Environmental Protection Agency (U.S. EPA) article on duct sealing states that in typical houses about 20 percent of the energy that passes though ducts is lost to leaks.[41]

In 2008, Louisiana Tech University used a protocol it developed for measuring and estimating return air leakage and average duct leakage for homes sampled in its study. Return leaks were found to be 28 percent of total duct leakage. They found average percent cooling and heating energy waste due to duct leakage for an individual home to be 30 percent.[42]

Tests conducted by the University of Illinois in 53 residential buildings included both site-built and manufactured housing to assess the distribution efficiency of forced air delivery systems.[43] The distribution efficiency, defined as the ratio of the energy required to heat the building if there were no duct losses to the actual heating energy required, ranged from less than 50 percent for homes with disconnected ducts to more than 90 percent for well-sealed and insulated systems. Duct retrofits were also performed at 20 of the test sites and, following the retrofits, on average, the homes required 16 percent – 17 percent less heating energy. These results showed that residential distribution system losses can be responsible for substantial energy loss and that duct retrofits are a viable energy conservation strategy for homes with distribution systems located outside of the conditioned space.

Work at LBNL[44] showed the importance of duct leakage on HVAC system efficiency and the efficacy of an aerosol sealing technique. Heating and cooling energy savings from duct sealing in residences are approximately 10 percent in basement ducts and 15 percent to 20 percent in attic ducts, with a 25 percent improvement in peak demand for attic ducts.

Aerosol sealing technology does not coat the ducts, does not require duct cleaning before application, uses a safe vinyl polymer, has no odors or outgassing, lasts up to 10 years, seals holes up to one-half inch across, and remains pliable.

A 2005 ASHRAE article by Modera[45] discusses data collected from thousands of houses around the country. Pressure tests in Sacramento, Calif., and Austin, Texas (regions that are

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[41] February 2009, *Duct Sealing*, EPA 430-F-90-050. https://www.energystar.gov/ia/products/heat_cool/ducts/DuctSealingBrochure04.pdf.

[42] 2008, Vitriol, N.M., Cost Due to Duct Leakage; Return Duct Leakage Vs. Supply Duct Leakage; and Sealing Energy Ductwork, Thereby Reducing Energy Usage In Existing Residential Buildings, Prepared by Louisiana Tech under Louisiana DNR Contract No. 2030-04-03 FINAL REPORT. http://dnr.louisiana.gov/assets/docs/energy/programs/residential/La._Tech_DNR_Final_Report_2008-06-09.pdf.

[43] June 6, 2006, Francisco, Paul W., *Measuring residential duct efficiency with the short-term co-heat test methodology*, University of Illinois, Energy and Buildings 38 (2006) 1076–1083. http://citeseerx.ist.psu.edu/viewdoc/download?doi=10.1.1.503.4280&rep=rep1&type=pdf.

[44] April 21, 2006, Modera, Mark, Remote Duct Sealing in Residential and Commercial Buildings: "Saving Money, Saving Energy and Improving Performance," Lawrence Berkeley National Laboratory. http://energy.gov/sites/prod/files/2015/08/f25/LBNL_Duct_Sealings.pdf.

[45] March 2005, Modera, Mark, *ASHRAE Standard 152 & Duct Leaks in Houses*, ASHRAE Journal. http://www.aerosealcanada.com/wp-content/uploads/2013/10/ASHRAE-Article-Do-Residental-Duct-Leaks.pdf.

dominated by attic duct installations of flex-duct systems) Show two interesting points: (1) between 60 percent and 85 percent of the houses in these regions would benefit from duct sealing, and (2) the distribution of results is remarkably similar in the two regions (there was similar consistency between dealers in each region). Efficiencies for a duct system with R-4.2 duct insulation and the leakage levels observed in the field for existing houses (15 percent supply, 15 percent return) range from 53 percent for cooling under design conditions to 75 percent for the average value over the heating season. Sealing the duct leakage has the largest effect, and that the combination of sealing and super-insulation of the ducts (for example, burying them in loose-fill insulation) can bring all efficiencies to more than 90 percent.

Integrated Ventilation Control

A heat-recovery ventilator (HRV) consists of a heat exchanger, one or more fans to move the air, and controls. The units are relatively simple to install in whole house ventilation systems and save energy in both the heating and cooling modes.[46] The unit typically transfers heat between the exhaust air and the incoming air, heating the incoming air in the heating mode and cooling it in the cooling mode. Savings in energy due to heat recovery must be balanced against the fan power used in the process.

The incoming air supply may also have a bypass fitted to it so that on summer days when it's cooler outside than in, cold outside air can be channeled straight into the home without meeting outgoing air (much like opening a sash window).[47]

The energy savings and cost-effectiveness potential of HRVs were evaluated by Pacific Northwest National Laboratory (PNNL)[48] using U.S. DOE's cost-effectiveness method[49] and energy prices and escalation rates from the U.S. Energy Information Administration's (EIA's)

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[46] Downloaded August 8, 2016, *Whole House Ventilation*, U.S. Department of Energy, Office of Energy Efficiency and Renewable Energy. http://energy.gov/energysaver/whole-house-ventilation.

[47] April 26, 2016, Woodford, Chris, *Heat Recovery Ventilation*. http://www.explainthatstuff.com/heat-recovery-ventilation.html.

[48] December 18, 2015, Taylor, T., Mendon, V., and Zhao, M., Cost-Effectiveness of Heat Recovery Ventilation, Proposal from Pacific Northwest National Laboratory to DOE. https://www.energycodes.gov/sites/default/files/documents/iecc2018_R-3_analysis_final.pdf.

[49] *DOE Cost-Effectiveness Methodology*. https://www.energycodes.gov/development/residential/methodology.

Annual Energy Outlook.[50] HRVs have a sensible heat recovery efficiency of 70 percent to 80 percent.[51] The analysis conservatively assumed a sensible heat recovery efficiency of 70 percent. The energy analysis indicated that HRVs yield about 10 percent energy cost savings for the total International Conservation Energy Code-regulated end-uses (heating, cooling, lighting, and water heating) in colder climate zones (6, 7, and 8) with higher savings achieved in the coldest climate zones.

Use of a residential economizer was shown to save energy and provide acceptable indoor humidity in hot, dry climates and in a marine climate such as San Francisco.[52]

One environmental asset available in some climate zones (CZs) is nighttime diurnal temperature swings. Fan economizers (essentially, whole-house fans connected to a smart thermostat) make use of these temperature differences for cooling and can minimize or eliminate the need for mechanical air conditioning. The occupant sets a thermostat, and when the outside air temperature is below the temperature set point, the economizer is triggered; it pulls air in from outside and flushes out the warm interior air with cooler exterior air, usually during the night. If there is sufficient mass and insulation in the residence, it will cool down and remain sufficiently cool during the following day.[53] In previous studies, it was shown that

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[50] *EIA, Annual Energy Outlook 2015*, table accessed 2 Dec 2015 from http://www.eia.gov/beta/aeo/#/?id=3AEO2015&cases=ref2015 ; nominal 2018 prices.

Fuel	Price (2018$)	Effective[50] Escalation Rate (per year, real)
Electricity	$0.137/kWh	0.69%
Natural Gas	$1.154/therm	1.74%
Fuel Oil	$2.299/therm	1.84%

[51] See *EnergySavers* website http://www.energysavers.gov/your_home/insulation_airsealing/index.cfm/mytopic=11900.

[52] September 2015, Turner, W.J.N/, et al., *Residential Pre-Cooling: Mechanical Cooling and Air-Side Economizers,* LBNL, Draft Report LBNL-180960. https://buildings.lbl.gov/sites/all/files/lbnl180960.pdf.

[53] January 2015, Kensek, K., et al., Economizer Performance and Verification: The Effect of Human Behavior on Economizer Efficiency and Thermal Comfort in Southern California, CH-15-021, ASHRAE Transactions 121 (2015) 241-252. http://search.proquest.com/openview/3cb0c1a37462440eca4c134726a51a45/1?pq-origsite=gscholar.

it is possible in half of California's 16 CZs to design homes that can be comfortable without the need for air conditioners.[54, 55, 56]

A study of ventilation systems in cold climates showed that energy efficiency could be improved by up to 67 percent compared to a traditional exhaust ventilation system by using a heat recovery system with a nominal temperature efficiency of 80 percent.[57]

Comparison between an exhaust fan and a heat recovery ventilation system in a cold climate showed that a heat exchanger air-to-air ventilation system can save up to 2710 kWh per year compared to a traditional ventilation system and space heating.[58]This amounts to an increase in energy efficiency of around 30 percent in an insulated house.

Zonal Control

Conditioning occupied areas of the home, while letting unoccupied areas remain unconditioned, can save energy if the conditioned and unconditioned spaces can be thermally isolated. The space conditioning device needs to permit modulation such as a variable speed device with separate controls and indoor delivery means for each area.

The U.S. DOE estimated that zoned heating can produce energy savings of more than 20 percent compared to heating both occupied and unoccupied areas of a residence.[59]

Foster[60] cited information from a Nuclear Regulatory Commission study using a test house in Maryland showing that zoning could increase cooling energy use (by 20 percent) or reduce total energy use (by 25 percent) depending upon placement of the thermostats and the

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[54] 2010, Milne, Murray and Kohut, Tim, *Eliminating Air Conditioners in New Southern California Housing.* http://www.energy-design-tools.aud.ucla.edu/papers/ASES10-TK-2.pdf.

[55] 2002, Bourassa, N., Haves, P. and Huang, J.Y., *A Computer Simulation Appraisal of Non-Residential Low Energy Cooling Systems in California*, LBNL-50677. http://eetd.lbl.gov/sites/all/files/publications/50677.pdf.

[56] February 2004, Fisk, W. J., Seppanen, O., Faulkner, D., and Huang, J. *Economic Benefits of an Economizer System: Energy Savings and Reduced Sick Leave* LBNL-54475. http://eetd.lbl.gov/sites/all/files/publications/lbnl-54475.pdf.

[57] 2003, J. Jokisalo, J. Kurnitska, and A. Torkki, *Performance of balanced ventilation with heat recovery in residential buildings in cold climate*, International Journal of Ventilation, 2003, Vol. 2. http://www.aivc.org/resource/performance-balanced-ventilation-heat-recovery-residential-buildings-cold-climate.

[58] 1986, D. Hekmat, H.E. Feustel, and M.P. Modera, *Ventilation Strategies and their Impacts on the Energy Consumption and Indoor Air Quality in Single-Family Residences*, Energy and Buildings, Vol. 9, p. 239-251, 1986. http://www.sciencedirect.com/science/article/pii/0378778886900241.

[59] Accessed July 12, 2016, *Electric Resistance Heating*, DOE http://energy.gov/energysaver/electric-resistance-heating

[60] January 22, 2015, Foster, Richard, *Create Energy Saving Comfort and Cut Costs with Affordable Zone Controls*, ZONEFIRST, 2015 NAHB International Builder's Show, Las Vegas. http://www.nahbclassic.org/assets/docs/ises/17424CreateenergySavingComfort_20150112084655.pdf.

amount of overheating or overcooling that occurred before zoning. He also referenced surveys that have shown that 60 percent of customers would pay $1200 for zoning if it were offered.

In studies of advanced occupancy sensing in commercial buildings, PNNL found that occupancy sensing saved 17 percent to 23 percent of the lighting and HVAC energy use, with HVAC accounting for around 90 percent of these savings.[61]

Work at the University of Florida, presented in the ASHRAE Transactions,[62] showed that residential energy consumption could be reduced 16 percent to 25 percent using zoning with a two-speed compressor, variable speed blower, dampers, and zone thermostats.

Auto Demand Response

Reducing demand is a major focus of utility operations, particularly during summer periods induced by hot weather. Enlisting residential customers to opt for programs that permit the utility to limit their demand during these periods can be one element of their strategy. The versatility of the Next Gen Residential Space Conditioning System provides tools to help entice residential customers to opt for demand-limiting programs while maintain a reasonable level of comfort. The variable speed capability of the Next Gen Residential Space Conditioning System enables more efficient operation at part load. This affords the opportunity to reduce the demand of the HVAC unit more than the capacity reduction of the unit. In other words, if the utility sends out an auto demand response (ADR) signal to reduce demand by 50 percent, the unit can operate at this demand level while providing more than 50 percent of the capacity.

Some demand response (DR) controllers take into account home characteristics, personal preferences, occupancy, and weather to maximize load shed and comfort. The controller can learn from each data point for each home, minimizing the discomfort of the residents while maximizing the demand savings, taking advantage of unoccupied periods, pre-cooling, and the need for occupant control.

Benefits Summary

Table 52 and the following paragraphs summarize some of the benefits and disadvantages of the Next Gen Residential Space Conditioning System features. The final paragraphs of the Market Participants section of this chapter discuss how these attributes might be valued by key market participants.

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[61] January 2013, Zhang, Z., et.al, *Energy Savings for Occupancy Based Control (OBC) of Variable Air-Volume (VAV) Systems*, Pacific Northwest National Laboratory, PNNL-22072. http://www.pnnl.gov/main/publications/external/technical_reports/PNNL-22072.pdf.

[62] 1991, Oppenheim, P., *Energy Implications of Blower Overrun Strategies for a Zoned Residential Forced-Air System*, University of Florida, ASHRAE Transactions, Vol. 97, Part 2. http://www.energy.ca.gov/title24/2013standards/prerulemaking/documents/current/Reports/Nonresidential/HVAC/Attachment%201_Blower%20Overrun%20Study.pdf.

Table 52: Potential Benefits and Disadvantages of Next Gen System Features

Feature	Benefits	Liabilities
Variable Speed	Greater comfort, lower energy costs, improved noise, smoother operation, possible reliability improvements	Higher equipment cost and complexity, harmonics
Alternative Refrigerants	Lower GWP, performance improvement and lower energy cost	Possible higher equipment and servicing cost, flammability issues, possible higher insurance premiums
Dual Fuel	Lowest possible heating costs based on fuel selection	Higher equipment costs
Fault Detection and Diagnostics	Improved maintenance and reliability, lower energy costs	Higher cost of control systems, possibility of unnecessary service calls
Reduced Duct Losses, Improved Delivery Effectiveness	Lower energy costs	Cost of sealing and insulating ducts
Integrated Ventilation Control	Lower energy costs	Higher equipment costs
Zonal Control	Better controllability, greater comfort	Higher equipment costs
Auto Demand Response	Lower demand charges, greater comfort	Higher equipment costs

Source: EPRI

Market Participants and Influences/Attributes and Future Actions

Several parties finance, specify, install, operate, and use the Next Gen Residential Space Conditioning System in the market. Table 53 lists these market participants/market actors and their functions, as well as market barriers that could impede the Next Gen Residential Space Conditioning System and suggested generic actions/interventions that could overcome these barriers.

Each market participant (and their functions) interacts and overlaps to the extent that one function/participant can influence another. For example, a product that is not easy to use, operate, or maintain will not be favored by users. This information is likely to be passed on to specifiers who are consequently not likely to specify the product for future installations.

Each market participant has particular needs that must be satisfied if a new technology is to be specified, financed, installed, operated, and used. Most technology product attributes are universally required across all of the functions/chains but some are particularly critical to a subset of the market participants. Information products need to focus their contents on the attributes most important to the market participants or chain of market participants they are designed to address.

Funders, for example will be interested in cost effectiveness, particularly first cost, as well as cost/energy savings. For example, utilities offering rebates or other incentives to new technologies will want to know how the technologies save energy compared to conventional systems. Utility incentive programs in the future will be essential, for successful market transformation of this technology.

Specifiers will be interested in cost effectiveness, and how well the system performs compared to specifications. Reduced energy use, improved user comfort and reliable operation will also be of great interest to this market function/chain. The more technically oriented participants in the "specifier" group will need information on how well the technologies performed in terms of delivering their expected advantages. Similarly, important is how well perceived disadvantages, such as high first cost, have been overcome. The information products that are focused on influencing this group, therefore will pay particular attention to documenting proven performance advantages and how perceived disadvantages have been and can be overcome.

Suppliers need to see or anticipate demand for their product. Information that provides valid performance evaluation, delivered through the most effective channels to specifiers and others in the supplier chain, will encourage dealers and distributors to stock the product, and manufacturers to produce it.

Installers would be most receptive to receiving application guidelines showing the simplest and most effective techniques for installing the new technologies and for assuring that they are operating correctly.

Maintenance/service personnel need information on FDD operation as well as maintenance procedures that could be used to keep the systems working effectively, assuring their continued high performance.

Users of the technology such as consumers will be concerned with comfort, ease of operation, safety, temperature and humidity control, and acoustics. A fact sheet/tech brief for users will help them understand the virtues of the Next Gen Residential Space Conditioning System and will help them operate the system correctly, assuring that the advantages are consistently realized.

Codes and standards setting bodies need information on the performance of new technologies to assure that they adhere to existing requirements. If the new technologies do not comply with existing requirements but nevertheless satisfy the objectives of the codes or standards, then this information should be clearly provided. This will enable the language of the regulation be changed to permit effective usage of the Next Gen Residential Space Conditioning System.

Valuing Next Gen System Features

Potential advantages and disadvantages of the Next-Generation Residential Space Conditioning System features are presented in Table 53. Comparing these advantages and disadvantages to the drivers important to each type of market participant provides important insight into how to transfer Next Gen technology information to each type of market participant.

For example, indoor environment is most important to HVAC system **users**, therefore features providing improved comfort, better acoustics and improved reliability and maintenance should be attributes that users would favor in making product selection decisions. Knowing this and looking at Table 52, the features that should be of interest to users in making these decisions should include variable speed, auto demand response, FDD and zonal control. Marketing materials focused on the user should stress these features and their attributes. Other issues such as environmental improvement and energy cost may be important to some users and should therefore be included in the marketing materials, but perhaps to a lesser extent. Higher equipment costs will be an initial negative driver for the consumer and will need to be overcome with clear information on cost effectiveness (payback or life cycle cost information).

Similar reasoning for **codes and standards officials** would result in stressing features that provide cost effective energy and demand savings and thus could include all the features being tested with the possible exception of dual fuel.

Service personnel would be most interested in the FDD feature of the Next-Generation Residential Space Conditioning System and thus should be made aware of this feature in material provided to this element of the market chain.

Cost effectiveness is likely to be most important to many market participants such as specifiers, funders, codes and standards officials (as mentioned above) and suppliers as well as to a lesser extent, users. It is important, therefore to determine the equipment costs and operating savings accruing to each of the Next-Generation Residential Space Conditioning System features tested and to determine the payback/life cycle cost effectiveness for each feature. It is expected that such features will be developed in future activities, which are beyond the scope of this project.

Table 53: Market Influences on the Next-Generation RSCS System in California

Market Actor	Market Functions	Market Issues	Future Interventions/Stimuli
Federal Government	Set codes and standards Provide funding for research and development Provide information, guidance and oversight in planning, design, funding, building, improving and maintaining facilities and equipment	Aversion to risk Long budgeting cycle Need assistance in developing and disseminating information	Documented performance/case studies assuring code compliance or presenting information to alter codes Documented performance/case study information demonstrating the life cycle cost effectiveness of the new technologies A range of information products to permit dissemination of training, guidance and oversight to constituents
State Governments	Set codes and standards Provide funding for research and development Provide information, guidance and oversight	Aversion to risk Long budgeting cycle Need assistance in developing and disseminating information Inconsistencies among state requirements (for example, differing refrigerant phase down dates)	Documented performance/case studies assuring code compliance or presenting information to alter codes Documented performance/case study information demonstrating the life cycle cost effectiveness of the new technologies A range of information products to permit dissemination of training, guidance and oversight to constituents
Local Government	Influence/set codes and standards, zoning, and permitting provisions	Aversion to risk, need for proven performance, cost effectiveness	Timely presentation of documented performance/case studies assuring code compliance or presenting information to alter codes Documented performance/case study information demonstrating the life cycle cost effectiveness of the new technologies

Market Actor	Market Functions	Market Issues	Future Interventions/Stimuli
Utility	Provide funding for efficient equipment installations Provide information, guidance, and oversight Provide financial incentives	Need proven performance to determine product eligibility Need assistance in developing and disseminating information	Documented performance/case study information demonstrating the life cycle cost effectiveness of the new technologies A range of information products to permit dissemination of training, guidance and oversight to constituents Successful implementation of utility incentive programs
Construction Manager	Construction, installation oversight Program management	Risk of nonperformance when using outside contractors and new technologies	Documented performance/case study information demonstrating the life cycle cost effectiveness of the new technologies Information on field-proven performance including design, installation and operation information for the new technologies
Architect	Facility appearance Follow codes and standards Design the space to achieve functionality Establish budget, specifications	Risk of nonperformance when using new technologies Often measured more on low first cost, (adherence to schedule and budget) than on reduced operating costs and performance improvements	Documented performance/case study information demonstrating the life cycle cost effectiveness of the new technologies Information on field-proven performance including design, installation and operation information for the new technologies

Market Actor	Market Functions	Market Issues	Future Interventions/Stimuli
Engineer/ Designer	Design systems to meet loads, space restrictions, and architectural requirements Design systems to comply with fire safety, energy and indoor air quality codes Specify systems that can be built within budget and installed by local trades	Risk of nonperformance when using new technologies May lack experience with the design and installation of new technologies Often measured more on low first cost (adherence to schedule and budget) than on reduced operating costs and performance improvements	Documented performance/case study information demonstrating the life cycle cost effectiveness of the new technologies Information (guidelines, presentations and training) on field-proven performance including design, installation and operation information for the new technologies Performance information on existing installations Analysis tools that account for advanced system attributes
Professional Society/Trade Association	Set test procedures and codes and standards Help promote/lobby for new technologies and products	Assurances that new technologies meet code requirements	Documented performance assuring that the objectives of the code requirements are being met
Environmental Advocates	Environmentalists and environmental organizations that evaluate and help promote technology and products that mitigate climate change and environmental effects	Legislative regulatory and other policy developments provide both mandates and incentive related programs	Documented Performance/case study information demonstrating the life cycle cost effectiveness of the new technologies
Contractor	Responsible for installation, adhering to designer's specifications, budgets, schedule and complying with applicable codes/regulations	Risk of nonperformance when using new technologies May lack understanding of installation requirements of new technologies	Documented performance/case study information demonstrating the life cycle cost effectiveness of the new technologies Information (guidelines, presentations and training) on field-proven performance including design, installation and operation information

Market Actor	Market Functions	Market Issues	Future Interventions/Stimuli
Manufacturer	Production (supply), pricing, and marketing	Want products with high market volume, proven performance Concerned with production volume, margins	Provide information on the advantages of the products, market need and available/anticipated support for the product
Distributor	Intermediary in the supply chain Distributors will stock products that have high demand	Want products with high market volume, proven performance	Provide information on the advantages of the products, market need and support for the product
Dealer/Retailer	Supplier selling to installer Wants to sell products in stock and products with high sales volume	Want products with high market volume, proven performance Concerned with complexity of installation, servicing, reliability	Provide information on the advantages of the products, market need and support for the product
Maintenance, Service	Understanding maintenance and repair issues, permitting ease of operation, and high reliability	Lack of expertise with new technology Lack of time to address problems Wary of sophisticated equipment and concurrent installation and maintenance requirements Operation/maintenance differs from conventional systems	Documented performance/case study information demonstrating the life cycle cost effectiveness of the new technologies Information (guidelines, presentations and training) on field-proven performance including design, installation and operation information for the new technologies. Information on equipment maintenance, servicing, control and operation
Consumer	User desires comfortable indoor environment (temperature, humidity, fresh air, good acoustics), attractive appearance, reasonable cost, and is concerned with environmental issues	Dependent on dealer/ retailer to provide a healthy, comfortable environment Fear of the unknown Cost, reliability, comfort	Fact sheets, information in newspapers, or other vehicles explaining the concepts of the Next Gen Residential Space Conditioning System and their advantages

Source: EPRI

Barriers to Commercialization

This section characterizes market barriers affecting adoption of Next Gen System technologies and approaches to overcoming these barriers.

Specifiers and other decision makers tend to favor systems and technologies that have performed well for them in the past. As such there often exists a very substantial resistance to change. Information needs to be provided to overcome this resistance to change, to minimize performance uncertainties and to reduce the extra effort and consequent cost of deploying new technologies.

Impediments to implementation of new technology have been characterized as market barriers in the market transformation literature. Market barriers can be overcome with suitable market interventions resulting in the desired market influence.

Table 54 lists traditional market barriers, as outlined originally by LBNL[63] and amplified by EPRI[64] in work for the CEC. Accompanying each of the market barriers are possible approaches suggested for addressing that barrier. Missing from the barriers presented in the LBNL report are cost-related impediments. The reasoning by LBNL is that cost in itself is not a barrier, it is an effect. Effectively overcoming the barriers listed in Table 54 will result in reduced transaction costs, increased market penetration and resultant reduced cost of production and lower prices. LBNL implies that rebates or other financial incentives do not have a lasting effect on the market.

Recent evidence shows that rebates may indeed have a transformative effect on the market, increasing customer acceptance, manufacturer volume and resulting in economies of scale. These effects can be sustainable even after rebates have been removed. Thus high first cost is included in Table 54 as a barrier and rebates and other financial incentives are offered as possible approaches to overcoming this barrier.

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[63] Items A to N are from Eto, J.; Prahl, R.; and Schlegel, J., *A Scoping Study on Energy-Efficiency Market transformation by California Utility DSM Programs*, LBNL-39058, UC-1322, July 1996

[64] January 10, 2014, Blatt, Morton, *Using Advanced Electronics to Save Energy in Consumer Electronics and Motorized Appliances,* Task 7 Technology Transfer Plan and Execution, EPRI for the California Energy Commission

Table 54: Market Barriers to Adoption and Approaches to Reducing Barriers

Market Barriers	Approaches to Reducing Barriers
Information or search costs in locating and understanding the Next Gen System	Next Gen System characteristics and performance; for specifiers and installers
Performance uncertainties – understanding and believing Next Gen performance	Case study information documenting performance. Testimonials from colleagues with similar requirements
Asymmetric information and opportunism – sellers of the Next Gen System know more than buyers. Obtaining equivalent information may be costly or impossible for the buyer.	Information, training and technical assistance on ownership costs, energy, performance, and other attributes, to assist the buyer/specifier in making selection/purchase decisions.
Hassle or transaction costs – costs of acquiring the Next Gen Residential System	Information on how to specifying, buy and install the product.
Hidden costs – unexpected operation, monitoring, servicing and maintenance costs	Information for operating, monitoring, servicing and maintaining the product for maintenance personnel
Access to financing – lack of recognition of life cycle cost savings, lack of accounting for energy efficiency, environment in borrowing costs	Information for Energy Service Companies (ESCOs), specifiers, sources of funding, on the economic benefits of the Next Gen System
Bounded rationality – use of rules of thumb (such as a two-year payback period) that limit the scope of consideration for a given decision	Information documenting cost effectiveness and other advantages to overcome rules of thumb that might otherwise inhibit consideration
Organization practices – rules, policies and practices that make it difficult to adopt new technologies	Case study information focused on economic, health and productivity benefits that overcome the bureaucratic tendency to be risk averse.
Misplaced or split incentives – institutional relationships where the person deciding on adopting a new technology is not the person who benefits from it	Case study information to help the specifier understand the energy, health and productivity benefits of the new technologies
Product or service unavailability – higher prices or unavailability because of newness	Aggressive publicity and market interventions, creating demand for product, reducing prices
Externalities – costs (such as environmental costs) not reflected in the price	Information on the how Next Gen Residential Space Conditioning technologies reduce energy use and thus reduce CO_2 production and global warming
Non externality mispricing – other factors moving price away from cost, such as ratemaking based on average costs	Provide information to convince the local utility to provide training and financial incentives that reflect the economics of "integrated resource planning."

Market Barriers	Approaches to Reducing Barriers
Inseparability of product features – desirable features are coupled with features not of interest	Explain the benefits of each of the features of the Next Gen Residential Space Conditioning System
Irreversibility – purchase decisions are irreversible to the extent that a purchased product has a usable life and is not likely to be replaced during that useful life	Can the technologies be deployed in retrofit situations? If retrofits are possible, provide clear instructions on modifying the building for these retrofits and how to install the product.
High purchase price – immature product using new technology/components and low initial manufacturing volume results in high product cost and high purchase price	Rebates or other financial incentives can reduce product acquisition cost for the user and bolster manufacturer volume to enhance economies of scale.
Codes and standards – existing codes and standards may not properly account for the attributes of the Next Gen Residential System making it difficult to favorably deploy the System in its most suitable applications	Document performance of the Next Gen System, providing evidence that a waiver or alteration is required in the existing codes and standards. Prepare, package and present information to influential individuals and organizations to affect desired codes and standards changes.

Source: EPRI

Technology Transfer Activities

Most of the approaches to overcoming market barriers involve providing information to market participants on the project, its results, and its potential and implications for the industry at large as well as to California's energy savings goals and California ratepayers in general. Therefore, the crux of transferring this technology to the market at large involves individual and group contact and publicity in the forms of presentations at conferences, trade shows, and symposia; development of collateral materials, articles, and white papers for distribution through a multitude of channels; and personal contact with industry leaders, influencers, and decision makers to compel market adoption and incentives support.

Tables 55 and 56 list key market actors and participants, their mission, and important industry meetings and publications that are the targets and vehicles for conveying project information.

Table 55: Resources for Information Dissemination on the Next-Generation Residential Space Conditioning System

Market Actor	Organization/Trade Association/Professional Association	Key Meetings	Key Periodicals/ Publications
Federal Government	US EPA (United States Environmental Protection Agency) U.S. EPA Environmental Information Center 75 Hawthorne Street, 13th Floor San Francisco, CA 94105 415 947 8000 or 866 EPA-WEST toll free 866 372 9378 r9.info@epa.gov Energy Star Media Inquiries media@energystar.gov Bill Keener, keener.bill@epa.gov, 415-972-3940	Energy Star partner products meetings Webinars	Energy Star Publications, including Residential HVAC Equipment, Brochures Newsletters
	US DOE (United States Department of Energy) http://www.eere.energy.gov/ Media Inquiries EE.Media@ee.doe.gov Homes HVAC Buildings 202-586-9127 Buildings@ee.doe.gov BTP Webmaster webmasterbtp@nrel.gov Appliances and Commercial Equipment Standards EERE_ACES@ee.doe.gov Partners in Energy Star (see EPA)	Events	Building technology publications, webinars, software
	USDOE/Building America Eric Werling, Program Director eric.werling@ee.doe.gov Contact at Building America Solutions Center: basc@pnnl.gov	News and Events Meetings and Webinars Expert Meetings	Tools and Resources Publications Library

Market Actor	Organization/Trade Association/Professional Association	Key Meetings	Key Periodicals/ Publications
State Government	CEC (California Energy Commission) Sets standards for energy use in the State, Title 24 Standards for Buildings Title24@energy.state.ca.us Title 20, California Code of Regulations appliances@energy.ca.gov Research and Development Programs EPIC Program jackson.thach@energy.ca.gov	Technical Advisory Committee Meetings for CEC-EPC-14-021 Calendar for Hearings for Title 24 Opportunities for participation in Title 20 pending decisions Workshops and Meetings	R&D Reports and Publications Project EPC-14-021 Tech Briefs, Fact Sheets and other publications Existing and Pending Title 24 Standards Existing Title 20 Standards Title 20 2016 Rulemaking CEC Docket List
	The California Building Standards Commission (CBSC) administers the development, adoption, approval, publication, and implementation of California's building codes. 916 263-0916 cbsc@dgs.ca.gov	Calendar and Meetings	Building Standards bulletins Publications Educational Publications/ Guidebooks Title 24 Building Codes
	California Air Resources Board (ARB) is the agency tasked in California with coordinating state climate change initiatives, as well as the state's carbon cap-and-trade and refrigerant programs.	Rulemaking activities and Board meetings	Resource Directories, including fact sheets, test methods, and research
	The California Public Utilities Commission oversees California's investor-owned utilities and utility rate, incentive and demand response programs.	Meetings and events calendar	Current Publications

Market Actor	**Organization/Trade Association/Professional Association**	**Key Meetings**	**Key Periodicals/ Publications**
	Department of General Services Programs and Services DGS helps to better serve the public by providing a variety of services to state agencies through procurement and acquisition solutions. DGSPublicAffairs@dgs.ca.gov	Buying Green: California's Guide to Sustainable Purchasing News and Events	Buyer's Guide Suppliers Guides
State Government, *continued*	Division of State Architect DGSPublicAffairs@dgs.ca.gov Acts as policy leader for building design and construction, and provides design and construction oversight for State facilities Chester.wisdom@dgs.ca.gov	News and Events Lists various board meetings dealing with safety, compliance, design, policy and procedures and standards	Archived News And Events Publications
Local Government	Numerous, including the City of Los Angeles (and LA Department of Water and Power), Southern California Association of Governments, South Bay Cities Council of Governments, San Diego Association of Governments	Various, including council meetings, hearings, board meetings	
Local Utility	Major utilities assist customers concerned with planning, design, financing and other aspects of facility improvement and operation. Pacific Gas & Electric Energy Center rxmu@pge.com 415-973-2277 Southern California Edison CTAC SCE Residential Programs mediadesk@sce.com Sacramento Municipal Utility District Energy and Technology Center jnewman@smud.org customerservices@smud.org San Diego Gas & Electric Energy Innovation Center 800-411-7343	Training sessions held periodically at utility energy centers in Irwindale, Sacramento, San Francisco and other locations in California PG&E Classes and Seminars SCE Classes and Events SMUD Workshops and Training SDG&E Seminars	PG&E Link to Publications SCE Center Fact Sheet SMUD Saving Energy & Money SDG&E Brochures and Fact Sheets

Market Actor	Organization/Trade Association/Professional Association	Key Meetings	Key Periodicals/ Publications
Architect	AIA (American Institute of Architects) Committee on Education infocentral@aia.org	Events including conference in Portland, May 17-20, 2017	AIA webinars and resources *Building Design and Construction* *Environmental Design and Construction*
Construction Manager	CMAA (Construction Management Association of America) Info@cmaanet.org Representing architecture, engineering, construction and facilities management. John McKeon jmckeon@cmaanet.org 703-677-3631	Events Including Symposia in New Orleans April 2-4, 2017, and Denver, March 18-20, 2018, and Conferences and Trade Shows in Washington, D.C., October 8-10, 2017, and Las Vegas, October 14-16, 2018 Webinars	CMAA Publications
Retailers	National Association of Retail Dealers of America (NARDA) nardasvc@narda.com 312-648-0649	Webinar on Job Rates	Fact Sheet Blue Books, Job Rate Guides Information/Resources
Home Builders	National Association of Home Builders (NAHB) 800-368-5242 info@nahb.org The home building industry's technical info resource	Events and Education Course Calendar	Research

Market Actor	Organization/Trade Association/Professional Association	Key Meetings	Key Periodicals/ Publications
HVAC Manufacturer	AHRI (Air-Conditioning Heating and Refrigeration Institute) ahri@ahrinet.org American Society of Heating Refrigerating and Air-Conditioning Engineers (ASHRAE) ashrae@ashrae.org	AHRI Meetings and Events ASHRAE Meetings and Events **Winter Conferences** and **Annual Conferences**	Low GWP Research Reports Other AHRI Resources ASHRAE Periodicals *ASHRAE Journal* *ASHRAE Handbooks* *Engineered Systems* *HPAC Engineering*
HVAC Engineer/ Designer	ASHRAE (American Society of Heating, Refrigerating and Air-Conditioning Engineers) ashrae@ashrae.org	ASHRAE Meetings and Events Winter and Annual Conferences,	ASHRAE Periodicals *ASHRAE Journal* *ASHRAE Handbooks* *Engineered Systems* *HPAC Engineering* *Consulting-Specifying Engineer*
HVAC Contractor	ACCA (Air Conditioning Contractors of America) glenn.hourahan@acca.org	ACCA News ACCA Events	Indoor Environment and Energy Efficiency *IE3 Media* *Contractor Excellence* magazine *Contracting Business* *The Air Conditioning/ Heating/Refrigeration News*

Market Actor	Organization/Trade Association/Professional Association	Key Meetings	Key Periodicals/ Publications
HVAC Distributor, Dealer	AHRI (Air-Conditioning Heating and Refrigeration Institute) American Society of Heating Refrigerating and Air-Conditioning Engineers (ASHRAE) ACCA (Air Conditioning Contractors of America) The Heating, Air-Conditioning & Refrigeration Distributors (HARDI) hardimail@hardinet.org	AHRI Meetings and Events ASHRAE Meetings and Events ACCA Events HARDI Events (Meetings and Webinars)	ASHRAE Periodicals *ASHRAE Journal* *ASHRAE Handbooks* *Engineered Systems* *HPAC Engineering* ACCA Directories of Contractors, Suppliers HARDI Resources
HVAC Users	ACCA (Air Conditioning Contractors of America) Consumer Reports Provides unbiased testing and evaluation of residential equipment e-mail Consumer Reports	ACCA Calendar of Events Consumers Union Annual Meeting annualmeeting@cu.consumer.org	Air-Conditioning Consumers Information Directories of Contractors, Suppliers Consumer Reports magazine

Source: EPRI

Table 56: Organizations Disseminating General Information on the Next-Generation RSCS

Organization/ Program	Mission	Key Publications	Key Meetings	Key Periodicals
USGBC (United States Green Building Council) LEED for homes leedinfo@usgbc.org	Coalition of building industry leaders to promote cost effective energy efficient green buildings	LEED Residential Space Conditioning LEED Publications Guides to LEED Certification	Greenbuild Int'l Conf./ Expos Future Greenbuild Expos Boston, Nov. 8-10, 2017 Chicago, Nov. 14-16 2018 *LEED Training Workshops* at locations nationwide	Newsletters
Alliance to Save Energy 202-857-0666 Ron Kweller, Director Media Relations 202-530-2203 rkweller@ase.org	Promotes energy efficiency worldwide through research, education and advocacy	State by state information and publications	Key events: Initiatives	Efficiency News
American Council for an Energy Efficient Economy	Catalyst for transfer of ideas between utilities, researchers and energy efficiency practitioners	Publications	Conferences Denver, Aug 15–17, 2017 Arizona, Oct 30–Nov 1, 2017	Newsroom

Organization/ Program	Mission	Key Publications	Key Meetings	Key Periodicals
SBIC (Sustainable Buildings Industry Council) National Institute of Building Sciences rcolker@nibs.org 202-289-7800	Supports technology-driven, energy efficient sustainable design and construction	Resources Reports	Events	Newsletters *Journal of the National Institute of Building Sciences*
Consortium for Energy Efficiency (CEE) 617-589-3949 http://www.cee1.org/ Mark Hoffman Deputy Executive Director mhoffman@cee1.org	Establishes feasible efficiency levels that utilities and others can use for rebates and other incentives	Program Library Residential Heating and Cooling Residential Heat Pumps	Meetings	News Archive
Savings by Design Funded by California utility customers and administered by California's investor owned utilities	Provides services and incentives to help raise energy performance to a top priority for architects and building owners	Resources Links to investor owned utility programs, including rebates, external programs and energy modeling software	Training sessions held periodically at utility energy centers	Links to energy efficient HVAC options from Energy Design Resources

Source: EPRI

Codes and Standards

Reference to codes and standards and their possible influence on design, installation and operation of the Next Gen Residential Space Conditioning System has been made in previous sections. Table 57 shows some of the organizations that deal with codes and standards issues that could impact deployment of the Next Gen Residential Space Conditioning System. Greater insight into the codes and standards issues and the possible ramifications of their application to the Next Gen Residential Space Conditioning System will be important.

At this time, the following issues appear to be most important:

- The most relevant codes and standards that could effectively incorporate the energy efficiency improvements embodied in the Next Gen Residential Space Conditioning System are likely to be found in Title 20 and Title 24. In order to maximize the impact of the improvements found in this Program, proposed changes to Title 20 and Title 24 could be presented at CEC workshops where public comment on rules changes are obtained.
- Guidelines affecting design of the Next Gen Residential Space Conditioning System may be included in ASHRAE Standards. As such, relevant Standards could be identified and studied and modifications, to effectively incorporate key features of the Next Gen System.
- Test procedures affecting the evaluation of the Next Gen Residential Space Conditioning System performance may be included in AHRI Standards. As such, relevant Standards could be identified and studied and modifications, to effectively incorporate key features of the Next Gen System.

Table 57 provides summary information on the organization concerned with codes and standards as they apply to the products being studied in this project. The desirable attributes of the improved product(s) will be initially documented, into the Production Readiness Plans.

Interactions with Codes and Standards Organizations

Improvements in energy efficiency and other attributes found in the current study deserve recognition and may affect current codes and standards. The project researchers will identify appropriate potential changes to the applicable codes and the best manner of affecting code changes, as well as how to present the information and to whom.

Participation with relevant ASHRAE technical committees (TCs) identified in Table 57 could establish connections to facilitate the use of study results to make changes to HVAC equipment standards:

- TC 1.11, Electric Motors and Motor Control, HVAC compressor and fan/blower motors and drives are covered by this committee. Project information needs to be provided to this committee and to the standing committees for Standard 90.2 about effective use of efficient variable speed drives in residential buildings
- TC 3.3, Refrigerants and Secondary Coolants, is concerned with all properties and functions of refrigerants and would therefore be interested in project information concerning R-32 testing.

- TC 5.5 Air-to-Air Energy Recovery, is concerned with air to air heat exchangers such as used for heat recovery in the Next Gen Residential Space Conditioning System and would therefore be interested in project results in this area
- TC 8.11, Unitary and Room Air Conditioners and Heat Pumps, is concerned with factory-engineered vapor compression systems such as the Next Gen Residential Space Conditioning System and would be the primary ASHRAE point of contact and means of dissemination for project results.

Once the meeting schedules for Title 24, part 6, and Title 20 revisions are set, a future plan could be devised for connecting with the individuals and groups that have influence in getting changes made that affect treatment the technologies being studied in this project.

For utility "Savings by Design" programs, connections could be made via e-mail and telephone contact with influential individuals in both these groups. EPRI members and other mutual contacts could be used to facilitate communication with these organizations.

Connections with AHRI, GSA, LEED and Energy Star could be made, along with colleagues at AHRI, and EPA, and contact with EPRI members belonging to other relevant utilities.

Table 57: Codes and Standards Organizations and Issues Relating to Next-Generation RSCS

Organization	Function	Issue	Action Needed	Applicable Documents
California Energy Commission	Responsible for energy efficiency standards including the: California Energy Code: California Code of Regulations, Title 24, Part 6, relating to building energy efficiency	Title 24, part 6 deals with Building and HVAC equipment and operation	Get information to Title 24 decision makers to assure that product enhancements are treated properly.	2016 Building Energy Efficiency Standards for Residential and Nonresidential Buildings, CEC-400-2015-037, June 2015, effective January 1, 2017
	California Code of Regulations, Title 20, Public Utilities and Energy, Chapter 4 Energy Conservation, Article 4, Appliance Efficiency Regulations	Title 20, Chapter 4, Article 4 deals with appliance energy efficiency including unitary heat pumps	Get information to Title 20 decision to assure that product enhancements are treated properly.	2016 Appliance Efficiency Regulations, CEC 140-2012-002, October 2016
Division of State Architect	They check compliance of building plans with all codes and standards.	DSA needs to be made aware of desired changes to State Codes	Get information to DSA decision makers to assure product enhancements are treated properly.	2016 Building Energy Efficiency Standards for Residential and Nonresidential Buildings

Organization	Function	Issue	Action Needed	Applicable Documents
ICC (International Code Council) Consolidation of BOCA (Building Officials and Code Administrators International), ICBO and SBCCI in 2003	The 2016 California Energy Code cited above is available from the ICC at order@iccsafe.org , Headquarters at 500 New Jersey Avenue NW, 6th Floor, Washington, DC 20001	Title 24, part 6 deals with Building and HVAC equipment and operation	Get information to Title 24 decision makers to assure that product enhancements are treated properly. These will then be incorporated by ICC into new standards	2016 California Energy Code, California Code of Regulations, Title 24, Part 6 California Building Standards Commission, effective January 1, 2017
Savings By Design Funded by California utility customers and administered by California's investor owned utilities. PG&E Contact SMUD contact SCE Contact SDG&E Contact	Provides services and incentives to raise energy awareness to architects and building owners. Training sessions are held at utility energy centers. Design assistance is available for design strategies, modeling, and systems integration. Financial incentives are available to owners and designers when a new building exceeds the minimum Savings by Design Thresholds.	Changes adopted for Title 24 would have an impact on Savings By Design. Output of this current CEC Program with regard HVAC and Appliance energy use could have an impact on Savings by Design.	Follow the progress of Savings By Design to assure that their requirements are being addressed. Facilitate incorporation of Program study into their ratings and information materials including their Energy Design Resources materials	Savings By Design Resources Program Handbook Savings by Design Participant Handbook Energy Design Briefs are provided for energy efficient alternatives, including HVAC options, as part of Energy Design Resources on the Savings By Design web site.

Organization	Function	Issue	Action Needed	Applicable Documents
American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) http://www.ashrae.org	Sets guidelines and standards for HVAC equipment. Technical committees (TCs) provide oversight on issues that might impact codes and standards. TCs that might be concerned with Next Gen System are as follows: ▪ TC 1.11 Electric Motors and Motor Control, ▪ TC 3.1, Refrigerants and Secondary Coolants, ▪ TC 5.5 Air-to-Air Energy Recovery, and ▪ TC 8.11, Unitary and Room air conditioners & Heat Pumps ▪ ASHRAE also has standards for residential buildings (90.2) that encourage use of efficient technologies	TC 1.11 members understand and provide information to their peers and 90.2 about effective use of efficient variable speed drives for compressors and fans/blowers	Keep TC 1.11 informed of adjustable speed drive compressor and blower technology	TC1.11 Scope
		TC 3.1 members are concerned with low GWP refrigerants and would welcome information on use of R-32 in heat pumps	Keep TC 3.1 informed about heat pump performance with R-32 as a replacement for R-410A	TC 3.3 Scope
		TC 5.5 Air-to-Air Energy Recovery are interested in heat recovery heat exchangers	Keep TC 5.5 informed of heat recovery ventilator test results	TC 5.5 Scope

Organization	Function	Issue	Action Needed	Applicable Documents
		TC 8.11, Unitary and Room Air Conditioners & Heat Pumps have oversight on factory-engineered integrated heat pump systems such as the Next Gen Residential Space Conditioning System	Keep TC 8.11 informed about all project findings and seek to get on a future TC 8.11 program in 2018	TC 8.11 Scope
		ASHRAE 90.2-2010 provides minimum requirements for the energy-efficient design, construction, and operation and maintenance of low-rise residential buildings	Work with TC and Standards Committee members to assure Program results are incorporated into Std 90.2	ASHRAE Standard 90.2-2007, Energy Standard for Low-Rise Residential Buildings

Organization	Function	Issue	Action Needed	Applicable Documents
AHRI (Air-Conditioning Heating and Refrigeration Institute) ahri@ahrinet.org	Develops test procedures in compliance with ASHRAE and ANSI standards. Publishes product ratings	Assure test procedures for HVAC equipment that incorporate Next Gen System technologies adequately measure and reflect the benefits of this equipment.	Keep AHRI informed regarding project results	All Standards ANSI/AHRI 210/240-2008 with Addenda 1 and 2: Performance Rating of Unitary Air Conditioning & Air-Source Heat Pump Equipment
USDOE (Department of Energy) Appliance and Equipment Efficiency Standards	The U.S. Department of Energy's Appliances and Commercial Equipment Standards Program develops test procedures and minimum efficiency standards for residential appliances including central air conditioners and heat pumps	Current standards are not likely to include provisions for the features of the Next Gen Residential Space Conditioning System studied in this project. Inclusion of study results may cause standards levels to be increased and over all residential space conditioning energy use to be reduced., Central Air Conditioners & Heat Pumps,	Provide information to DOE Appliance Efficiency Standards personnel for inclusion in Central Air Conditioner and Heat Pump Standards	Central Air Conditioner and Heat Pump Standards Central Air Conditioner and Heat Pump Test Procedure

Organization	Function	Issue	Action Needed	Applicable Documents
U.S. General Services Administration (GSA), Office of the Chief Architect	GSA oversees the buildings, services, products, technology, and other essentials to support federal workers in government-owned and leased buildings (including residential buildings). GSA strives to facilitate procurement of state of the art products and services	GSA needs to be made aware of the advantages to the United States government before they will consider codifying the use of the equipment improved in this Program	Keep GSA informed regarding project results. Work with GSA to assure proper application and operation of improved technologies	Facility Standards for the Public Building Service P100, November 2010 Chapter 5 Mechanical Engineering Chapter 6 Electrical Engineering
U.S. EPA (United States Environmental Protection Agency) and USDOE Energy Star	ENERGY STAR is a program of the U.S. EPA and U.S. DOE helping to save money and protect the environment through energy efficient products and practices.	Products studied in this Program need to be covered by Energy Star and other EPA programs	Provide Program information to EPA for inclusion in ongoing and emerging EPA programs	Energy Star Publications, including Residential HVAC Equipment Brochures Newsletters
United States Green Building Council (USGBC)	LEED Green Building Certification System Oversees LEED (Leadership in Energy and Environmental Design) and the Green Building rating system for new construction and major renovation of commercial and residential buildings	The rating system assigns points based on the attributes of the site and building. Certification criteria need to include improved efficiency levels implied by the Next Gen System studied in this Program	White papers should be addressed to LEED staff with the need to include Next Gen Residential System improved efficiency levels in LEED rating system criteria.	Program for new buildings and major renovations

Organization	Function	Issue	Action Needed	Applicable Documents
Uniform Mechanical Code (UMC)	Mechanical code utilized by California and developed by the International Association of Plumbing and Mechanical Officials (IAPMO) to govern the installation, inspection and maintenance of HVAC and refrigeration systems	The IAPMO, through the UMC, also dictates code acceptance for refrigerants in the state.	The UMC must incorporate lower-GWP (and mildly flammable) refrigerants that the California Air Resources Board is looking to promote for HFC phase down and HFC related emissions reductions	2018 Uniform Mechanical Code

Source: EPRI

Technology Transfer Activities and Products

The market connections effort could produce a number of information products designed to influence the market participants identified in the Market Participants section of this chapter, and overcome the market barriers outlined in the Market Barriers section. The dissemination activities described in the Technology Transfer Conduits section should produce desired market effects.

The types of activities and products for the Next-Generation Residential Space Conditioner could include the following:

- Fact sheet /Tech Brief
- Journal Article
- Technical Paper
- Industry Conference
- Presentation at Industry Conference
- Poster (or Presentation) at the EPIC Symposium
- Utility Meetings and Conferences
- EPRI Utility Advisory Council Meetings
- "Word of Mouth" contacts using the materials above

Some details are provided, as follows:

Fact Sheet/Tech Brief

The fact sheet format will likely consist of the following material:

- A Description of the Situation (Describe the problem and the current technology being used.)
- The Technology (Describe the technology, what it looks like, how it works, and how this differs from current practice. Provide a schematic or photo of the technology.)
- Advantages and Opportunities (Clearly outline the advantages of the new or improved technology and situations where it can best be applied.)
- Applications (With examples of effective applications with initial cost, and operating cost)
- Sources of Information (Provide authoritative references, opportunities to obtain additional information and technical assistance in implementing the new or improved technology.)
- Other Issues (present information on ancillary issues, applicable codes and standards, health and human performance improvement; things not covered in the main body of the fact sheet.)

A substantial amount of the Fact Sheet material will be useful for all audiences. Additional material will also be provided with more details addressing areas of concern to a particular market participant.

Journal Articles/Technical Papers

Contacts were explored with appropriate engineering journals and trade publications (such as ASHRAE Journal) to place articles that explain the technology and applications of the Next-Generation Residential Space Conditioning System and have been published.[65] [66] [67] [68]

Presentations

Presentation venues were sought (such as the ASHRAE Winter or Annual Conference) and presentation materials for explaining the benefits of the Next-Generation Residential Space Conditioning System will be prepared for delivery at those venues in order to increase understanding of this equipment by officials with the goal of encouraging them to include these systems in specifications for new residential projects and major modernizations. As described below, several presentations were given to key market participants.

Industry Conferences

The project researchers presented papers at HVAC and energy efficiency industry conferences to audiences consisting of key market participants.

For example, in August 2018, project researchers presented "Evaluation of the Next-Generation Residential Space Conditioning System for California" at the ACEEE Summer Study Program on Energy Efficiency in Buildings in Asilomar, California. The audience consisted principally of energy efficiency advocates, including university, utility, and government researchers. In January 2019, project researchers presented the same paper at the ASHRAE winter meeting in Atlanta, Georgia. There, the audience consisted principally of HVAC industry participants including academics, architects and engineers, manufacturers, dealers/retailers, contractors, utilities, and federal and state government standards practitioners.

EPIC Workshops and Symposia

The plan was to participate in EPIC workshops, symposia and webinars that are designed to disseminate information on EPIC technologies. Project participants planned on engaging in panel discussions/breakout sessions and preparing posters and related materials for poster sessions at the symposia.

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[65] Beaini et al., "Next Generation Residential Space Conditioning System for California," (AT-2019-C035). ASHRAE 2019 Winter Conference Proceedings. https://engagefully.org/Sessions/Details/520360

[66] Beaini et al., "Evaluation of the Next Generation Residential Space Conditioning System for California," 2018 ACEEE Summer Study on Energy Efficiency in Buildings. https://wcec.ucdavis.edu/wp-content/uploads/2018-ACEEE-Res-Heat-Pumps.pdf

[67] Krishnamoorthy,S., Modera,M. and Harrington,C. (2017) "Efficiency Optimization of a Variable-Capacity/Variable-Blower-Speed Residential Heat-Pump System with Ductwork."; Energy and Buildings – 150: p.294-306.

[68] Krishnamoorthy, S., Modera, M.P. & Harrington,C. "Improving system efficiency for a variable-capacity/variable-blower-speed residential heat-pump system with multizone ductwork, Science and Technology for the Built Environment, DOI: 10.1080/23744731.2018.1499384

An EPIC symposium was held on February 7, 2018, in the Sacramento Convention Center. These symposia are held annually in California. The Next-Gen RSCS project results were showcased at the symposium, as a poster session. Previously, a presentation was given at the CEC EPIC Symposium in Folsom in February 2016. Under the Phase 3 project deliverables, a webinar was held on March 27, 2019 to share project results with stakeholders and manufacturers. The objective of this webinar was to share the results of the project's laboratory and field evaluation results and technology recommendations to California stakeholders and manufacturers, for the California Next-Generation Residential Space Conditioning Systems. Additionally, the webinar provided technology transfer activities, lessons learned and next steps. More than 100 stakeholders attended the webinar and actively participated in the interactive discussions at the end of the presentations. Details of the webinar are presented in Chapter 4 of this report.

Codes and Standards Activities

The plan was to showcase the features of the Next Gen Residential Space Conditioning System (Next-Gen RSCS) for California to assure that they are appropriately accounted for in industry standards by participating in corresponding Standards setting activities including the newly approved and published AHRI Standard 1380, "Demand Response through HVAC Equipment in Residential and Small Commercial Applications."[69] EPRI staff attended meetings regarding AHRI Standard 1380 to stay informed about the AHRI 1380 standard development for ADR and to provide feedback based on the results from this CEC-14-021 project as well as EPRI's Next-Generation Heat Pump initiative to inform the Standards committee of project findings.

The plan was also to follow ASHRAE technical committees that have oversight on the operation and features of the Next-Gen RSCS and major standards activity associated with these committees, such as participating in TC 8.11 (Unitary and Room Air Conditioners and Heat Pumps Technical Committee).

Also, the opportunity was taken to connect the outcomes of Next Gen Residential Space Conditioning System project to California Title 24 Upcoming Standards Updates principally by relating pertinent project results to CEC Buildings and Codes Standards personnel during and after this project's duration. California Energy Commission staff responsible for Title 24 updates participated in critical project review meetings that were held on August 31, 2016 and October 16, 2017 and site visit on January 8, 2019. An important set of interactions that occurred at these meetings was briefing of CEC Title 24 Building Codes and Standards personnel with the goal of incorporating project results in the next round of Title 24 updates.

EPRI Meetings and Activities

EPRI has regular meetings with key market participants that will be used to transfer the information developed in this project to these key participants.

Meetings include EPRI advisory meetings with influential utility members that are used to shape EPRI research, develop demonstration and marketing opportunities for technologies and provide a conduit for the advisors to impart information to colleagues at their "home" utilities.

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[69] 2019 Standard for Demand Response through Variable Capacity HVAC Systems in Residential and Small Commercial Applications, AHRI, 2311 Wilson Boulevard, Suite 400, Arlington, VA22201

Advisory meetings are held twice a year (spring and fall), usually in February and September. The Next-Gen RSCS project for the CEC and the EPRI Next -Generation Heat Pump Deployment Initiative is working together to improve the efficiency and demand-response capability of heat pump technology. Progress on the Initiative and the Next-Gen RSCS project has been regularly presented at EPRI advisory committee meetings held twice a year in the spring and fall, usually in February and September, as well as specific EPRI Next-Generation Heat Pump workshops (March 2017 in Dallas, TX). The audience at the EPRI advisory meetings consisted of EPRI and influential utility personnel. Utilities are provided with information that they can use to formulate their own local programs to encourage acceptance of the technology and to pass on to their local HVAC industry market participants.

The EPRI Energy Efficiency and Demand Response Symposium is held annually as a forum for utility company members, manufacturers, researchers (EPRI and other), industry stakeholders, and government agencies to discuss changes in the industry and actions that need to be taken. Key findings of the Next-Gen RSCS project have been presented at this symposium. The 2016 meeting was held in Long Beach, California, while the 2017 meeting was held in October in Orlando.

EPRI Electrification 2018 conference explored the critical issues, benefits, and opportunities of electrification. Session tracks included Residential and Commercial Electric Technologies and afforded an excellent opportunity to transfer Next-Gen RSCS information to the target audience. Conference attendees typically include utilities, industry, government, and academic leaders. The Next-Gen RSCS was featured in a booth by the technology provider Daikin/Goodman at this conference.

Word-of-Mouth Contacts

The team created a word-of-mouth movement by involving opinion leaders in the process of convincing them of the benefits of the Next-Generation Residential Space Conditioning System. Program technical staff worked closely with influential market participants to assure that the project was shaped to meet their needs and that the results are accepted by their peers. Program staff plan to work with manufacturers to promote awareness of the market need for developing a new line of products, or adapting existing products, to provide the benefits of the Next-Generation Residential Space Conditioning System

Program staff attended key meetings to “get the word out” to professionals and other influential market participants. Program staff will interact with federal and state government personnel, professionals at technical meetings, utilities, customers and users at utility energy centers across the state, and a range of experts and market participants.

CHAPTER 7: Production Readiness Plan (Task 7)

System Features

Features included in the future space conditioning system are as follows:

- Variable capacity compressors, indoor blowers and fans
- Alternative refrigerants
- Dual fuel –electric and gas
- Fault detection and diagnostics (FDD)
- Reduced duct losses
- Integrated ventilation control
- Zonal control
- Automated demand response

Improvements resulting from widespread deployment of this system are expected to be reduced energy use and peak demand, greater comfort, improved noise control, lower energy and demand, better controllability, smoother operation, reduced global warming potential (GWP), improved maintenance and reliability

Preliminary Screening of the Features

Based on the laboratory and field tests, the following features are believed to be the most attractive options for commercialization:

- Variable capacity compressors, indoor blowers and fans
- Alternative refrigerants
- Dual fuel – electric and gas
- Fault detection and diagnostics (FDD)
- Reduced duct losses
- Automated demand response (ADR)

A large improvement in efficiency was not seen due to integrated ventilation control and zonal control; however, they were included in the Production Readiness Plan. As technology develops, and costs of these components come down, they could very well be integrated in future systems.

It is to be noted that this project did not consider one of the features that customers will be looking for in the future – indoor air quality (IAQ). As buildings are constructed with better insulation, resulting in minimal outside air intrusion, IAQ will become an important feature.

Production Requirements

Variable Capacity Compressors, Blowers, and Fans

Variability in capacity is obtained by using variable speed motors that incorporate inverter drives. The main technologies of these variable capacity motors include are low-cost scroll compressors and the appropriate inverter drives. Currently, most of inverter drives are imported from Europe or Asia. Inverter board components are limited by an allocation perspective and made mostly overseas. Compressors are also procured from external sources. In-house manufacturing of low-cost variable capacity scroll compressors will be required for the product to be economical.

For manufacturing localization, in-house manufacturing of inverter boards and compressors is particularly challenging. To overcome that challenge, further emphasis needs to be focused on supply chain management; only then local manufacturing will become economical. The biggest effort is localizing these processes.

Alternative Refrigerants

Next generation residential space conditioning will likely incorporate next generation refrigerants. On a global basis, space conditioning will move toward the phase down of higher global warming pollutant (GWP) refrigerants under the United Nations Kigali Amendment to the Montreal Protocol. More specific to California, however, there will be new bans as part of the pending California Air Resources Board (ARB) regulations on refrigerants. These regulations are scheduled for adoption at the end of 2019.

As part of the September 2018 "Global Climate Action Summit," co-hosted by Governor Jerry Brown, a number of HVAC and chemical companies (Carrier, Daikin, Lennox, Trane, Chemours, Honeywell) and the Natural Resources Defense Council (NRDC) committed to voluntarily stop sales of all stationary air conditioning products (excluding chillers) containing refrigerants greater than 750 GWP by January 1, 2023. This commitment was developed in concert with the Governor's Office and ARB, and is considered the basic framework for the 2023 phase-down to come in the ARB regulations.

Although many manufacturers have agreed to these limits in principle, a number of commercial and other barriers to adoption remain. Meeting this 2023 ban will require considerable engineering and other resources, as well as production changes, testing, training, and a host of additional new costs. Most notably, building and fire safety code requirements must also change in time for the commercial adoption and sale of product containing this equipment. Many lower-GWP refrigerants also have flammability characteristics and will need adoption by ASHRAE, the Uniform Mechanical Code, and other standard bodies before commercial adoption in California can occur.

Some of these refrigerants also bring performance and other environmental co-benefits. For example, according to Daikin, HFC-32 (or "R-32"), with a GWP of 675, has these advantages based on research on split systems:

> HFC-32 is currently considered to be the most promising substance for use as a next-generation refrigerant in both residential and commercial air conditioners. The potential refrigerating effect of HFC-32 is 1.5 times that of HCFC-22 or R-

410A. More specifically, pressure losses are lower with HFC-32 than HCFC-22 or R410-A for the same capacity and the liquid density of HFC-32 is also 10% lower. Thus the piping diameter can be smaller. As a result, the charging volume can be 30% less than with HCFC-22 or R410-A. The cooling seasonal performance factor (CSPF) of HFC-32 is higher than conventional refrigerants. Its peak power consumption is also lower, helping to alleviate power shortages in large cities during periods of high demand.[70]

Manufacturers will need to balance a number of factors when selecting these alternative refrigerants. Safety, through the lifecycle of the equipment, including transport, storage, installation, use, servicing, recovery, and recycling, will remain a critical consideration. Similarly, environmental effects between refrigerant alternatives across the lifecycle of production must be assessed (in addition to the base GWP of each refrigerant). Single molecule refrigerants, for example, may be easier to reclaim and recycle than blends. Energy efficiency performance is also critical, across both heating and cooling functions, and in more extreme conditions. Finally, cost-effectiveness is a key concern: the relative cost of particular alternatives and their production and raw materials costs, ease of installation, and reclamation.

For future manufacturing, selection of the right refrigerant should take into account non-GWP characteristics such as the effect on equipment performance and required charge amount. Also, there is a need to take special care and adopt processes when using mildly flammable refrigerants, such as R-32. This is true at the manufacturing facilities and also in the field by contractors.

This project tested R-32 refrigerant in the lab and found it to be very effective. However, it could not be used in the field trials, since U.S. EPA’s approval is required, prior to its use. As to the R-32 manufacturing facility, there need to be investments in the laboratory to be able to use R-32. Specific laboratory training on the use of such refrigerants is also required.

Dual Fuel (Electric and Gas)

For heating purposes, the system needs to be designed to use an electric heat pump and then shift to gas heating, as economics warrant. The transition point from heat pump to gas furnace operation is set using the breakeven temperature in the system controls, which is used as the compressor lockout temperature.

In terms of new development, manufacturers should consider developing a controller that not only optimizes for lowest cost, but also for other aspects such as highest efficiency and lowest carbon footprint. Such controllers are already being developed in Europe; technology transfer to the United States is simple, as long as incentives are put in place to do so.

Fault Detection and Diagnostics

As technology progresses and the cost of Internet of Things becomes lower, most manufacturers are installing sensors and controllers in space conditioning equipment. Manufacturers are expected to install onboard sensors for checking the sufficiency of refrigerant charge, for providing automatic feedback when filters are clogged, and performing

[70] “Next-Generation Refrigerant HFC-32 for Stationary Air Conditioners and Heat Pumps,” Daikin Asia, 2016

other tasks. It is expected that the manufacturers will provide cloud-based real-time feedback to customers on the operation of their equipment. Such capability will enable manufacturers to monitor and ensure their equipment continues to perform as it should after installation. This is mostly existing technology, and it is not a technically challenging area for production readiness.

FDD is also becoming capable of predicting future malfunctions of equipment.

Automated Demand Response

ADR technology is developing. Currently AHRI is developing specifications for the inclusion of demand response (DR) technology, under AHRI Standard 1380. The purpose of this standard is to establish requirements for DR strategies in variable capacity HVAC systems that are less than 65,000 Btu/hr to benefit the electric grid in a predictable manner. The standard facilitates end users' participation in price response, grid response, or similar incentive DR programs offered by electric utilities or other entities such as aggregators.

The standard is intended for the guidance of HVAC systems and electric utility industries, including manufacturers, designers, installers, contractors, users, and demand side management program managers. By providing standardized requirements for DR-ready HVAC systems, DR program managers can be assured the equipment is able to communicate in standardized messages on OpenADR 2.0 standardized DR communication protocols.

Once this standard has been approved by the AHRI Standards Development Organization, it is expected that many manufacturers will incorporate DR controllers in their overall control scheme. The results of this project are expected to positively affect the development and application of DR controllers in such HVAC systems.

Ventilation (ERV/HRV)

As more homes become "zero net energy (ZNE)", leaks and air losses will be reduced dramatically. As leaks and air losses are reduced in ZNE homes, indoor air quality (IAQ), such as carbon monoxide, carbon dioxide, volatile organic compounds, and pollen, will need to be mechanically maintained. Ventilation devices, such as energy recovery ventilators (ERV) and heat recovery ventilator (HRV), are equipped with air filters and will play a crucial role to enable that. The ventilation devices can be integrated with IAQ sensors and control devices, such as smart thermostats, so that IAQ can be optimized and visualized for homeowners.

In addition, ERV and HRV can reduce cooling and heating loads of the main HVAC systems by recovering heat before the air is exhausted. The difference between the two devices is that ERV can recover both sensible and latent heat, whereas HRV can recover only sensible heat. Therefore, the recovery efficiency of ERV is normally higher than that of HRV. However, in a hot and dry climate such as many of the inland climate zones in California, the difference becomes insignificant as there is little latent heat existing in the air.

Testing has shown that the future will provide more sophisticated ventilation schemes. The challenges are believed to be not so much on the manufacturing side as on how to integrate ventilation devices with the rest of the ecosystem.

Duct Zone Control

Zoning can reduce energy consumption and peak power by cooling and heating only the zones that are occupied. HVAC systems with zoning cool and heat the occupants, not the buildings. When the systems are not mini/multi-split ductless systems, duct zone control devices are needed to provide zoning to the occupants. Duct zone control devices use dampers to manage the air flow and volume to each zone. However, only 7 percent to 10 percent of contractors currently install duct zoning. Additional contractor training is required to boost the percentage.

Estimated Cost of Production

Production cost information is difficult to obtain from manufacturers, but for its part, Daikin has noted that for a multi-faceted system like that contemplated by this plan for next generation equipment, it may still be the case that more than half of the costs could be a function of inverter technology (variable capacity compressor and variable speed blower features), when compared against the balance of other features discussed here (such as FDD, integrated ventilation, and demand response control).

Expected Investment to Launch Commercial Product

Investment information from manufacturers has also been difficult to obtain given its proprietary nature, but it appears that manufacturers are spending considerable research and development dollars to develop variable capacity heat pump products and systems. In Daikin's case, for example, roughly half of research and development spending for the Daikin-branded residential products is related to variable capacity systems, and the majority of this total is dedicated to heat pump inverters. Daikin has also made appreciable investments in new inverter and smart thermostat product development and commercialization. The "Daikin FiT" unitary inverter and "Daikin One+" EcoSystem, that includes smart thermostats, ducted zoning controls, and HRV/ERVs, are new products that have been launched into the market following this research.

Intellectual Property Information

No intellectual property has been developed under this contract. Manufacturers are expected to maintain their trade secrets for optimized manufacturing and reduction of costs.

CHAPTER 8: Conclusions, Summary and Recommended Next Steps

Summary Results from Phases 1, 2, and 3 Testing and Analysis

The following paragraphs provide a synopsis of the laboratory testing performed in phases 1 and 2 and the field testing performed in Phase 3. For more details on the testing and results please see Chapter 2 for Phase 1, Chapter 3 for Phase 2, and Chapter 4 for Phase 3.

Phase 1 Laboratory Results

Phase 1 laboratory evaluation involved the assessment of the operation and performance of six technology features through a variety of steady-state and dynamic mode tests at three independent laboratories. The key results are summarized as follows:

Variable Capacity

Variable capacity heat pumps (VCHP) with a variable capacity compressor and variable speed blower provided 22 percent to 32 percent cooling energy savings across California climate zones compared to a baseline 14 SEER single-speed system.

The Next-Gen RSCS can provide more than 90 percent of the annual heating load without requiring backup heating for most of the of the 16 California climate zones modeled.

Cooling and heating part-load efficiencies are better than full-load efficiencies at mild temperatures (between 35°F [2^{o}C] and 90°F [32^{o}C]). Higher part-load efficiency corresponds to higher SEER/HSPF values.

The VCHP is able to modulate and match well with the imposed dynamic load.

Integrated Ventilation Control

The use of a heat recovery ventilator with the VCHP provides an additional 1 percent to 4 percent cooling savings compared to a baseline SEER 14 fixed-speed HVAC system for California Climate Zone 10 (cooling design condition of 101°F (38^{o}C) and a heating design condition of 35°F (2^{o}C)).

Modeling results for the heating season showed that the capacity of the Next-Gen RSCS system (without backup) could be increased by around 1 percent in cooler California climate zones (1 and 2 and 11 through 16) when using an HRV.

Automated Demand Response

The testing demonstrated VCHP’s capability to act as a flexible demand response resource.

During automated demand response (ADR) events, VCHP’s capacity reduced non-linearly with reduced load, providing an opportunity to save energy and demand with less discomfort to the occupants. As shown in Table 19, at a 50 percent speed setting, demand is reduced by 65 percent while capacity is reduced by only 36 percent. At a 30 percent speed setting, demand is reduced by 75 percent while capacity is reduced by only 49 percent.

Dual Fuel (Intelligent Heating)

The economics of the dual fuel heat pump depend on the outdoor temperature and the prevailing gas and electricity rates. Substantial heating season cost savings are possible using a dual fuel heat pump compared to a high efficiency gas furnace with favorable electricity rates. Calculations were performed to assess the economics of dual fuel heat pumps in selected locations in California. California energy prices tend to be high relative to national averages, and natural gas prices have not kept pace with electricity prices. Despite these factors there are still situations where the Next Gen dual fuel heat pump (DFHP) can provide attractive savings.

The fact that operation of the DFHP can be adjusted as utility prices vary permits the homeowner the option to benefit from future changes in utility prices that might reduce the ratio of electricity to gas prices in the future. The assurance that the homeowner will be able to experience the lowest future heating costs possible, is an important attribute of dual fuel heat pump capability and increases the value of this feature to potential purchasers of the Next Gen Residential Space Conditioning System.

Laboratory testing confirmed the functionality of the dual fuel heat pump concept in all possible modes of operation.

Duct-Loss Assessment for Single-Zone Configuration

A VCHP connected to a duct system shows great part-load energy-saving potential. For example, when the compressor and fan speeds are synced with each other, the variable-capacity/variable-speed blower system performs at maximum system efficiency at a speed setting of 60 percent of the rated speed when the outdoor dry-bulb temperature is 85°F (29°C). This speed setting would have to increase to 80 percent as the outdoor temperature gets warmer than 85°F (29°C) and further increase to 100 percent speed setting at outdoor temperatures of 115°F (46°C) or higher.

Ducting affects system efficiency at hotter temperatures. For the same outdoor temperature, the compressor and fan can be operated at lower speed settings when the indoor wet-bulb temperature is greater because the equipment is doing more dehumidification. Since the duct losses occur through sensible heat gains, a lower percentage of total cooling produced is lost through the ducts. The result also implies that duct losses play a larger role in the hot and dry California climates compared to hot and humid climates. Adding more insulation to ducts, or keeping at least a portion of them in conditioned spaces will render the variable-capacity/variable-speed cooling technologies more beneficial.

A mathematical model of the system with ducts agreed within 5 percent accuracy with experimental data.

Phase 2 Laboratory Results

Phase 2 laboratory evaluations involved the assessment of the operation and performance of four technology features through a variety of steady-state and dynamic mode tests at three independent laboratories. The key results are summarized as follows:

Alternative Refrigerant Testing

The performance of the variable capacity heat pump (VCHP) was evaluated using R-32 as a refrigerant drop in to a heat pump designed to operate with R-410A. Laboratory findings are provided for R-32 as a refrigerant for this residential HVAC system that can be added to industry literature for use pending legislative approval of this refrigerant. Three key points should be considered while reviewing both labs' test results.

R-32 (GWP 675) demonstrated an ability to be an effective, low GWP replacement for R-410A (GWP 2100) as a drop-in refrigerant in the variable capacity heat pump from an equipment performance and functionality perspective. The use of R-32 in HVAC equipment offers a potential mechanism for peak power reduction in the warmest California climates, while reducing refrigerant charge. R-32 reduced system charge by 29 percent compared to R-410A.

In the cooling mode, trends observed of the R-32 variable capacity heat pump are comparable to the trends observed for R-410A as the refrigerant. At 95°F (35°C) outdoor temperature, where nominal capacity is determined, the minimum output of the R-32 system was 29 percent of the maximum capacity. In R-410A testing of the variable capacity system, the minimum capacity was 30 percent of the maximum capacity at 95°F (35°C). The R-32 variable capacity system demonstrated increased efficiency at part-load operation, and the relative increase in efficiency from maximum to part-load operation increased with decreasing outdoor temperature.

The retrofit of R-410A to R-32 resulted in cooling efficiency increases of 6 percent to 9 percent, 1 percent to 3 percent, and 2 percent to 3 percent for maximum, intermediate, and minimum operation, respectively.

With the implementation of R-32 in the variable capacity heat pump, the peak cooling performance improved by 6.7 percent to 8.2 percent. For residential equipment ranging from 2 to 4 tons, the R-410A variable capacity heat pump provides a potential peak reduction of 80W – 200W over a baseline 14 SEER system. Implementation of R-32 in the variable capacity heat pump provides an additional potential peak reduction of 125W–475W depending upon size of the equipment.

In the heating mode, the COP with R-32 ranged from approximately 2.4 to 4.1 for outdoor temperatures between 15°F and 65°F (–9°C and 18°C). The maximum heating capacity curves of R-32 and R-410A variable capacity heat pump compared the same with two improvements of R-32 by 10 percent and 5 percent at 62°F and 25°F (17°C and 4°C), respectively.

Integration of Zonal Control and Variable Capacity Space Conditioning

Zonal control and variable capacity offer a potentially effective integration of two technologies for improved efficiency. Understanding the functionality and utility of zonal control with a variable capacity heat pump system can provide targeted energy savings. The efficiency impact of zoning is largely dependent upon on the temperature offset for unoccupied zones and the subsequent load reduction on the HVAC system. Laboratory testing demonstrated altered variable capacity performance and functionality with the implementation of zoning. Field evaluations should further assess the performance and functionality of a zoned, variable capacity system.

Variable Capacity Space Conditioning Connected to a Ductwork System in Multi-Zone Configuration

The multi-zone operation of a variable-capacity/variable-fan residential air-conditioner using ductwork routed through an attic has been studied experimentally and analytically at the UC Davis Western Cooling Efficiency Center laboratory. The objective was to determine the optimum zoning controls and the optimum operating speeds for both the compressor and indoor fan at the optimum zoning percentages for achieving maximum system efficiency in hot and dry California climates. The data collected describes the performance characteristics of the system operating when a) varying compressor speed and indoor fan speed together and b) varying indoor fan speed while holding compressor speed fixed. The results highlight the potential for implementing zonal control in variable-capacity/variable-speed cooling systems using R-410A refrigerant to reduce residential energy use in California, and corroborates the proposal to combine these technologies to create an integrated efficiency solution for maximum energy efficiency. The major results of the laboratory can be summarized as follows:

- Reducing the number of active zones improves the delivery effectiveness of the duct system (reduces duct losses) and increases fan power.
- In general, the optimal number of zones for maximizing system COP increases as the capacity/airflow percentage is increased.
- It was shown in Phase 1 testing that during very hot outdoor conditions the highest system efficiency occurred at maximum operating speed. Phase 2 testing showed that reducing the number of active zones while operating the equipment at low speed produced similar system COPs as the maximum achieved in Phase 1 tests. This result is especially relevant when considering how to control variable capacity systems during a demand response event.

Examined in more detail, the multi-zone testing results demonstrated that:

- Delivery effectiveness of the duct system has an inverse relationship with the number of zones employed for any given capacity/airflow percentage and duct-zone temperature. That is to say, the delivery effectiveness is highest for single-zone operation and progressively decreases as the number of zones increases.
- System COP values when operating the equipment under zoned conditions are generally higher than the values obtained under non-zoning operation for the same capacity/airflow percentage and duct-zone temperature. This behavior is due to the improved delivery effectiveness of the duct system, which increases at higher duct velocities. There is a tradeoff with higher fan power for zoned operation, which creates an optimal zoning that does not necessarily coincide with the zoning that achieves the highest delivery effectiveness. In general, the optimal number of zones for maximizing system COP increases as the capacity/airflow percentage is increased.
- Zoning is more effective at higher duct-zone temperatures. This is because the percentage increase in delivery effectiveness is higher due to zoning when the duct-zone temperatures are hotter, whereas the additional blower power consumption due to zoning is independent of temperature.

- For very hot duct-zone temperature, the heat pump equipment using R-410A is capable of operating at lower capacities/air flow rates using a zoning mechanism that yields a system COP value comparable to the maximum system COP when operating without zoning. Recalling that lowering the capacity/air-flow rates hurt the system COP for hotter duct-zone conditions when operating without zoning, this result implies that in very hot climates, zoning can be employed in variable-capacity equipment to respond to demand-response events from the utility without compromising on efficiency.
- When zoning is employed, operating with a higher fraction of indoor air flow than capacity increases system efficiency only at low capacities. When the capacity is 60 percent or more, increasing the blower fan speed relative to the compressor speed reduces the system COP when zoned, due to the higher fan power consumption.
- More efficient control strategies are needed to optimize the performance of heat pumps connected to ductwork located in an attic for hot and dry California climate zones. The system balance is affected by the duct-zone temperatures, which invites the need for revising ducting standards.
- A VCHP connected to a multi-zone configuration with ductwork has higher system COP under zoned conditions, compared to non-zoning for the same capacity/airflow percentage and duct-zone temperature.

Fault Detection and Diagnostics

Both the heat pump and the furnace have an extended list of faults that are detectable and that can aid in the repair and maintenance of the system. The testing of the fault detection and diagnostic (FDD) capabilities was limited in scope to primarily those faults that were thought to be the most likely to occur during normal usage. Thirteen of the 51 listed fault codes for the heat pump were triggered, as well as 5 of the 25 codes listed for the furnace. The result of the evaluation was that the FDD system was very good at correctly identifying the cause of a fault condition when something had gone very wrong, but it was not as good at alerting the user that something was going wrong and should be attended to for optimal performance and preventive maintenance. This capability is thought to be present with the existing components and may just require a more sophisticated software upgrade. This change should retain some conservatism to the extent that the system will not trigger too many alerts so that the end user stops paying attention and does not take action.

Fault detection and diagnostic (FDD) systems can be used to improve HVAC unit performance by alerting users and contractors when degradation or a malfunction is taking place. This permits timely correction of an incipient fault that could otherwise result in a system failure or rapid response to a failure that has already occurred.

Phase 3 Field Evaluation Results

The objective of the field evaluation was to assess the functionality of five Next-Gen RSCS features in residential homes and evaluate the performance with respect to the customer experience.

Variable Capacity Compressor and Blower

The cooling and heating seasonal analysis of the field data for VCHP both show energy consumption within 20 percent of the lab data model. Some discrepancies could be caused by the difference between the indoor chamber setpoint during lab testing, and the thermostat setpoints at the host sites.

The cooling performance of the units show a range of EER between 10 and 25, which agrees with the range of EER shown in the lab testing in Phase 1.

Dual Fuel (Intelligent Heating)

Once the breakeven temperature was set for each of the three host sites, the Next-Gen RSCS performed as expected in heat pump mode and gas furnace mode throughout the duration of the heating season. The Next-Gen RSCS dual fuel capability provides an important choice for the residential customer to adjust the unit settings based on economic factors (utility rates), efficiency factors (heat pump versus gas furnace), or environmental factors (reduction of carbon footprint).

Having a controller that automatically receives pricing information and sets the breakeven temperature accordingly would minimize the need for customer interaction. Future work is needed to develop a versatile, intelligent heating controller that can receive a signal based on utility prices and customer choice/preference (price, efficiency, fuel mix distribution), which would optimize the Next-Gen RSCS's performance in the heating mode.

Auto Demand Response

The unit capacity reduction is less than the unit power reduction, thus the customer comfort is not compromised while energy and demand savings can be realized during a utility-led demand response event.

Field automated demand response testing was conducted in the heating mode and will be conducted in cooling mode during the upcoming cooling season. Updated results will be provided in the Technical Update to follow in late 2019.

Zonal Control and Duct Delivery Effectiveness

The field evaluations all included zoning equipment based on the findings from the lab test results that showed significant efficiency gains when zoning variable-capacity equipment. There were some challenges implementing the zoning that led to too much airflow through smaller zones in the home during some periods of operation. For example, during defrost cycles, the indoor fan ramped up to high speed without consideration of the number of open zones. The project team was able to adjust the zoning controls to minimize the issues around over-flow in small zones, but there should be better coordination between the zone controller and the heat pump that allows the zone controller to adjust dampers based on feedback from the heat pump about the mode of operation or fan speed.

This project demonstrates a clear need for zoning when installing variable capacity equipment with ductwork located in unconditioned space. A minimum of two zones is recommended; however, there are challenges with adding additional zones. Two zone systems with proper controls would achieve much of the benefit of the optimized approach demonstrated in Phase 2.

Customer Feedback

The homeowners provided the following summary feedback on their experience with the Next-Gen RSCS. Homeowners appreciate:

- The quieter operation of the Next-Gen RSCS compared to their previous single speed AC unit.
- How quickly it cools or heats the space.
- The app-based controller to turn thermostats on and set the temperature of individual zones.
- The ability to control temperatures in individual spaces independent of other spaces.
- The theoretical convenience of zonal control; however, feedback indicated that zonal control added complexity to the VCHP system use for more than two zones. Airflow was too forceful in certain zones, thus making it noisy in certain rooms.

Recommendations for Future Research

The project results prompted recommendations for eight future endeavors that would further develop and extend the benefits of Next-Generation Residential Space Conditioning systems.

Configure the Next-Gen RSCS into Multiple Models

The objective of this project was to evaluate the Next-Gen RSCS, which has multiple energy efficient features integrated into a single system. From the project findings, a key recommendation to the manufacturers would be to configure different models of the Next-Gen RSCS, as illustrated in Table 58, grouping certain features for selected configurations, demographics, or climates. The premium model could include all seven energy efficiency features of the Next-Gen RSCS. The base model could be a variable capacity heat pump with demand response, fault detection and diagnostics and zonal control – all of which are crucial features to benefit from the efficient operation of variable capacity heat pump. The intermediate model could include intelligent heating as an additional feature to the base model.

Table 58: Recommendation for Manufacturers' Configuration of Next-Gen RSCS into Base, Intermediate, and Premium Models

Next-Gen RSCS Energy Efficiency Technology Feature	Efficiency/Cost Savings Potential compared to SEER 14 single speed without that feature	Base Model	Intermediate Model	Premium Model
Variable-Capacity Compressor & Blower	22% to 32% in this study	✓	✓	✓
Fault Detection & Diagnostics	Inform maintenance alerts to avoid inefficient operations. High energy reductions possible	✓	✓	✓
Demand Response		✓	✓	✓
Zonal Control	Up to 50% savings at 40% system capacity	✓	✓	✓
Dual Fuel (intelligent heating)	Up to 22% for cases run		✓	✓
Integrated Ventilation	1-4% seasonal cooling energy savings			✓
Alternative Refrigerants	1.2% to 3% for cooling			✓

Source: EPRI

Examine Cost Effectiveness of Each Feature in California

While the current project conducted the testing and evaluation of each of the features to quantify the potential for energy savings and performance benefits, a future project could develop a model the evaluates the cost effectiveness of each feature. The model would have to evaluate energy and demand cost savings for each feature for all California climate zones for representative housing, demographics, and occupancy situations. Thus, the model would:

- Determine the incremental equipment cost for each feature and potential combination of features.
- Assess cost effectiveness of each feature or combination of features for all California climates and selected demographics and residential construction/configuration options.
- Select base, intermediate, and premium models based on this assessment, matching Next-Gen RSCS features to climates, demographics and configurations.

Refining Zonal Control with Variable Capacity Heat Pump

From the field evaluation experience and system operation, it was apparent that adding more zones introduces further complexities to VCHP operation. Thus, adding more than two zones has to be more effectively planned and set up in the design stage of the duct work, so the zones are appropriately sized based on the minimum airflow of the VCHP that can condition a specific volume.

Codes and Standards (Title 24 and ASHRAE)

The zonal control and duct delivery effectiveness evaluation brought to light the importance of examining ways to limit the heat transfer to ducts in unconditioned spaces such as attics (for example, adding insulation, zonal control, and duct sizing). Codes and standards are needed to establish different combinations for efficient operation of ducted split system variable capacity heat pump if the ducts are in unconditioned spaces.

Develop Intelligent Heating Controller

While demand response-enabled devices can receive a signal to reduce the system peak demand in cooling mode, development of a versatile heating controller for a dual fuel heat pump that can receive a signal based on utility prices and customer choice/preference (price, efficiency, fuel mix distribution) would optimize the Next-Gen RSCS's performance in the heating mode. The controller would provide the customer to select their preferred heating mode of operation: efficient; economical; environmental; entertainment. The mode selection then drives the system controls for heating mode for an electric VCHP with back-up heat, whether electric or gas. For economic mode, the controller would receive the utility electric and gas prices to determine the breakeven temperature for economical operation. For the environmental mode, the controller would receive the fuel mix of the energy sources to determine the carbon footprint for heating demand. The entertainment mode would enable the customer to choose the operation mode to provide the most thermally comfortable conditions.

Dual Fuel Heat Pump Furnace Efficiency

Examine the influence of furnace efficiency on system cost effectiveness. Is a high efficiency furnace warranted for California climates (with limited heating hours and where some of those hours will served by the electric heat pump)?

Fault Detection and Diagnostics

There is a need to refine the sensitivity of the system controls to detect small changes in system performance. By anticipating maintenance, the system degradation can be halted as early as possible. This would require thorough testing in the laboratory by simulating incrementally small changes in selected parameters to examine the sensitivity of the controls and identify gradual degradation in performance.

Alternative Refrigerants

Further research is needed to address the technology and regulatory needs for the use of R-32 (GWP 675) (or equivalent) as a drop-in refrigerant for R-410a (GWP 2100) systems.

GLOSSARY AND LIST OF ACRONYMS

Term	Definition
ADR	automatic demand response
AHRI	air-conditioning, heating & refrigeration institute
ANSI	American National Standards Institute
ASHRAE	American Society of Heating, Refrigeration and Air-conditioning Engineers
ATS	PG&E Applied Technology Services
Btu, BTU	British thermal unit – a unit of energy required to raise 1 pound of water by 1°F
CEC	California Energy Commission
CFM	cubic feet per minute – a unit of air flow
COP	coefficient of performance – a unit of efficiency (dimensionless)
DB	dry bulb temperature (as in Tdb)
DOE	United States Department of Energy
DP	dew point temperature (as in Tdp)
DR	demand response
Drop-in refrigerant	a refrigerant that is replacing another refrigerant without any further modifications to the existing equipment. Ideal drop-in refrigerants will have similar properties and performance to the refrigerant it is replacing
DX	direct expansion, as a descriptor for vapor compression air conditioning
EA	exhaust air from condenser (EAT = exhaust air temperature)
EER	energy efficiency ratio – a unit of efficiency in Btu/h/W
EPIC	Electric Program Investment Charge
EPRI	Electric Power Research Institute
ERV	energy recovery ventilator
EXV	electronic expansion valve
HR or W	humidity ratio or absolute humidity (mass fraction of water vapor to dry air)
HRV	heat recovery ventilator
HSPF	heating season performance factor – weighted average heating EER for a heating season: total heating provided divided by electric energy consumed, including backup resistance heat when the load exceeds heat pump capacity. Units are Btu/Wh.

Term	Definition
HVAC	heating, ventilation and air conditioning
IOU	investor-owned utility
IW	inches of water column – a unit of pressure
OA	outside air (OAT = outside air temperature)
Latent	capacity metric that is due to a change in water vapor content
OEM	original Equipment Manufacturer
PG&E	Pacific Gas and Electric Company
PSIA	pounds per square inch, absolute – a unit of pressure
Q	symbol representing cooling or heating capacity (in tons or Btu/hr)
RA	return air from space (RAT = return air temperature)
RH	relative humidity
RTD	resistance temperature detector or resistance thermometer
SA	supply air to space (SAT = supply air temperature)
SCF	standard cubic foot. For gas, this uses a density referenced to 60°F and 14.73 PSIA.
SEER	seasonal energy efficiency ratio – weighted average cooling EER for a cooling season: total cooling provided divided by electric energy consumed. Units are Btu/Wh.
Sensible	capacity metric that is only a function of temperature difference
SHR	sensible heat ratio – sensible capacity divided by total capacity
TAC	technical advisory committee
Therm	quantity of energy equal to 100,000 Btu, normally applied to gas consumption (1 Therm ≈ 100 SCF)
Ton	a unit of capacity equal to 12,000 Btu/hr or 200 Btu/min
VCHP	variable capacity heat pump
WB	wet bulb temperature (as in Twb)

NOMENCLATURE

A	Surface area of the ducts [m2]
COP	Coefficient of performance [−]
Cp	Specific heat of air at constant pressure [kJ/kg K]
f ()	Function of
F	leakage fraction
h	Specific enthalpy of air [kJ/kg]
$\bar{h}$	Average specific enthalpy [kJ/kg]
P	Power consumed [kW]
Pr	Pressure [Pa]
$\dot{m}$	Mass flow rate of air [kg/s]
ξ	Effectiveness [–]
$\dot{Q}$	Capacity [kW]
T	Temperature [°C]
R	Overall heat transfer resistance [K m2/W]
U	Overall heat transfer coefficient [W/m2K]
$\dot{V}$	Volume flow rate [m3/s]

SUBSCRIPTS

atm	Atmospheric
br	branches of each supply duct
cool	During cooling season
comp	Compressor
cond	Condenser
db	Dry bulb
del	Delivery
dp	Dew point
equip	Equipment
evap	Evaporator
grille	At a single grille
grilles	Of all the grilles
leakage	Leakage in the supply ducts
ret	return duct
room	In the conditioned space
rp	Return plenum
sp	Supply plenum
sys	System
t	Supply trunk
wb	Wet bulb
v	Specific volume

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APPENDIX A:
Phase 1 PG&E Testing Instrumentation

Table A-1: PG&E Instrumentation List

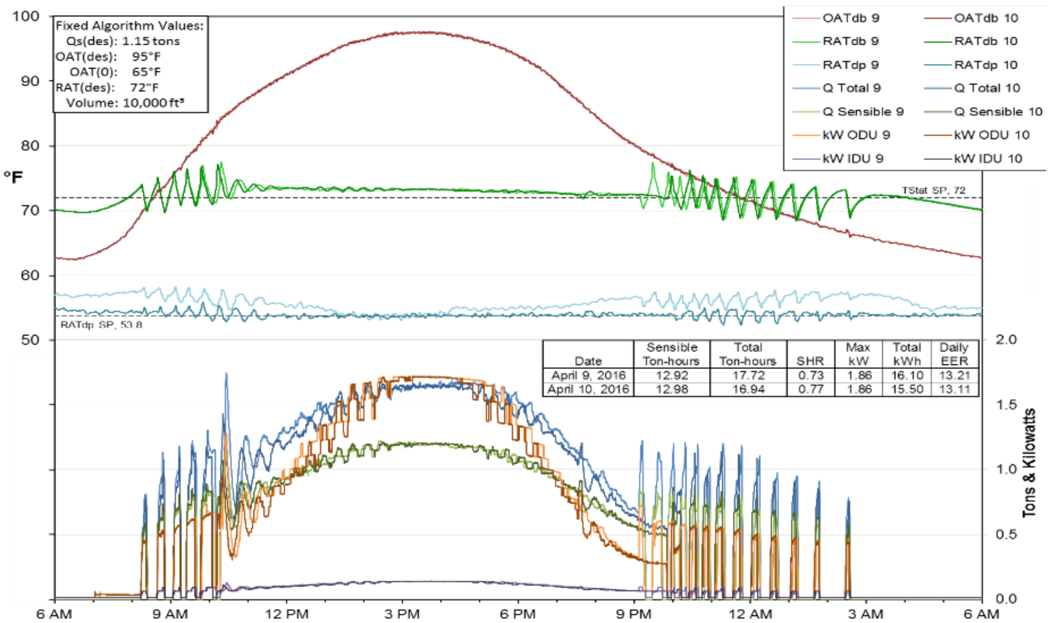
Measurement	Instrument	Make	Accuracy
Barometric Pressure	Multi-function weather station on roof of building	Vaisala WTX520	±0.007 PSIA (±50 Pa)
Return air dry-bulb temperature	Average of 9 Type-T thermocouples arrayed in a 3×3 grid across the intake duct. Plus, one RTD for reference.	Therm-X Burns Engineering	±0.5°F ±0.2°F
Return air dew-point temperature	Chilled mirror dew point sensor	General Electric Optica	±0.36°F
Supply air dry-bulb temperature	Average of 9 Type-T thermocouples arrayed in a 3×3 grid across the supply duct and downstream of mixing device. Plus, one RTD for reference	Therm-X Burns Engineering	±0.5°F ±0.2°F
Supply air discharge dew-point temperature	Chilled mirror dew point sensor	General Electric Optica	±0.36°F
Supply air static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the duct leaving the unit	Rosemount 3051C	±0.04% of span (–3 to 3 IW)
Supply-return differential pressure	Pressure transmitter connected between the supply and return duct manifolds.	Rosemount 3051C	±0.04% of span (–4 to 4 IW)
Supply airflow station upstream static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the flow box upstream of the nozzle partition	Rosemount 3051C	±0.04% of span (–1 to 3 IW)
Supply airflow station differential pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the flow box on both sides of the nozzle partition	Rosemount 3051C	±0.04% of span (0 to 4 IW)
Supply airflow station dry bulb temperature	Single fast-response RTD upstream of nozzles	Burns Engineering	±0.2°F
Outside air/ condenser intake dry-bulb temperature	Average of 16 fast-response resistance temperature detectors (RTDs) arrayed across the condenser air intake (4 per face).	Burns Engineering	±0.2°F

Measurement	Instrument	Make	Accuracy
Outside air dew-point temperature	Chilled mirror dew point sensor	General Eastern Hygro-M2+	±0.36°F
Exhaust air dry-bulb temperature	Average of 9 fast-response RTDs arrayed in a 3×3 grid across the duct attached to test unit exhaust	Burns Engineering	±0.2°F
Exhaust air dew-point temperature	Chilled mirror dew point sensor	General Eastern Hygro-M2	±0.36°F
Exhaust air static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the duct leaving the unit	Rosemount 3051C	±0.04% of span (-2 to 2 IW)
Exhaust airflow station upstream static pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the flow box upstream of the nozzle partition	Rosemount 3051C	±0.04% of span (-2 to 2 IW)
Exhaust airflow station differential pressure	Pressure transmitter attached to manifolded pressure taps at center of each side of the flow box on both sides of the nozzle partition	Rosemount 3051C	±0.04% of span (0 to 4 IW)
Exhaust airflow station dry bulb temperature	Single fast-response RTD upstream of nozzles	Burns Engineering	±0.2°F
Compressor suction pressure	Pressure transmitter attached to compressor suction Schrader valve (extra tap on outdoor unit)	Rosemount 3051C	±0.04% of span (0 to 300 psig)
Outdoor unit vapor line pressure	Pressure transmitter attached to vapor line Schrader valve	Rosemount 3051C	±0.04% of span (0 to 300 psig)
Outdoor unit liquid line pressure	Pressure transmitter attached to liquid line Schrader valve	Rosemount 3051C	±0.04% of span (0 to 400 psig)
Refrigerant temperatures (compressor suction and discharge, liquid and vapor lines at both indoor and outdoor units, and indoor coil)	Type-T thermocouples (7 total) clamped to outside of refrigerant tubing, coated with thermal paste and wrapped in insulation	Therm-X	±0.5°F
Refrigerant liquid flow rate	Coriolis mass flow meter (bidirectional)	Micro-Motion R025S	0.5% mass flow
Unit Supply Power, Voltage and Current	3-element true-RMS power meter with outputs for total power, voltage and current. One element used for the outdoor unit, and a second used for the indoor unit.	Yokogawa WT330	±(0.1% of reading +0.1% of range)

Measurement	Instrument	Make	Accuracy
Blower/Fan Speeds	Optical tachometers (2) reading reflective tape on indoor blower and condenser fan.	Monarch ACT-1B	±1 RPM or 0.005% of reading
Gas Quantity	Diaphragm meter with pulse output (2000 counts per cubic foot)	American Meter AC-250 with IMAC pulse head	
Gas Flow Rate	Thermal mass flow meter	Sierra SmartTrak 100	±1.0% of full scale
Temperature Calibration Reference Standard	Electronic thermometer	Fluke 1502A	±0.015°F
High Pressure Calibration Reference Standard	Pressure calibration standard	Condec UPC5200	±0.05% of f.s.
Low Pressure Calibration Reference Standard	Precision manometer	Dwyer 1430	±0.00025 I.W.

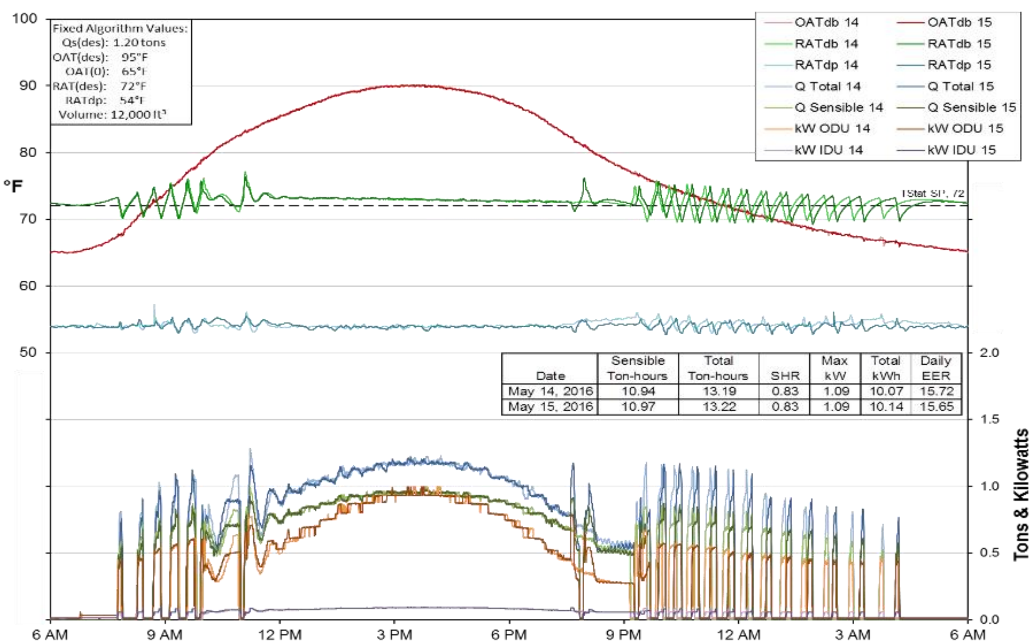
APPENDIX B:
Phase 1 PG&E Additional Figures

Figure B-1: Dynamic Cooling Test #6 (with Realistic Temperature Profile)



Source: PG&E

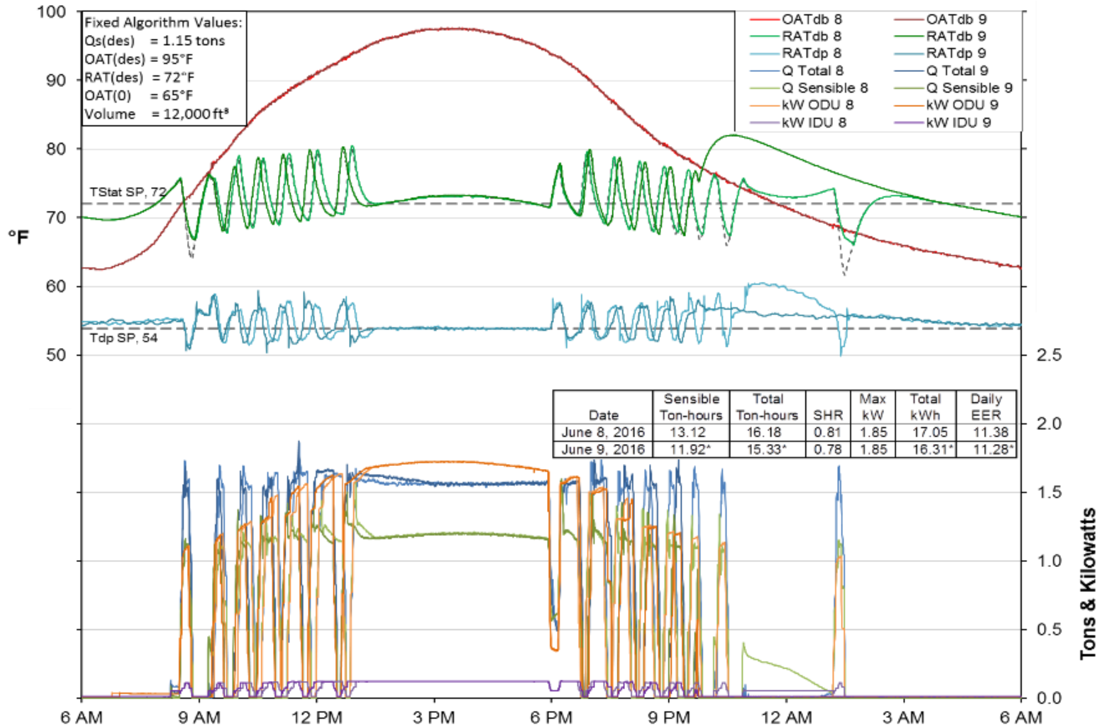
Figure B-2: Dynamic Cooling Test #8 (with Realistic Temperature Profile)



Source: PG&E

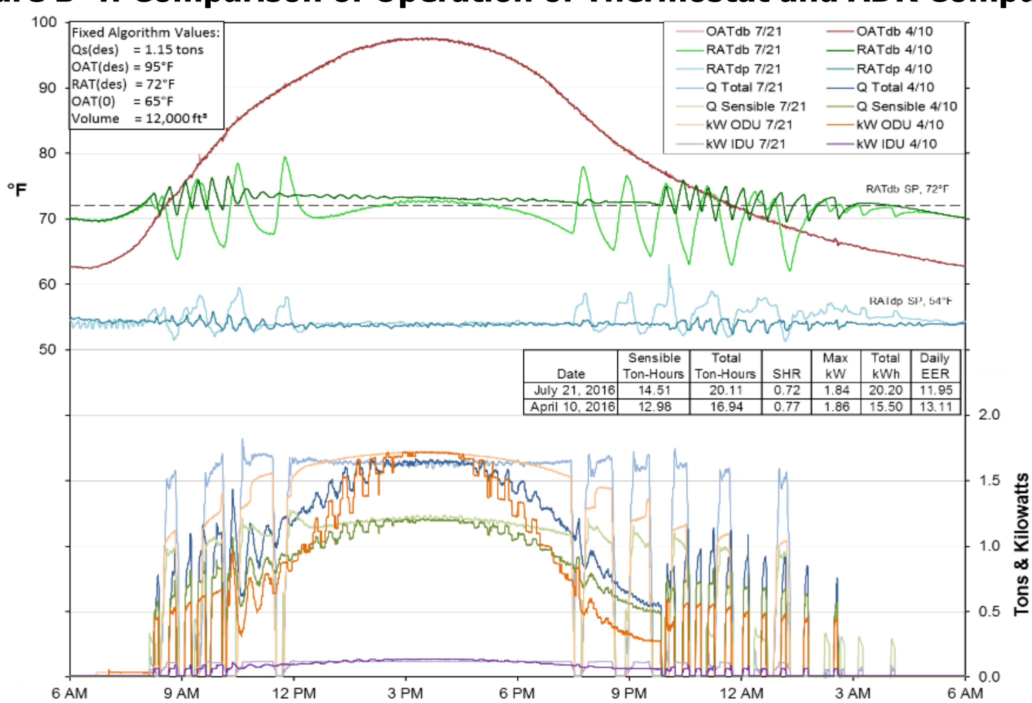
Figure B-3: Dynamic Cooling Test #10 (with Realistic Temperature Profile)

Unit Controlled by ADR Computer rather than Thermostat



Source: PG&E

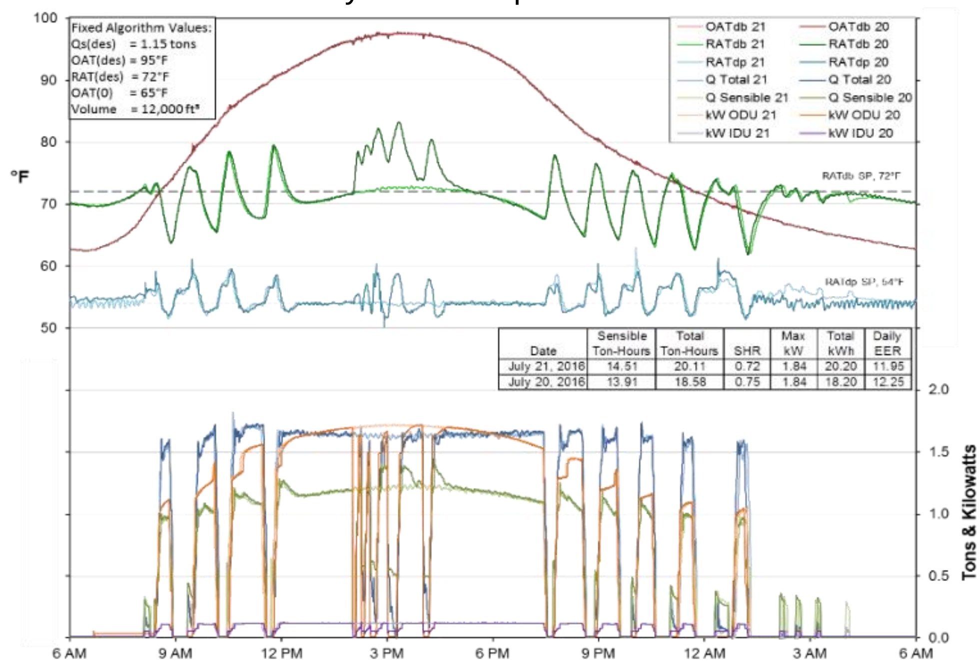
Figure B-4: Comparison of Operation of Thermostat and ADR Computer



Source: PG&E

Figure B-5: Dynamic Cooling Test #12 (with Realistic Temperature Profile)

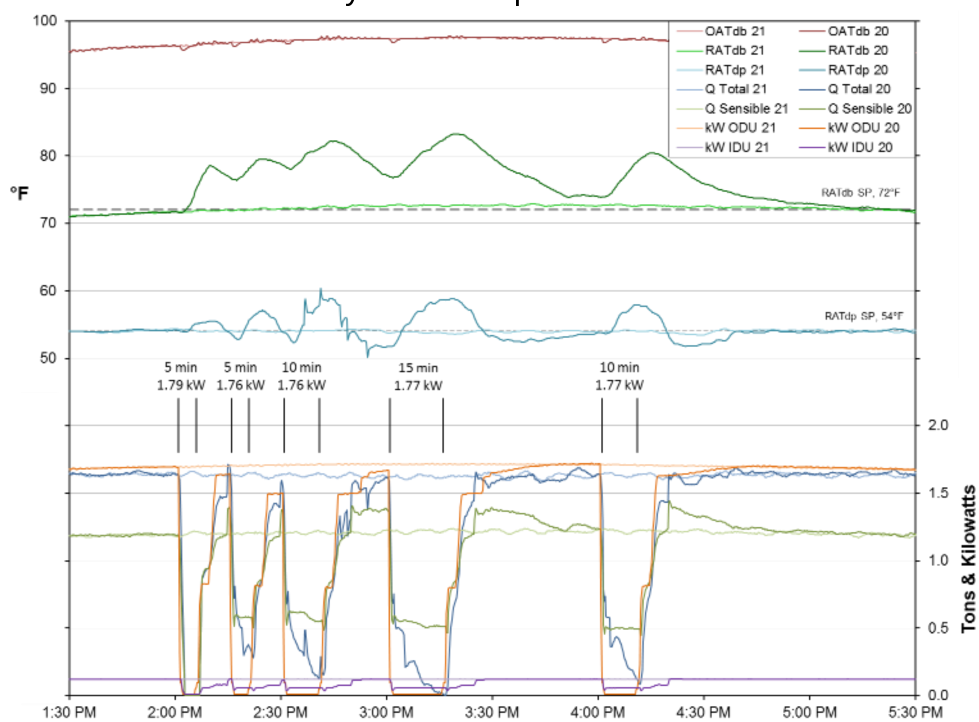
Unit Controlled by ADR Computer with five 0% Events



Source: PG&E

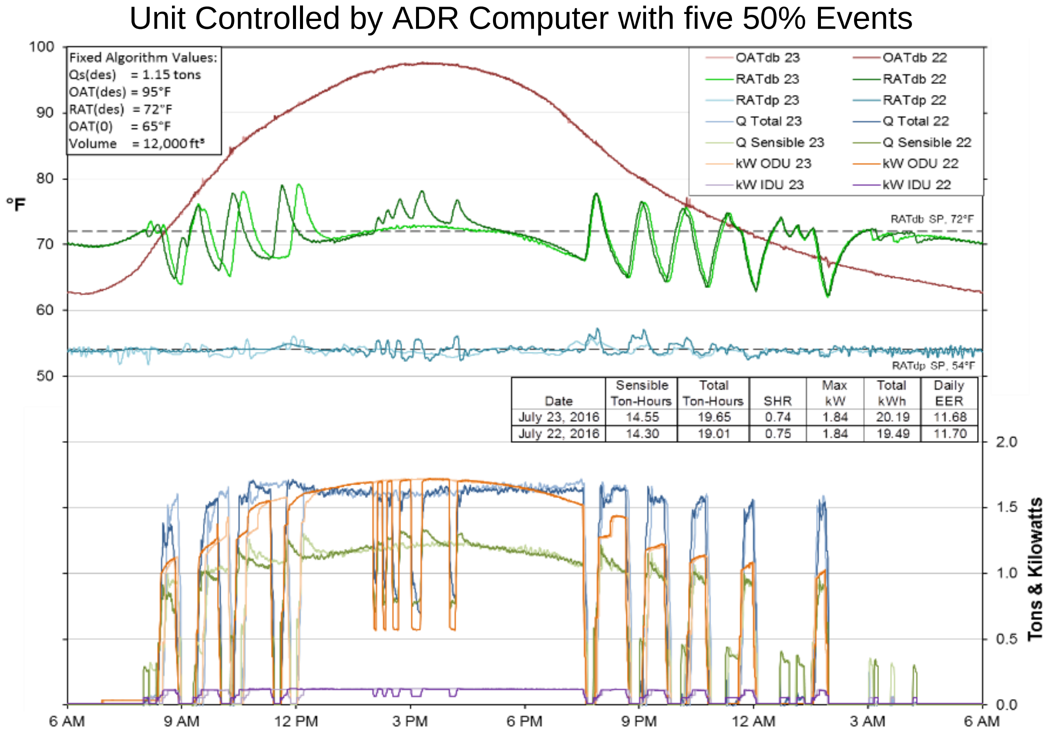
Figure B-6: Detail of Dynamic Cooling Test #12 (with Realistic Temperature Profile)

Unit Controlled by ADR Computer with five 0% Events



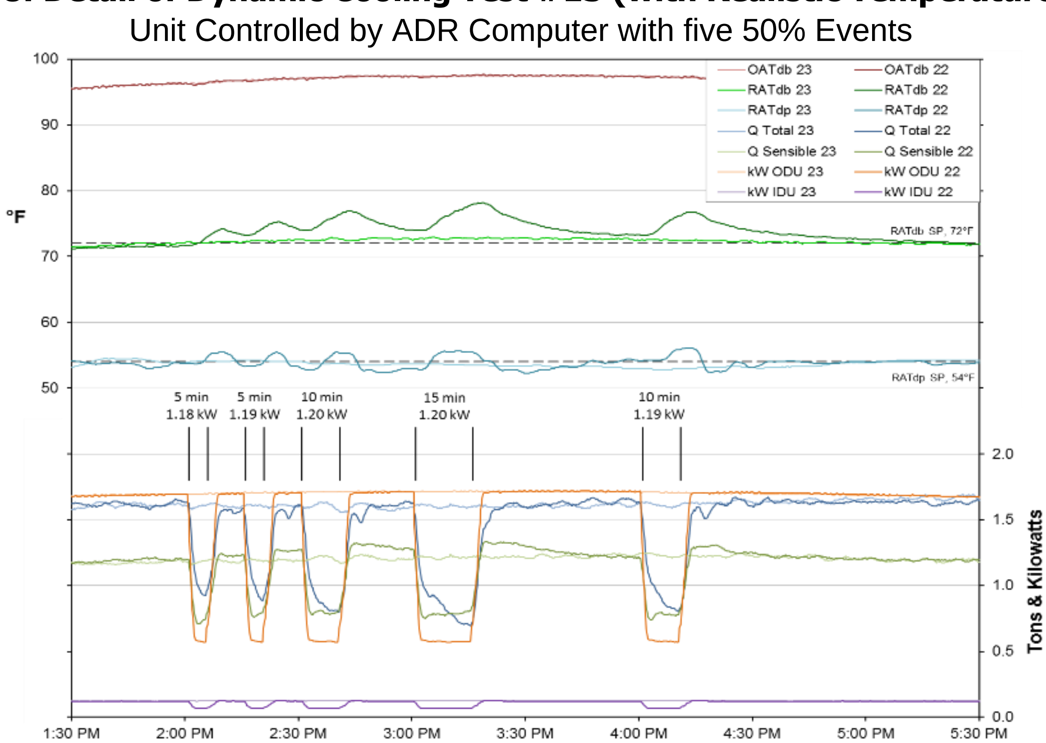
Source: PG&E

Figure B-7: Dynamic Cooling Test #13 (with Realistic Temperature Profile)



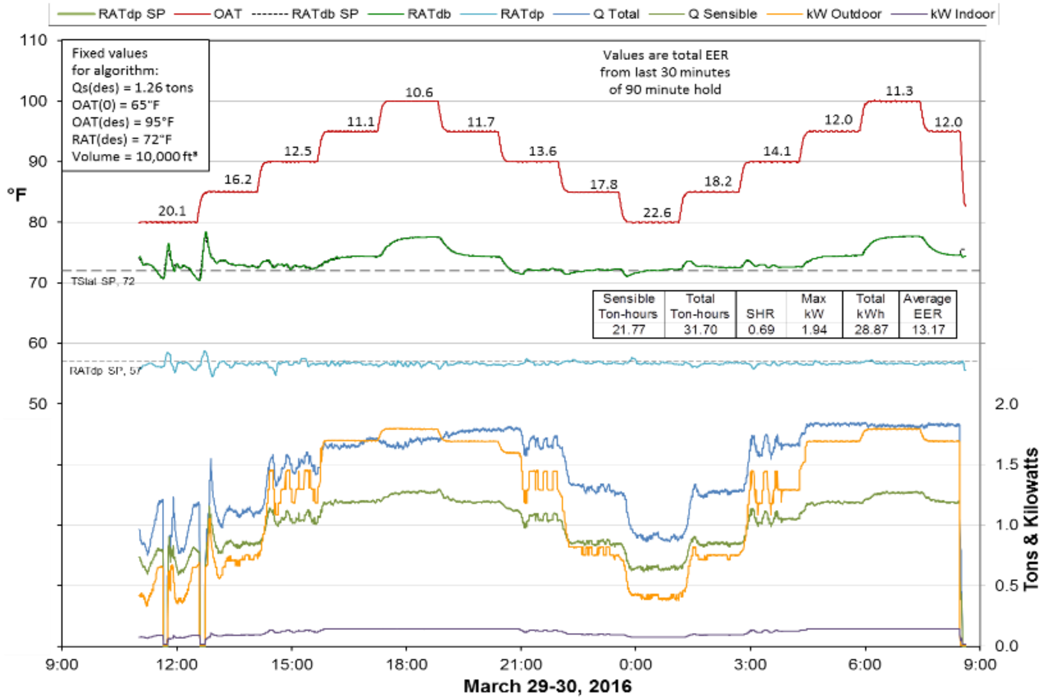
Source: PG&E

Figure B-8: Detail of Dynamic Cooling Test #13 (with Realistic Temperature Profile)



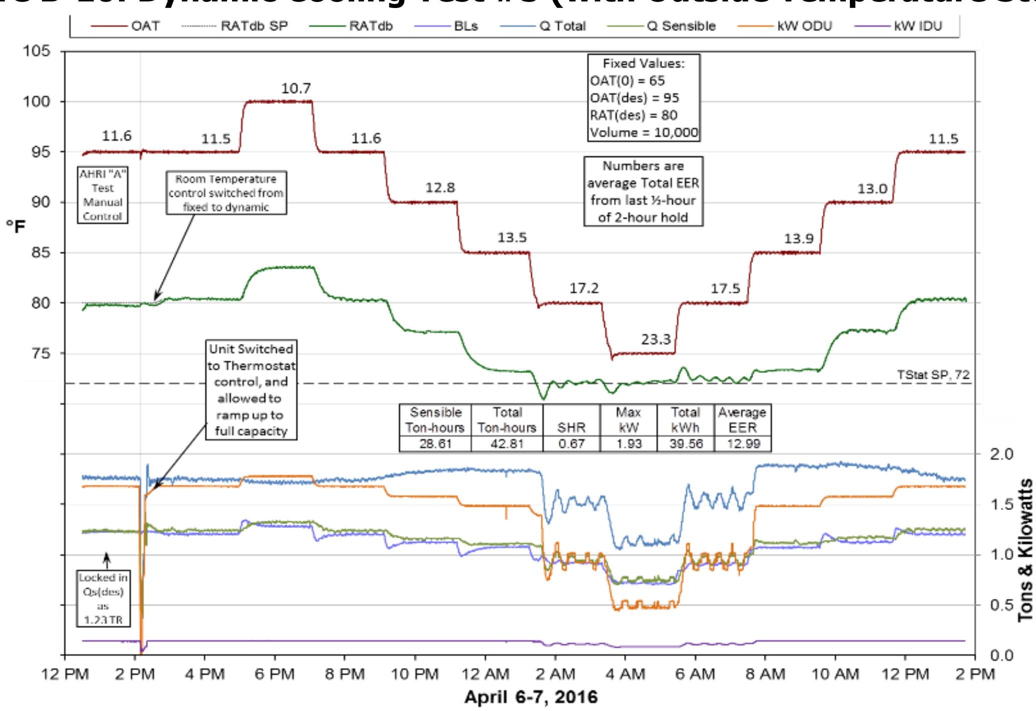
Source: PG&E

Figure B-9: Dynamic Cooling Test #3 (with Outside Temperature Steps)



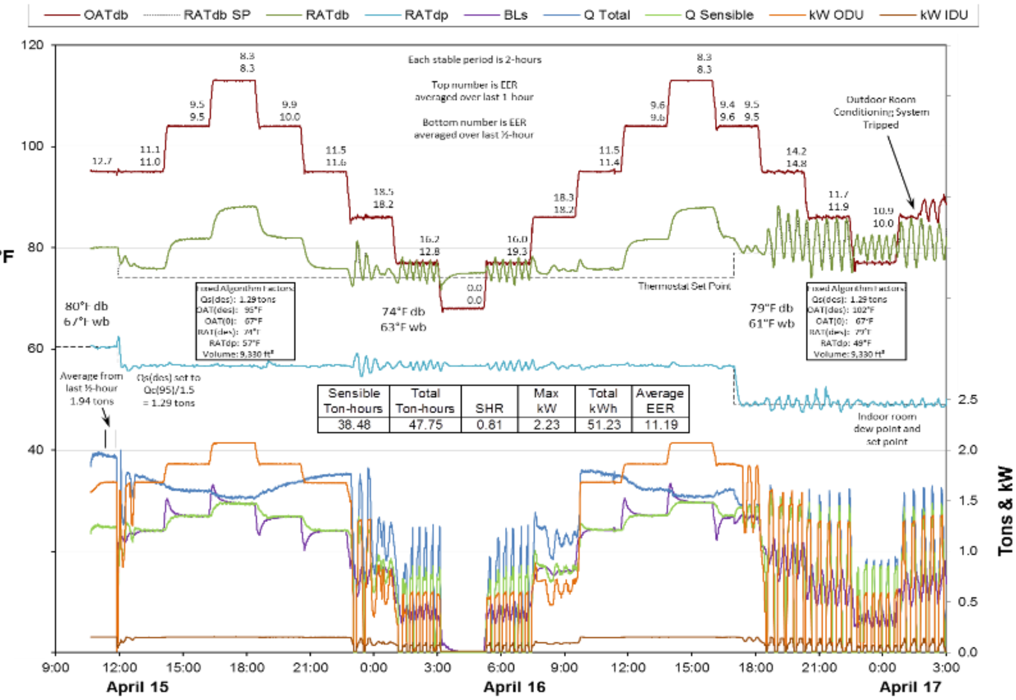
Source: PG&E

Figure B-10: Dynamic Cooling Test #5 (with Outside Temperature Steps)



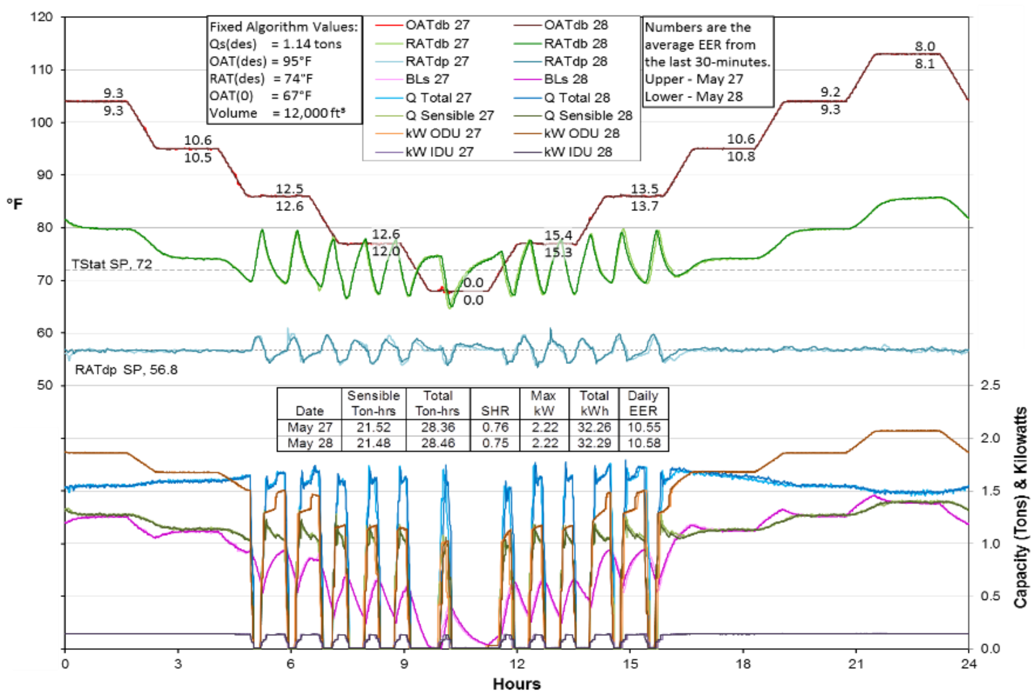
Source: PG&E

Figure B-11: Dynamic Cooling Test #7 (with Outside Temperature Steps)
Attempted Trial of Proposed CSA Standard Test Points



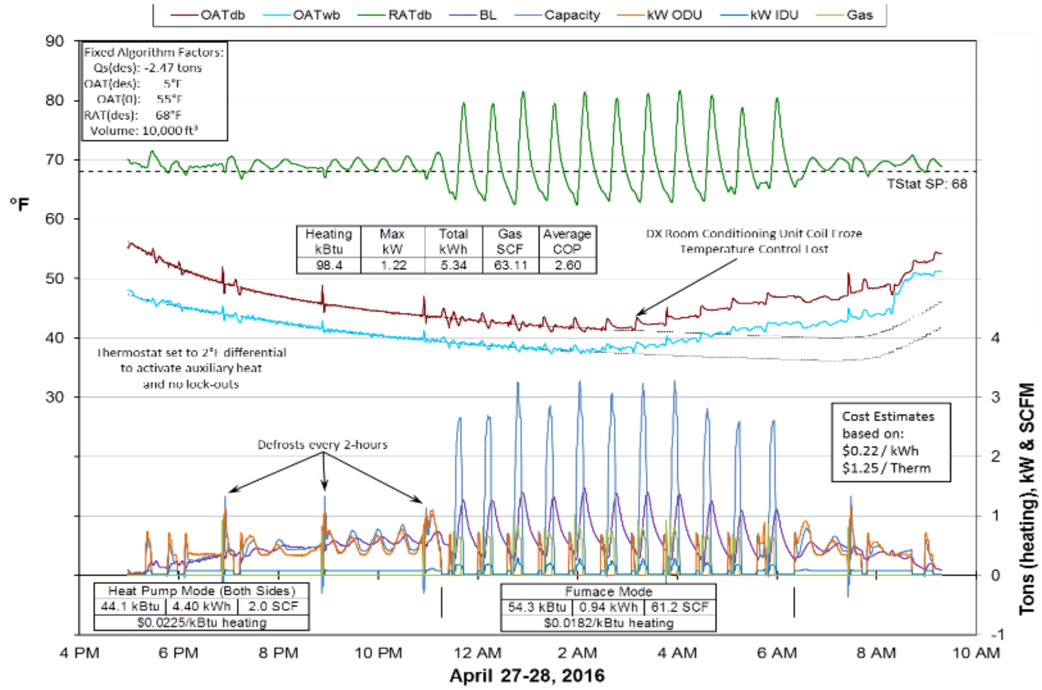
Source: PG&E

Figure B-12: Dynamic Cooling Test #9 (with Outside Temperature Steps)
Overlapping Daily Profiles - Unit Controlled by ADR Computer rather than Thermostat



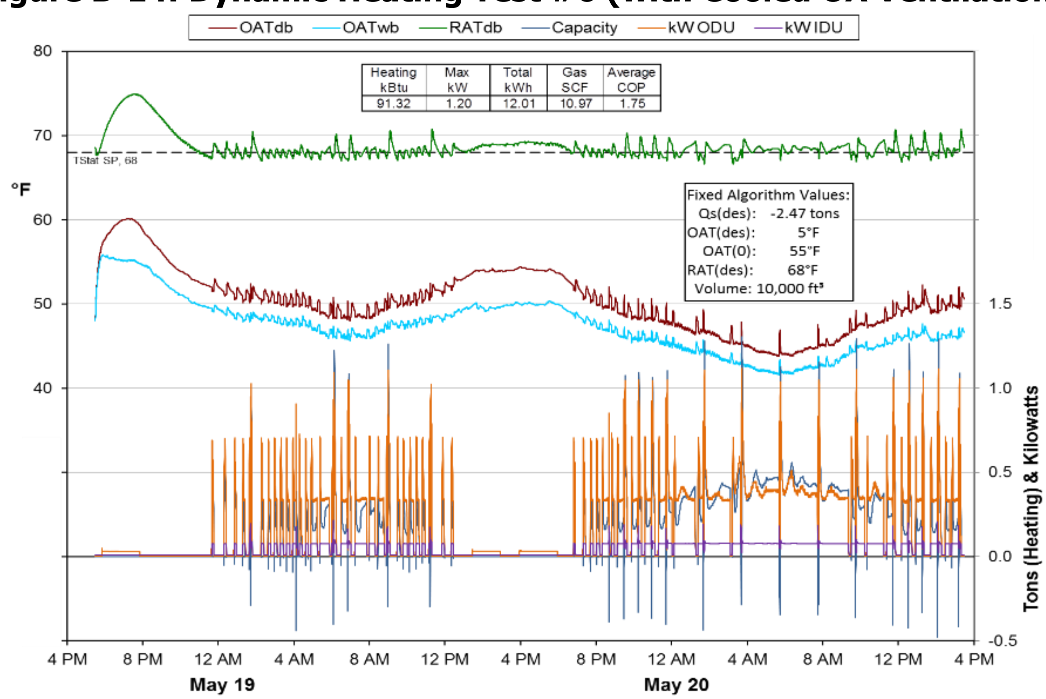
Source: PG&E

Figure B-13: Dynamic Heating Test #3 (with Realistic Temperature Profile)



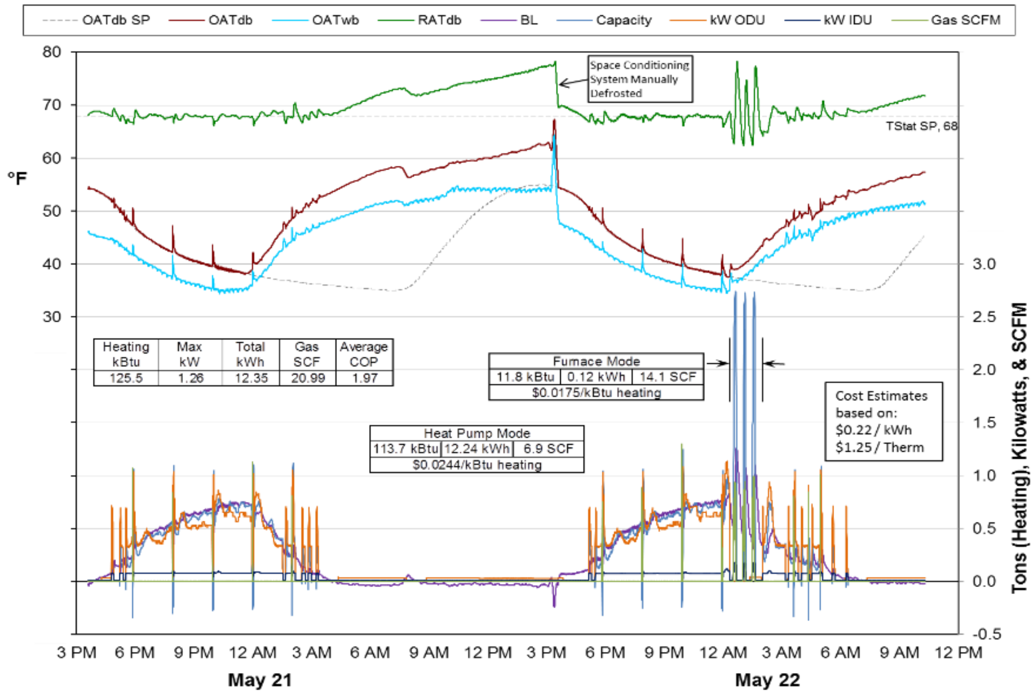
Source: PG&E

Figure B-14: Dynamic Heating Test #6 (with Cooled OA Ventilation)



Source: PG&E

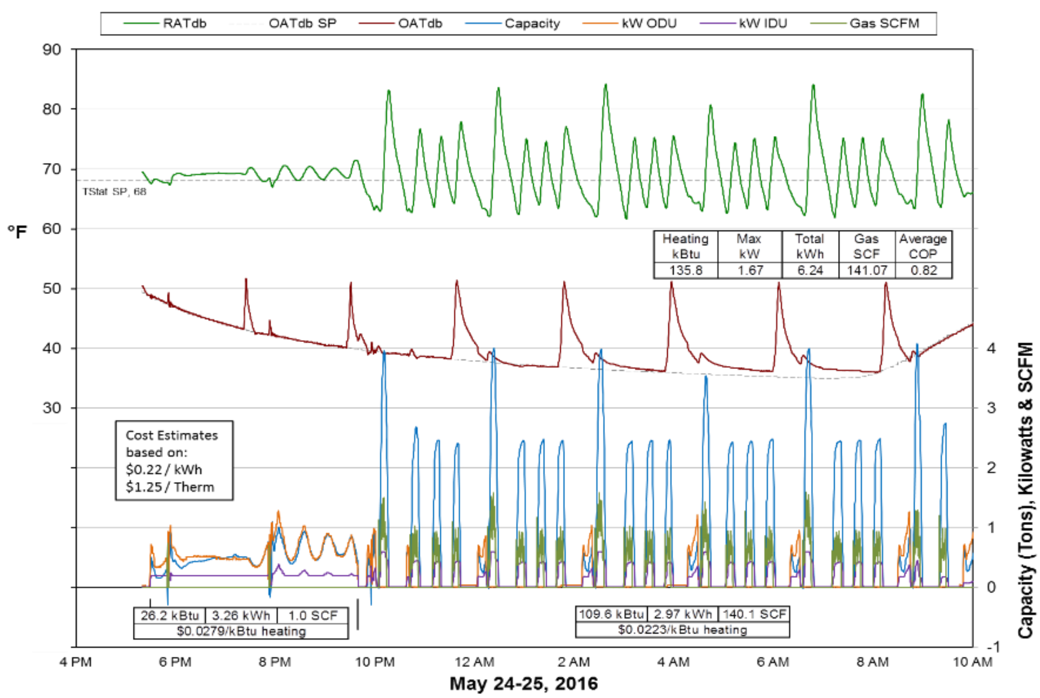
Figure B-15: Dynamic Heating Test #7 (with Realistic Temperature Profile)



Source: PG&E

Figure B-16: Dynamic Heating Test #8 (with Realistic Temperature Profile)

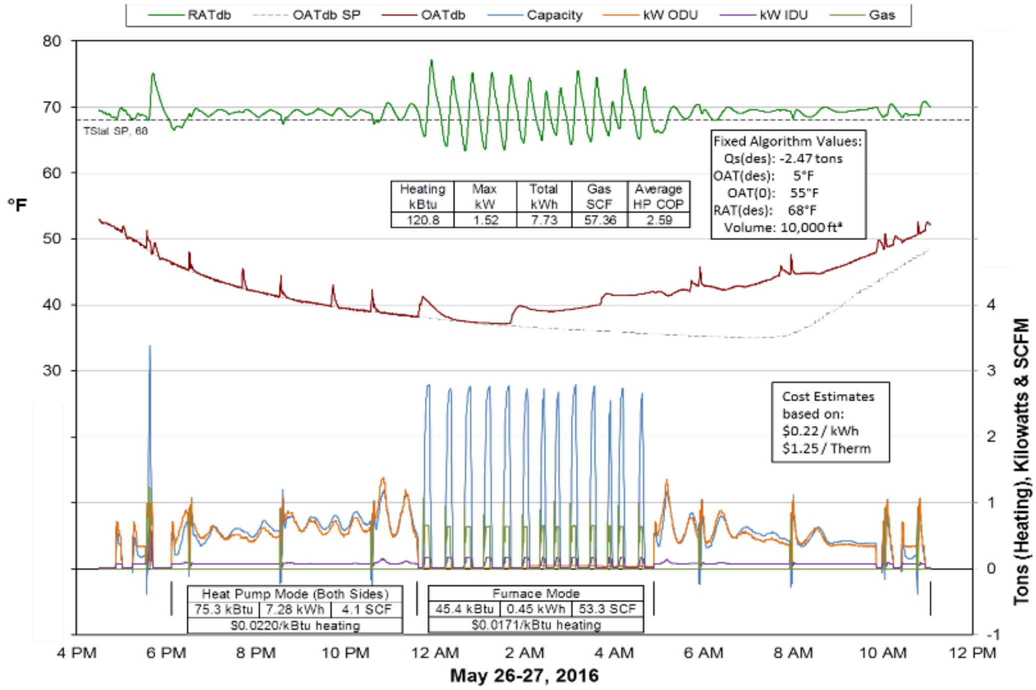
First attempt of defrost technique for space conditioning apparatus



Source: PG&E

Figure B-17: Dynamic Heating Test #9 (with Realistic Temperature Profile)

Second attempt of defrost technique for space conditioning apparatus



Source: PG&E

APPENDIX C: Phase 1 Dual Fuel Heat Pump Analysis and Testing

The dual fuel heat pump function of the Next-Generation Residential Space Conditioning System for California uses a variable speed electric heat pump as the primary heating means and a gas furnace for supplemental heat and when it is more economical to run the furnace. The following paragraphs and tables describe the operation and energy costs of the Next Gen DFHP for range of operating conditions and utility prices.

An Excel spreadsheet program was developed and utilized to perform the required analyses; determining the balance point temperature and the breakeven temperature, calculating heat pump and gas furnace heating costs and comparing these costs.

The building heating load was 44,867 Btu/hr @ 12°F and zero @ 62°F. The furnace capacity was 80,000 Btu/hr Next Gen heat pump capacity varied with speed with a capacity of 21,167 Btu/hr @ 37°F and 13,773 Btu/hr @ 17°F at maximum speed. Above the balance point the Next Gen DFHP varies the speed of the unit to provide the capacity to match the building heating load.

The balance point temperature, the temperature at which the heating load exceeds the heat pump capacity, was approximately 37.8°F. Above the balance point the heat pump can handle the entire load and no backup is required. Below the balance point backup is required and the furnace must supply the shortfall in capacity.

Calculations were performed to determine the breakeven temperature based on heat pump and furnace efficiencies and the relative prices of electricity and natural gas. Above the breakeven temperature, it is more economical to run the heat pump and below the breakeven temperature it is more economical to run the furnace. For the Next Gen heat pump efficiencies and an AFUE of .97, the prices used in this analysis resulted in breakeven temperatures ranging from 22°F to 47°F.

There are several situations possible for operation of the electric heat pump and natural-gas fired furnace:

If the breakeven temperature is above the balance point, then the heat pump will run above the breakeven temperature and the furnace will run below the breakeven temperature.

If the breakeven temperature is below the balance point, then the heat pump will run exclusively above the balance point; the furnace will provide backup heat between the balance point and the breakeven temperature; and the furnace will operate exclusively below the breakeven temperature. (If the breakeven temperature is well below the design heating temperature it is likely that the furnace will never be called on as the primary means of space heating and will only be called on as backup heat.)

Climate Sensitivity of Savings

Climate data for the 16 California climate regions was used to calculate dual fuel heat pump and gas furnace heating costs and net DFHP annual heating cost savings for average California gas and electric prices. Results are shown in Table C-1.

Even though Bishop has the coldest climate, Oakland and Arcata experience greater savings because the heating hours in those climates are in the range where the heat pump is most efficient and provides lower heating cost than the gas furnace. The encouraging result of the calculations (and that of Napa, Santa Maria and to some extent San Jose as shown in Table C-1) is that Oakland's climate and others with reasonable savings correspond to a relatively large population. Thus, DFHP savings could accrue to a large segment of California.

Utility Price Sensitivity

Calculations were performed for a range of electricity prices with natural gas prices fixed at the State average of 129¢/therm. An electricity price of 18.8¢/kWh represented the state average found on the internet in early December when these calculations were started. 15.54¢/kWh is the state average as of this time and 13.03¢/kWh is the current price in Bishop. 20.46¢/kWh and 11.09¢/kWh were chosen to extend the range of prices and to yield 22 and 47F breakeven temperatures.

As shown below in Table C-2, DFHP savings increase as the electricity prices decrease since the decrease in electricity price lowers the breakeven temperature permitting the heat pump to economically displace the furnace over a greater temperature range. California energy prices tend to be high relative to National averages and natural gas prices have not kept pace with electric prices. Despite these factors there are still situations, as illustrated below, where the Next Gen DFHP can provide attractive savings.

The fact that operation of the DFHP can be adjusted as utility prices vary permits the homeowner the option to benefit from future changes in utility prices that might reduce the ratio of electricity to gas prices in the future. The assurance that the homeowner will be able to experience the lowest future heating costs possible, is an important attribute of dual fuel heat pump capability and increases the value of this feature to potential purchasers of the Next Gen Residential Space Conditioning System.

Table C-1: Sensitivity of Next-Gen RSCS DFHP Energy Costs to Climate Average Residential Gas and Electric Prices, Balance Point =37.8°F

Location/Climate Zone	Furnace (AFUE)	Gas Price (¢/Therm)	Electric Price (¢/kWh)	Breakeven Temperature (°F)	Annual DFHP Heating Cost	Annual Furnace Cost	DFHP Annual Savings
Bishop (16)	0.97	129[1]	15.54[2]	40.65	**$1,213**	**$1,300**	**$87**
Blythe (15)	0.97	129	15.54	40.65	**$279**	**$355**	**$75**
29 Palms (14)	0.97	129	15.54	40.65	**$538**	**$632**	**$94**
Fresno (13)	0.97	129	15.54	40.65	**$664**	**$767**	**$103**
Stockton (12)	0.97	129	15.54	40.65	**$758**	**$877**	**$119**
Red Bluff (11)	0.97	129	15.54	40.65	**$809**	**$915**	**$106**
Riverside (10)	0.97	129	15.54	40.65	**$508**	**$629**	**$121**
Burbank (9)	0.97	129	15.54	40.65	**$415**	**$535**	**$120**
Long Beach (8)	0.97	129	15.54	40.65	**$333**	**$444**	**$110**
San Diego (7)	0.97	129	15.54	40.65	**$256**	**$358**	**$102**
Los Angeles (6)	0.97	129	15.54	40.65	**$303**	**$420**	**$116**
Santa Maria (5)	0.97	129	15.54	40.65	**$855**	**$1,031**	**$176**
San Jose (4)	0.97	129	15.54	40.65	**$630**	**$777**	**$148**
Oakland (3)	0.97	129	15.54	40.65	**$678**	**$873**	**$195**
Napa (2)	0.97	129	15.54	40.65	**$901**	**$1,073**	**$172**
Arcata (1)	0.97	129	15.54	40.65	**$1,218**	**$1,434**	**$217**

[1] Average California residential gas prices http://utilitieslocal.com/states/california/bishop/
[2] Average California residential electric prices http://www.electricitylocal.com/states/california/

Souce: EPRI

Table C-2: Sensitivity of DFHP Savings to Electricity Prices for Bishop, Oakland and Arcata (Balance Point = 37.8°F)

Location/Climate Zone	Furnace AFUE	Natural Gas Price (¢/Therm)	Electric Price (¢/kWh)	Breakeven Temperature (°F)	Annual DFHP Heating Cost	Annual Furnace Cost	DFHP Annual Savings
Bishop (16)	0.97	129[1]	20.46	47	**$1,290**	**$1,300**	**$10**
Bishop (16)	0.97	129	18.8[2]	43.6	**$1,258**	**$1,300**	**$42**
Bishop (16)	0.97	129	15.54[3]	40.65	**$1,213**	**$1,300**	**$87**
Bishop (16)	0.97	129	13.03[4]	32	**$1,132**	**$1,300**	**$168**
Bishop (16)	0.97	129	11.09	22	**$984**	**$1,300**	**$316**
Oakland (3)	0.97	129	20.46	47	**$838**	**$873**	**$35**
Oakland (3)	0.97	129	18.8	43.6	**$779**	**$873**	**$94**
Oakland (3)	0.97	129	15.54	40.65	**$678**	**$873**	**$195**
Oakland (3)	0.97	129	13.03	32	**$576**	**$873**	**$297**
Oakland (3)	0.97	129	11.09	22	**$491**	**$873**	**$382**
Arcata (1)	0.97	129	20.46	47	**$1,411**	**$1,434**	**$23**
Arcata (1)	0.97	129	18.8	43.6	**$1,336**	**$1,434**	**$98**
Arcata (1)	0.97	129	15.54	40.65	**$1,218**	**$1,434**	**$217**
Arcata (1)	0.97	129	13.03	32	**$1,050**	**$1,424**	**$384**

[1] **Average California residential gas prices http://utilitieslocal.com/states/california/bishop/**
[2] **Average California residential electric prices https://www.eia.gov/state/rankings/?sid=CA#series/28; also http://utilitieslocal.com/states/california/bishop/ for September 2016**
[3] **Average California residential electric prices http://www.electricitylocal.com/states/california/**
[4] **Lowest residential electric prices found for Bishop http://www.electricitylocal.com/states/california/bishop/**

Source: EPRI

DFHP Equipment Cost

Previous unpublished studies performed by the investigator for a large Southern U.S. utility in 2009 found the cost differential between a DFHP and a comparable furnace/air conditioning unit was about $300/ton. Escalating this using an inflation rate of 2% per year, results in a cost differential of about $350/ton in current dollars.

Comparing this incremental equipment cost of $350/ton to the savings in Table C-2 for the 2-ton Next Gen DFHP, indicates that reasonable payback periods could be achieved in some of the climate zones in California for the DFHP feature. For example, using 15.54¢/kWh for Arcata provides savings of $217 and a payback period of a little over 3 years. Using 15.54¢/kWh for Oakland provides savings of $195 and a payback period of about 3.6 years. Using 13.03¢/kWh for Bishop provides savings of $168 and a payback period of about 4.2 years.

Furnace Backup

Since the dual fuel heat pump diverts a lot of the potential hours of heating operation from the furnace to the heat pump, the economics of using a condensing furnace versus a non-condensing furnace for the DFHP may be questionable. The following paragraphs and tables explore this question.

Table C-3 presents the savings due to use of a DFHP with furnace backup having an AFUE of 0.97 (condensing furnace) and an AFUE (non-condensing furnace) of 0.786. Conditions favorable to furnace operation (electricity prices of 18.8¢/kWh) were selected for the comparisons. Comparing the results for AFUE = 0.97 and AFUE = 0.786 for the four climates illustrated, shows increased annual savings due to use of the condensing furnace ranging from $68 in Bishop to $180 in Arcata to $157 in Oakland. Some of the savings from use of a condensing furnace versus a non-condensing furnace ($303 for Bishop, $236 for Arcata and $204 for Oakland) is offset by use of the heat pump option above the breakeven temperature.

Functional Testing of the Next Gen DFHP

The DFHP control algorithms in the Next Gen DFHP were tested in the laboratory by setting the breakeven temperature to a value of 30°F with a balance point of around 38°F. This provided a situation where the breakeven temperature was below the balance point.

The heat pump ran exclusively at 40°F (above the balance point).

The furnace provided backup heat between the balance point and the breakeven temperature when the heat pump capacity was insufficient to raise the room temperature. In backup mode, after the unit switched to furnace operation, the furnace operated exclusively heating until the thermostat was satisfied. This cycle was repeated when the room temperature fell below the thermostat set point.

Below the breakeven-temperature the heat pump was locked out and only the furnace was operated to meet the load.

These tests confirmed the functional operation of the DFHP.

Table C-3: Influence of Furnace AFUE on DFHP Energy Savings

Location/Climate Zone	Furnace (AFUE)	Gas Price (¢/Therm)	Electric Price (¢/kWh)	Breakeven Temperature (°F)	Annual DFHP Heating Cost ($)	Annual Furnace Cost ($)	DFHP Annual Savings ($)	DFHP Savings Furnace AFUE 0.97 vs AFUE 0.786
Bishop (16)	0.97	129	18.8	43.6	$1,258	$1,300	**$42**	**$68**
Bishop (16)	0.786	129	18.8	39.6	$1,493	$1,603	**$110**	**—**
Arcata (1)	0.97	129	18.8	43.6	$1,336	$1,434	**$98**	**$180**
Arcata (1)	0.786	129	18.8	39.6	$1,492	$1,770	**$278**	**—**
Oakland (3)	0.97	129	18.8	43.6	$779	$873	**$94**	**$157**
Oakland (3)	0.786	129	18.8	39.6	$827	$1,077	**$251**	**—**

Source: EPRI

APPENDIX D: Phase 1 WCEC Summary of Results

Table D-1 and Table D-2 summarize results of the steady state experiments.

Table D-1: Summary of Results of Steady-State Experiments: Tests Conducted for Different Outdoor/Indoor Conditions with the Compressor Speed and Airflow Rates Kept in Sync

Room Air Condition (DB/WB, °F)	Outdoor Air Condition (DB, °F)	Compressor and Blower Speed (% of Maximum Speeds)	Measured Evaporator Airflow rate (CFM)	Power (kW) Indoor Unit	Power (kW) Outdoor Unit	COP of Compress or Only	Equip-ment COP	Overall System COP	Delivery Effectiveness
75/62.5	85	40	326	0.028	0.305	6.738	6.084	4.527	0.73
75/62.5	85	60	457	0.055	0.499	6.810	6.035	5.017	0.81
75/62.5	85	80	609	0.106	0.822	6.149	5.338	4.708	0.86
75/62.5	85	100	747	0.183	1.458	4.799	4.158	3.749	0.87
75/62.5	95	40	377	0.037	0.361	5.003	4.447	2.721	0.65
75/62.5	95	60	443	0.054	0.594	4.893	4.405	3.302	0.74
75/62.5	95	80	570	0.087	0.950	5.134	4.624	3.778	0.77
75/62.5	95	100	754	0.175	1.575	4.363	3.831	3.299	0.82
75/62.5	105	40	367	0.038	0.417	3.752	3.355	1.592	0.44
75/62.5	105	60	452	0.051	0.687	3.874	3.539	2.269	0.62
75/62.5	105	80	600	0.091	1.109	3.697	3.344	2.504	0.73
75/62.5	105	100	723	0.155	1.787	3.284	2.946	2.435	0.79
75/62.5	115	40	392	0.039	0.495	2.664	2.395	0.433	0.13
75/62.5	115	60	441	0.051	0.819	2.724	2.508	1.208	0.46
75/62.5	115	80	585	0.090	1.255	2.860	2.605	1.674	0.63
75/62.5	115	100	725	0.155	2.015	2.623	2.368	1.737	0.70
80/67	115	40	382	0.039	0.491	3.285	2.973	1.191	0.37
80/67	115	60	442	0.049	0.822	3.251	3.013	1.778	0.56
80/67	115	80	578	0.103	1.290	3.228	2.919	2.045	0.66
80/67	115	100	731	0.167	2.162	2.652	2.393	1.891	0.75

Source: WCEC

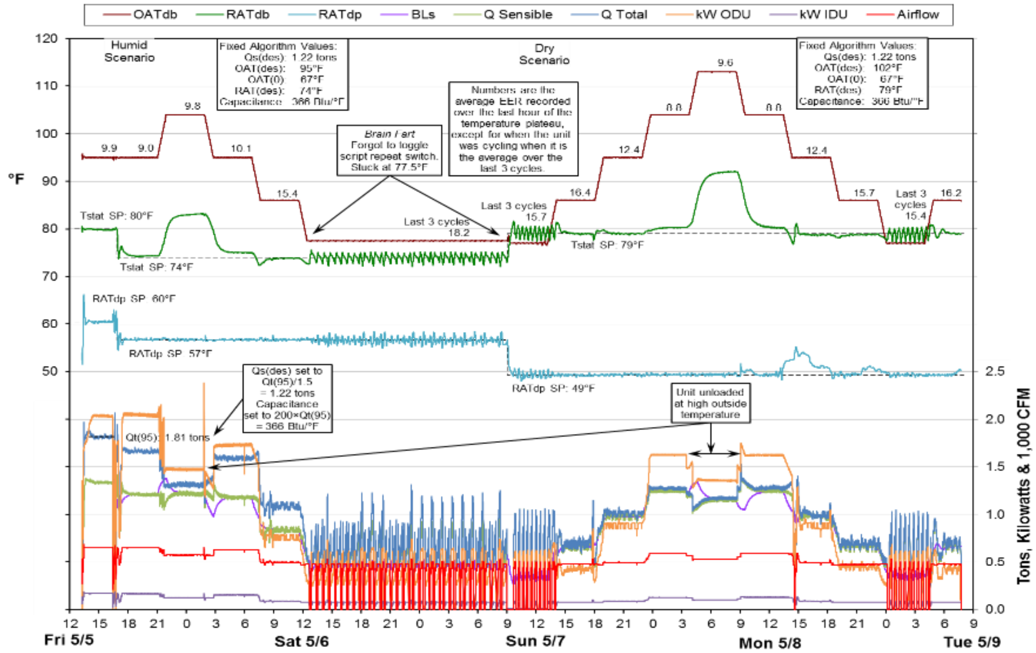
Table D-2: Summary of Results of Steady-State Experiments: Tests Conducted at 95°F DB Outdoor and 75°F DB/62.5°F WB Indoor with Varying Airflow Rates at a Fixed Compressor Speed

Room Air Condition (DB/WB, °F)	Outdoor Air Condition (DB, °F)	Compressor Speed (% of maximum speed)	Blower Speed (% of maximu m speed)	Measured Evaporator Airflow rate (CFM)	Power (kW)		Equipme nt COP	Overall System COP	Delivery effective-ness
					Indoor Unit	Outdoor Unit			
75/62.5	95	40	60	443	0.047	0.348	5.378	3.538	0.64
75/62.5	95	40	80	586	0.088	0.339	5.460	3.763	0.66
75/62.5	95	40	100	733	0.157	0.331	4.846	3.430	0.68
75/62.5	95	60	60	443	0.054	0.594	4.405	3.302	0.74
75/62.5	95	60	80	645	0.103	0.610	4.789	3.619	0.75
75/62.5	95	60	100	769	0.172	0.605	4.430	3.417	0.76
75/62.5	95	80	60	432	0.047	0.947	4.311	3.486	0.75
75/62.5	95	80	80	570	0.087	0.950	4.624	3.778	0.77
75/62.5	95	80	100	715	0.154	0.951	4.345	3.590	0.81
75/62.5	95	100	60	430	0.048	1.519	3.581	2.988	0.76
75/62.5	95	100	80	571	0.096	1.534	3.708	3.173	0.80
75/62.5	95	100	100	754	0.175	1.575	3.831	3.299	0.82

Source: WCEC

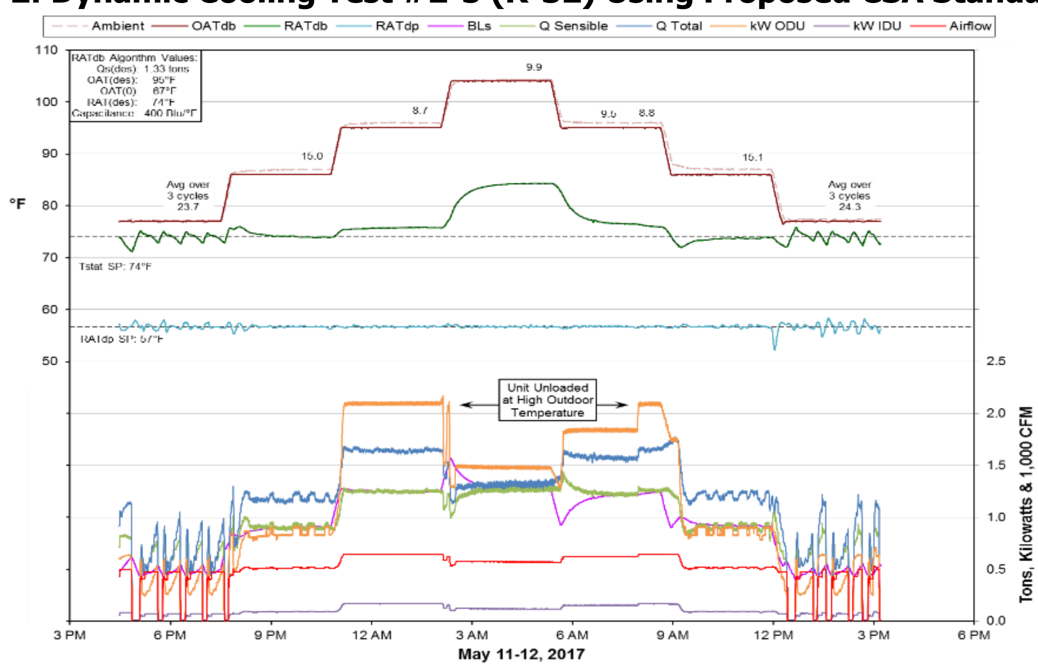
APPENDIX E: Phase 2 PG&E Additional Figures

Figure E-1: Dynamic Cooling Test #2-4 (R-32) Using Proposed CSA Standard Tests



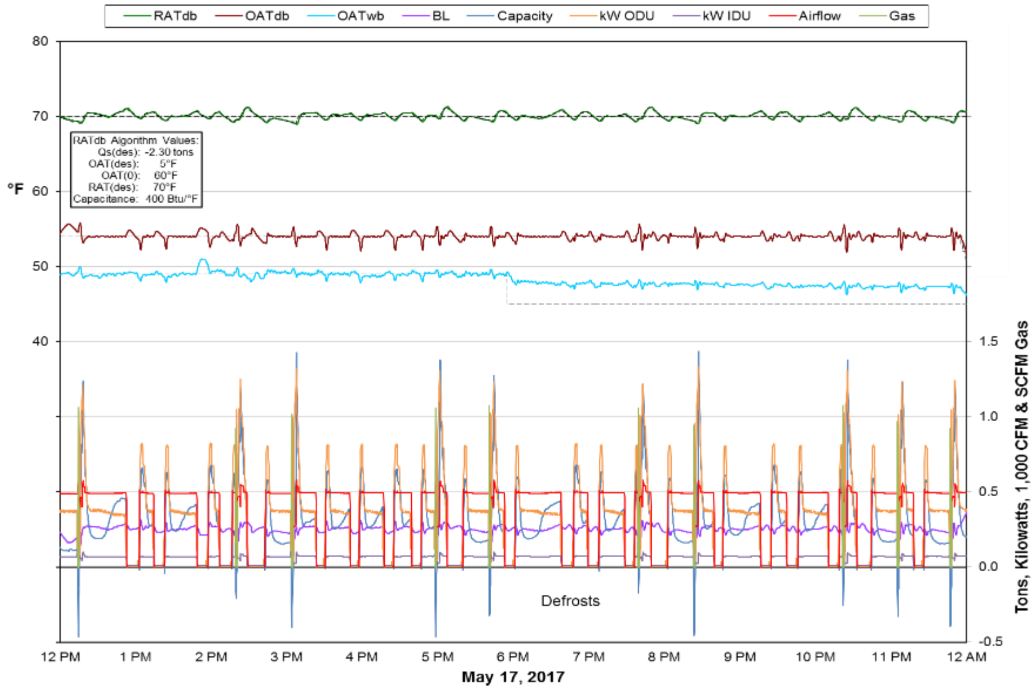
Source: PG&E

Figure E-2: Dynamic Cooling Test #2-5 (R-32) Using Proposed CSA Standard Tests



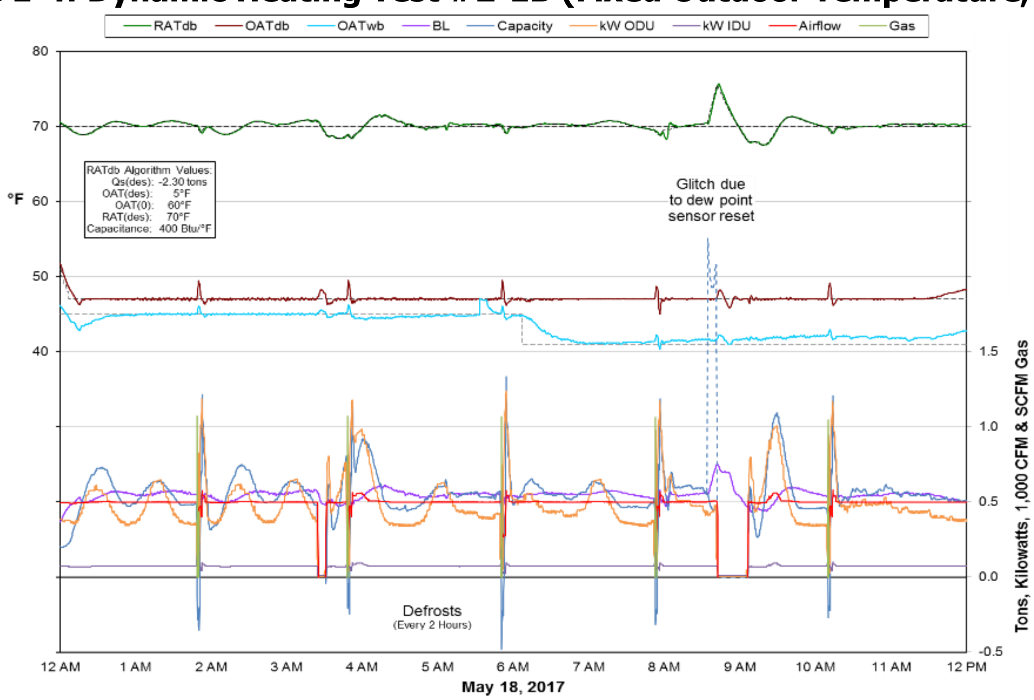
Source: PG&E

Figure E-3: Dynamic Heating Test #2-1A (Fixed Outdoor Temperature, R-32)



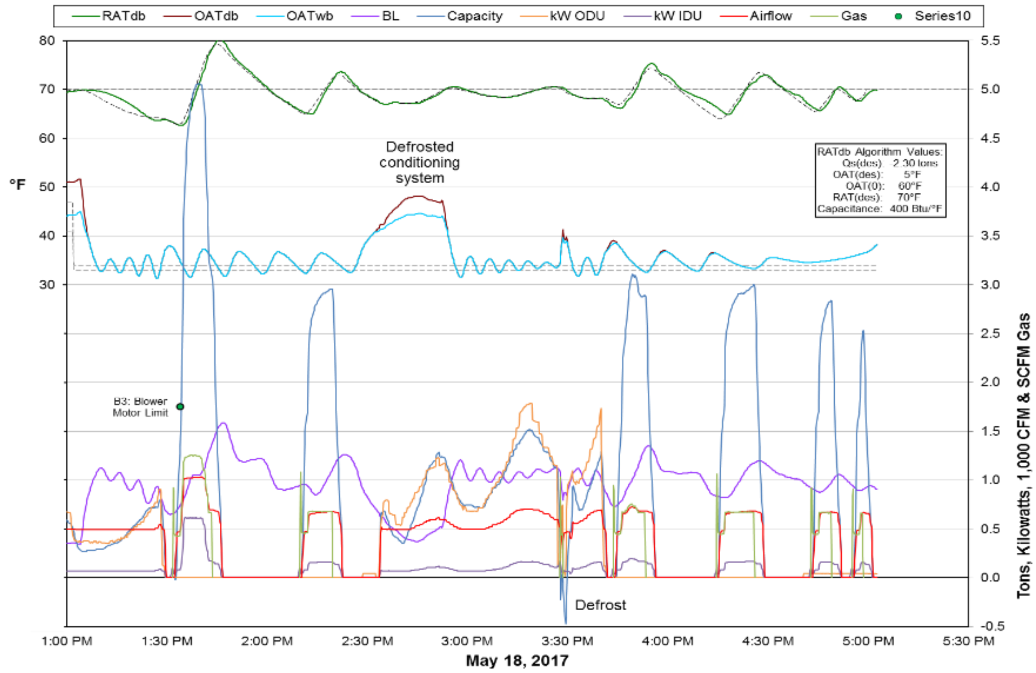
Source: PG&E

Figure E-4: Dynamic Heating Test #2-1B (Fixed Outdoor Temperature, R-32)



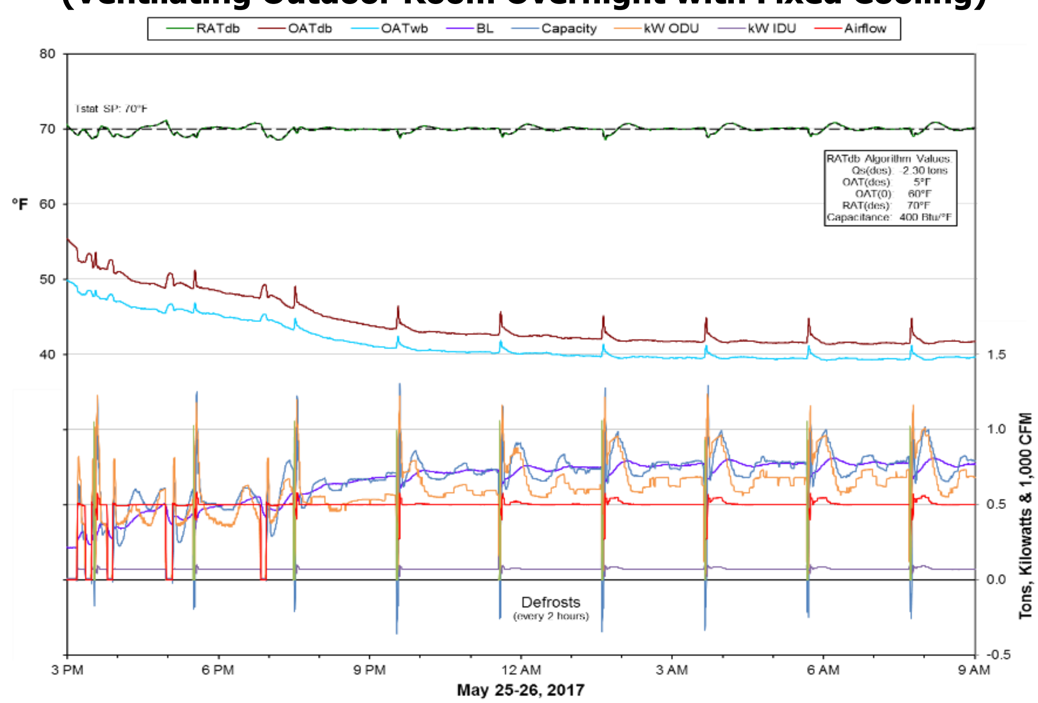
Source: PG&E

Figure E-5: Dynamic Heating Test #2-1C (with Fixed Outdoor Temperature, R-32) (Test unit switched to backup gas heating)



Source: PG&E

Figure E-6: Dynamic Heating Test #2-3 (with Variable Outdoor Temperature, R-32) (Ventilating Outdoor Room Overnight with Fixed Cooling)



Source: PG&E

APPENDIX F:
Phase 2 WCEC Lab Settings

Table F-1: Summary of Zoning Experiments: Tests Conducted for Different Duct-Zone Temperatures with the Compressor Speed and Airflow Rates Kept in Sync. Indoor = 75°F DB/ 62.5°F WB

Duct zone temperature (DB, °F)	Capacity and air flow rates (% of nominal values)	Measured Evaporator Airflow rate (CFM)	Power (kW)		Open Zones	Equipment Capacity (Btu/h)	Delivery Effectiveness	Equipment COP	Overall System COP
			Indoor Unit	Outdoor Unit					
85	40	352	0.090	0.514	Zone 1	10600	0.91	5.17	4.71
		347	0.178	0.510	Zone 2	10700	0.93	4.56	4.24
		340	0.168	0.511	Zone 3	10400	0.92	4.52	4.16
		345	0.125	0.510	Zone 4	10700	0.89	4.94	4.38
		354	0.060	0.504	Zones 1, 3	10400	0.86	5.38	4.64
		354	0.065	0.504	Zones 2, 3	10500	0.89	5.40	4.84
	60	450	0.138	0.645	Zones 2,3	13500	0.93	5.04	4.68
		474	0.091	0.644	Zones 1,4	13200	0.87	5.27	4.59
		478	0.106	0.644	Zones 1,3	13400	0.91	5.25	4.78
		487	0.082	0.641	Zones 2,3,4	13500	0.87	5.48	4.75
	80	648	0.142	1.069	Zones 1,3,4	20800	0.89	5.01	4.46
95	40	348	0.090	0.60	Zone 1	9160	0.85	3.87	3.30
		323	0.178	0.60	Zone 2	8730	0.88	3.28	2.90
		326	0.172	0.60	Zone 3	8710	0.88	3.30	2.90
		339	0.124	0.60	Zone 4	9310	0.83	3.76	3.12
		350	0.052	0.603	Zones 1, 3	9140	0.79	4.09	3.25
		345	0.063	0.600	Zones 2, 3	9080	0.83	4.01	3.35
	60	484	0.139	0.747	Zones 2,3	11300	0.90	3.73	3.34
		485	0.091	0.751	Zones 1,4	11900	0.86	4.12	3.52
		478	0.105	0.748	Zones 1,3	11400	0.87	3.92	3.39
		484	0.081	0.751	Zones 2,3,4	12200	0.81	4.29	3.48
	80	575	0.146	1.225	Zones 1,3,4	15900	0.84	3.40	2.85
105	40	354	0.090	0.703	Zone 1 only	7740	0.77	2.86	2.21
		340	0.181	0.699	Zone 2 only	7770	0.83	2.58	2.16
		340	0.063	0.697	Zones 2, 3	7600	0.78	2.93	2.29
	60	484	0.143	0.871	Zones 2,3	10800	0.88	3.12	2.74
		486	0.085	0.871	Zones 2,3,4	10500	0.79	3.20	2.53
115	40	354	0.090	0.801	Zone 1 only	6830	0.69	2.24	1.54
		338	0.178	0.816	Zone 2 only	6710	0.77	1.98	1.53
		345	0.168	0.802	Zone 3 only	6850	0.73	2.07	1.52
		348	0.125	0.805	Zone 4 only	6950	0.66	2.19	1.45
		357	0.057	0.803	Zones 1, 3	6790	0.59	2.31	1.37
		358	0.061	0.805	Zones 2, 3	6530	0.67	2.21	1.49
	60	481	0.138	0.996	Zones 2,3	9600	0.76	2.48	1.88
		482	0.092	1.003	Zones 1,4	9450	0.64	2.53	1.62
		482	0.102	0.999	Zones 1,3	9500	0.68	2.53	1.73
		486	0.087	1.01	Zones 2,3,4	9340	0.64	2.49	1.61
	80	645	0.142	1.616	Zones 1,3,4	14500	0.70	2.41	1.69

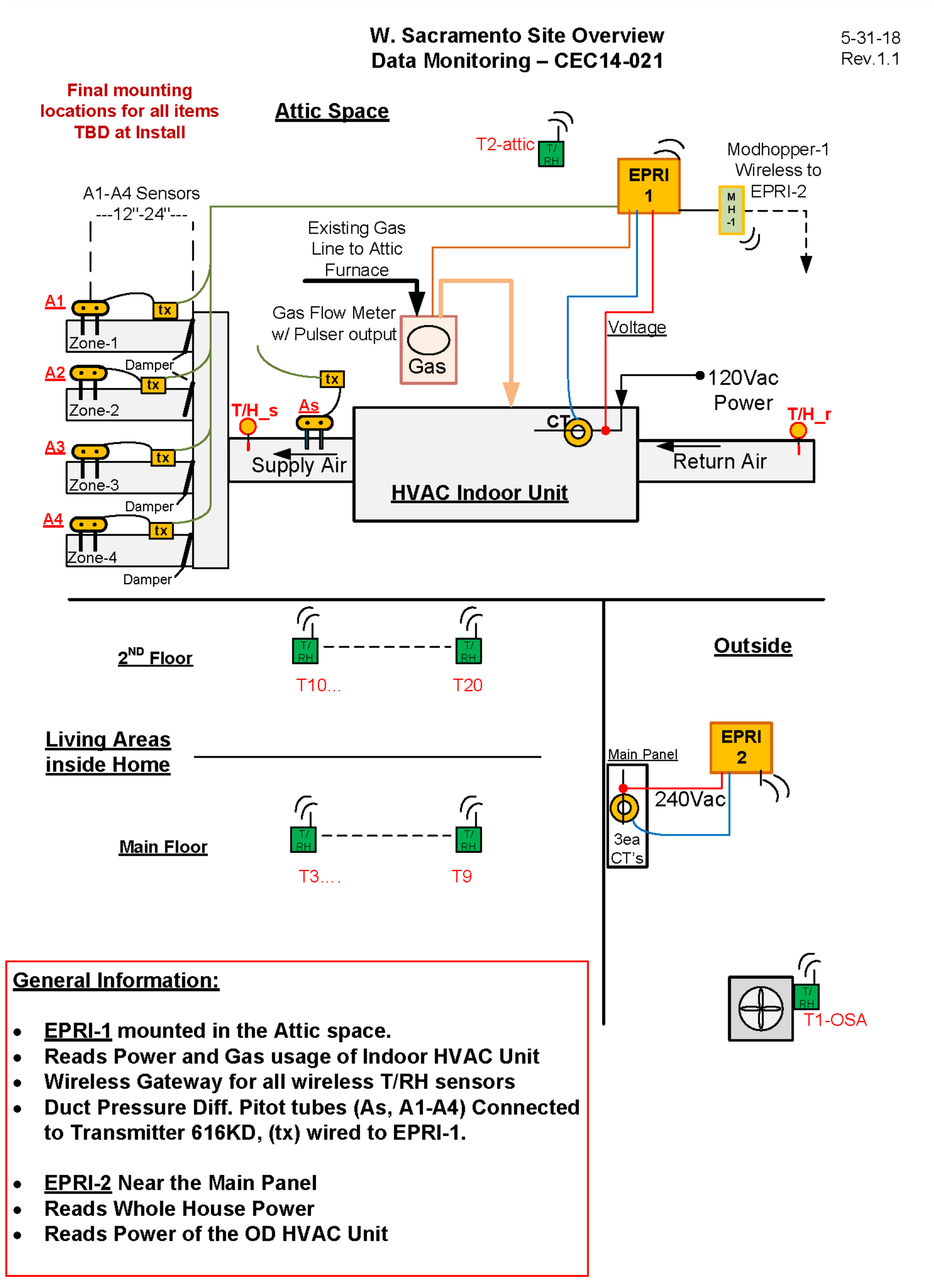
Source: WCEC

Table F-2: Summary of Zoning Experiments: Test Conducted for Different Duct-Zone Temperatures with the Compressor Speed and Airflow Rates Not in Sync. Indoor = 75°F DB/ 62.5°F WB

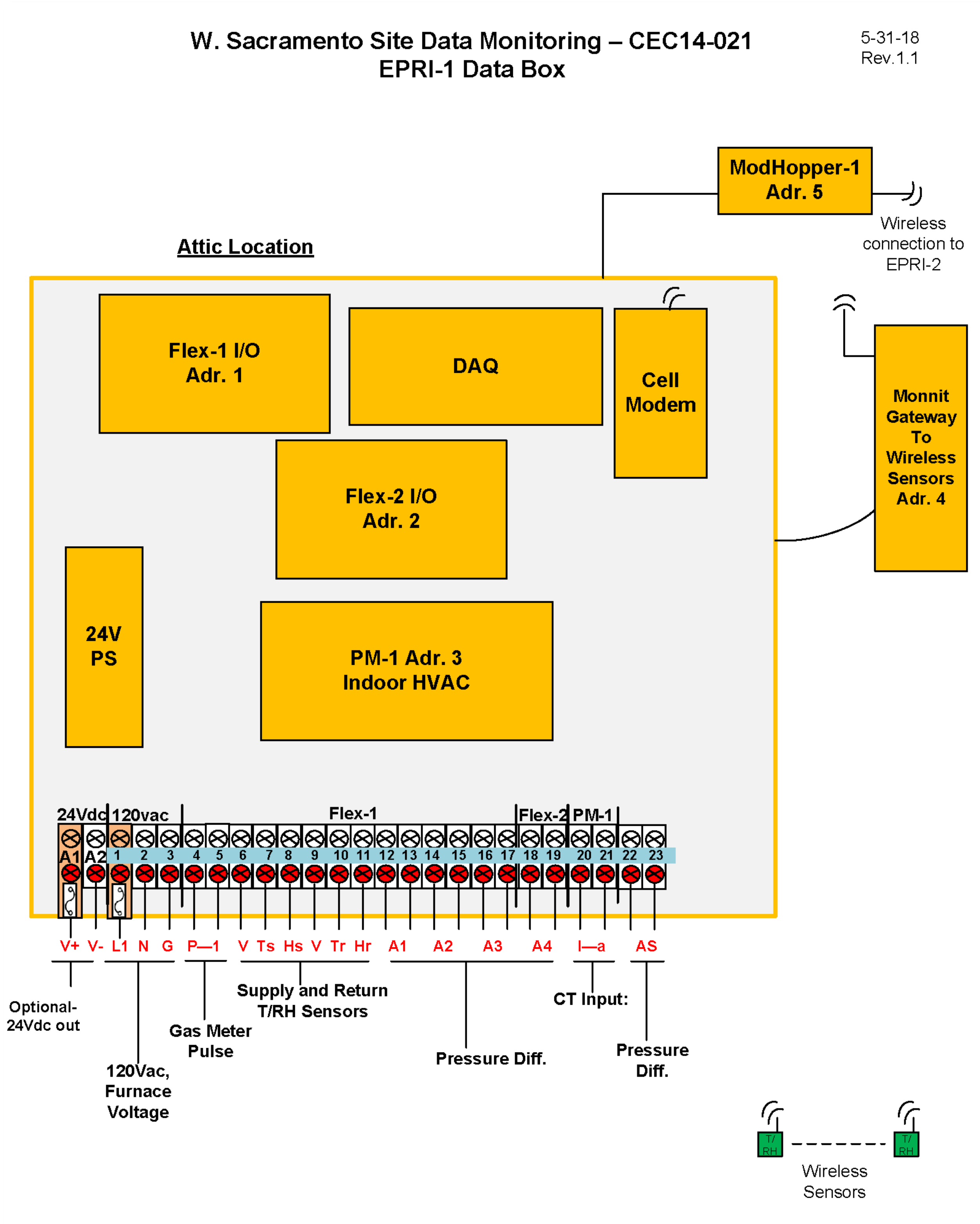
Duct zone temperature (DB, °F)	Capacity (% of nominal value)	Targeted Air Flow Rate (% of nominal value)	Power (kW)		Open Zones	Equipment Capacity (Btu/h)	Delivery Effectiveness	Equipment COP	Overall System COP
			Indoor Unit	Outdoor Unit					
85	40	60	0.098	0.505	Zones 1,3	11900	0.87	5.78	4.64
	40	60	0.138	0.500	Zones 2,3	11900	0.94	5.45	5.11
	60	80	0.156	0.639	Zones 2,3,4	14200	0.87	5.22	4.54
	80	100	0.242	1.079	Zones 1,3,4	20800	0.89	4.62	4.12
95	40	60	0.101	0.594	Zones 1,3	10000	0.83	4.23	3.52
	40	60	0.140	0.596	Zones 2,3	9980	0.85	3.97	3.37
	60	80	0.166	0.753	Zones 2,3,4	12400	0.81	3.95	3.19
	80	100	0.252	1.244	Zones 1,3,4	16100	0.83	3.16	2.63
115	40	60	0.102	0.813	Zones 1,3	8190	0.63	2.62	1.65
	40	60	0.139	0.802	Zones 2,3	7930	0.72	2.47	1.79
	60	80	0.159	1.009	Zones 2,3,4	9370	0.67	2.35	1.57
	80	100	0.251	1.623	Zones 1,3,4	14800	0.70	2.32	1.63

Source: WCEC

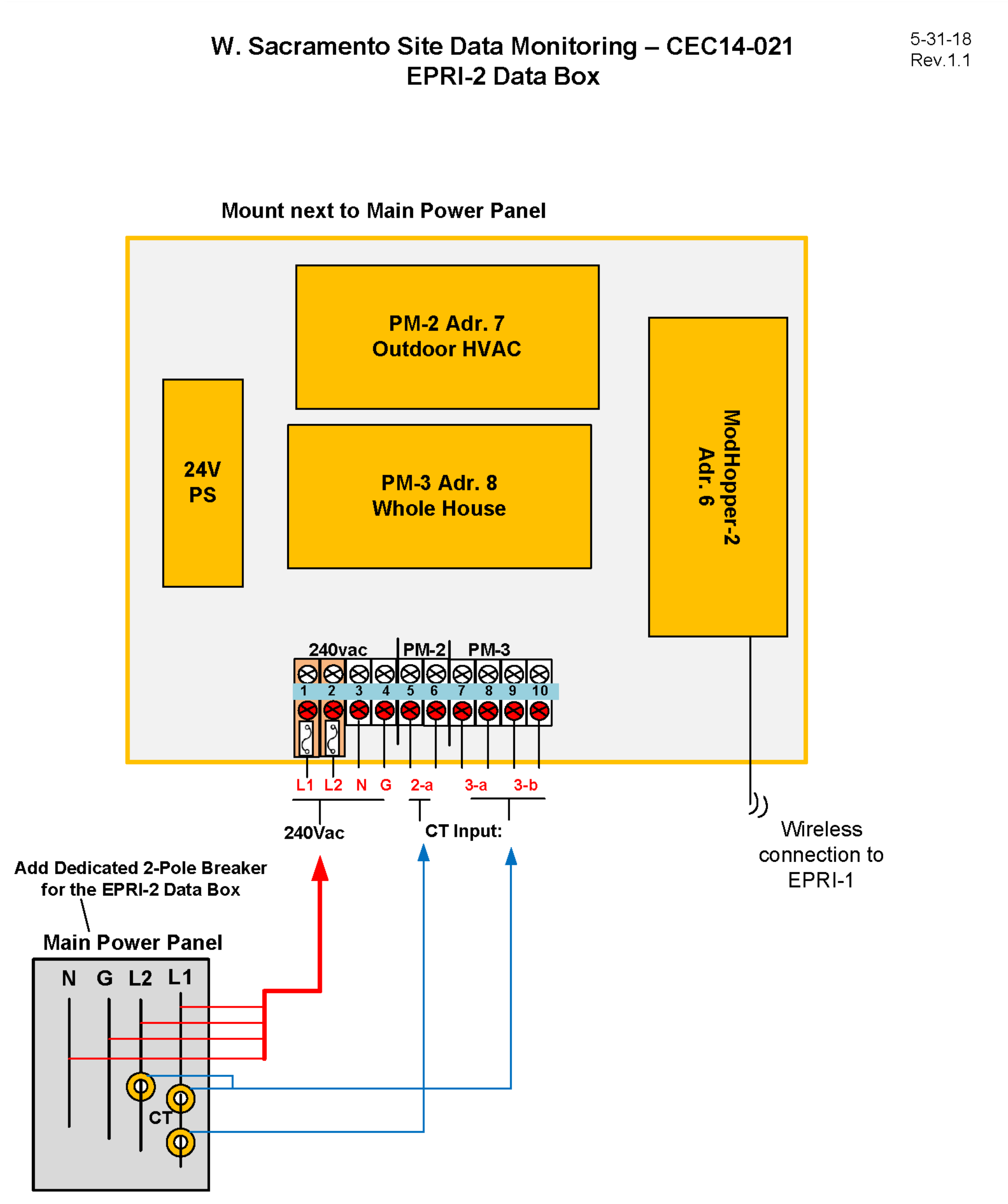
APPENDIX G:
Phase 3 Documentation



Source: EPRI


W. Sacramento Site Data Monitoring – CEC14-021
EPRI-1 Data Box
5-31-18
Rev.1.1
ModHopper-1
Adr. 5
Wireless connection to EPRI-2
Attic Location
Flex-1 I/O
Adr. 1
DAQ
Cell Modem
Monnit Gateway To Wireless Sensors Adr. 4
Flex-2 I/O
Adr. 2
24V PS
PM-1 Adr. 3
Indoor HVAC
24Vdc
120vac
Flex-1
Flex-2
PM-1
A1 A2 1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23
V+ V- L1 N G P—1 V Ts Hs V Tr Hr A1 A2 A3 A4 I—a AS
Optional-24Vdc out
120Vac, Furnace Voltage
Gas Meter Pulse
Supply and Return T/RH Sensors
Pressure Diff.
CT Input:
Pressure Diff.
T/RH
T/RH
Wireless Sensors

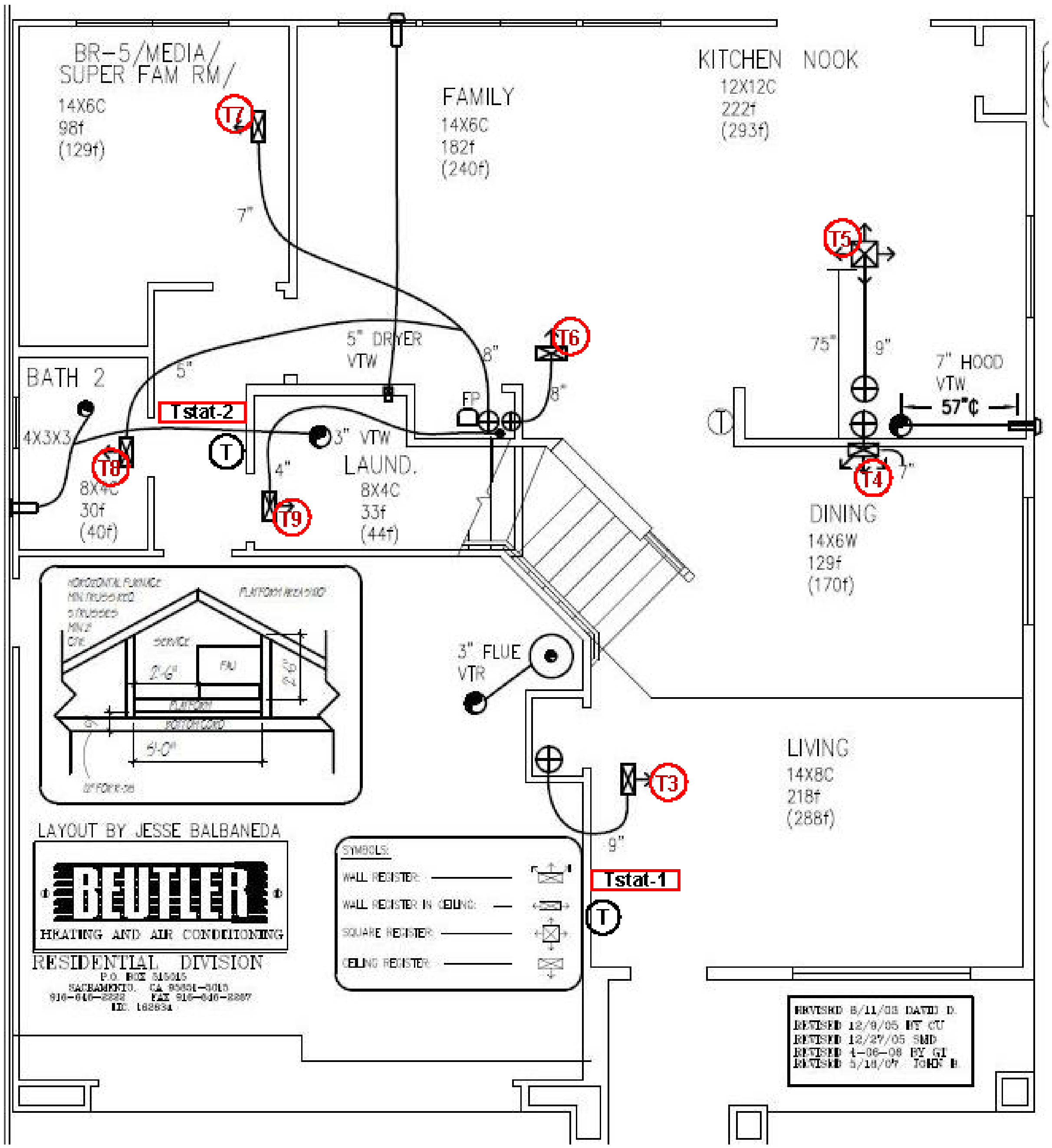
Source: EPRI


W. Sacramento Site Data Monitoring – CEC14-021
EPRI-2 Data Box
5-31-18
Rev.1.1
Mount next to Main Power Panel
PM-2 Adr. 7
Outdoor HVAC
24V
PS
PM-3 Adr. 8
Whole House
ModHopper-2
Adr. 6
240vac
PM-2
PM-3
1 2 3 4 5 6 7 8 9 10
L1 L2 N G
2-a
3-a
3-b
240Vac
CT Input:
Wireless
connection to
EPRI-1
Add Dedicated 2-Pole Breaker
for the EPRI-2 Data Box
Main Power Panel
N G L2 L1
CT

Source: EPRI

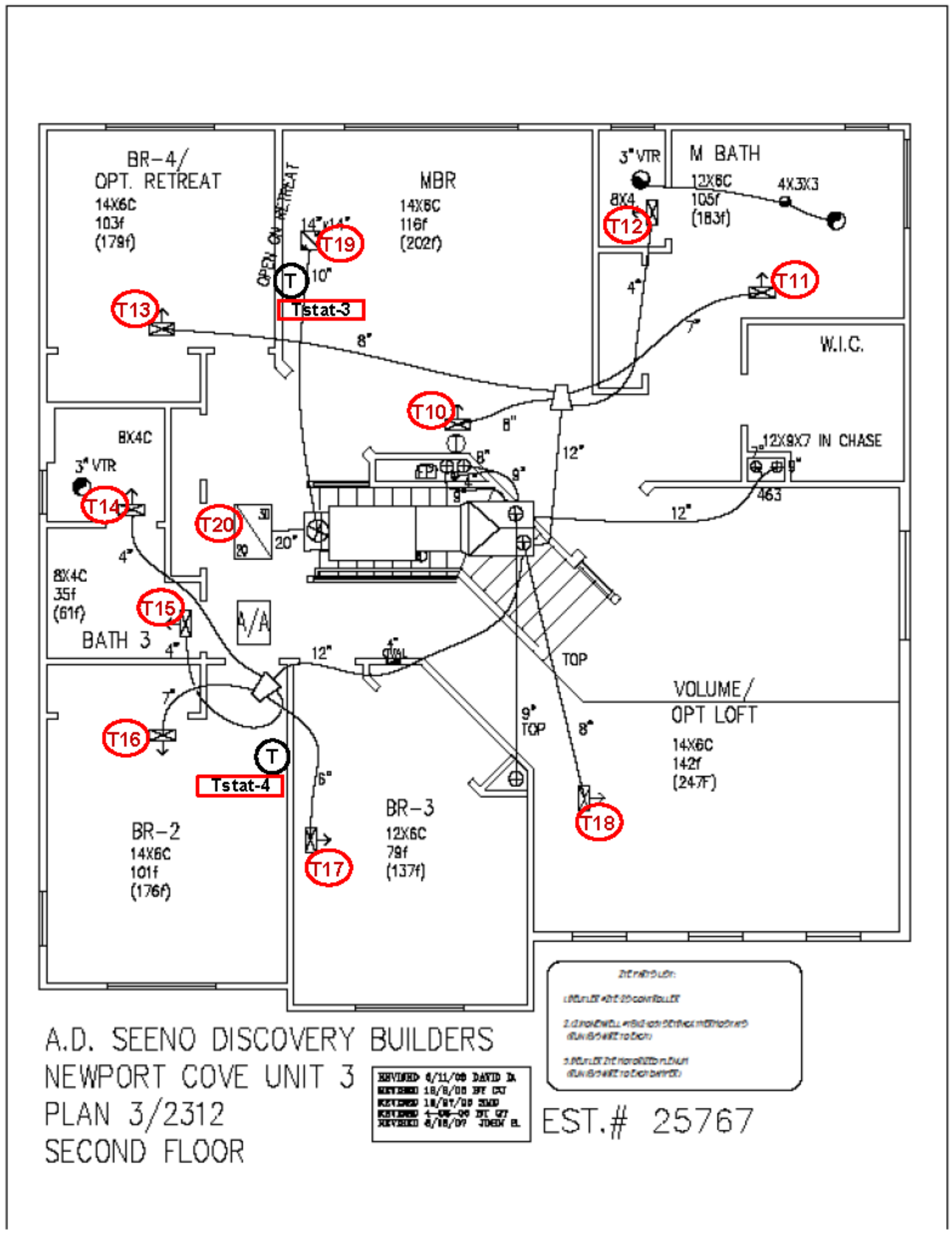
W. Sacramento House – Main Floor

5-31-18
Rev.1.1



Source: EPRI

W. Sacramento House 2nd Floor



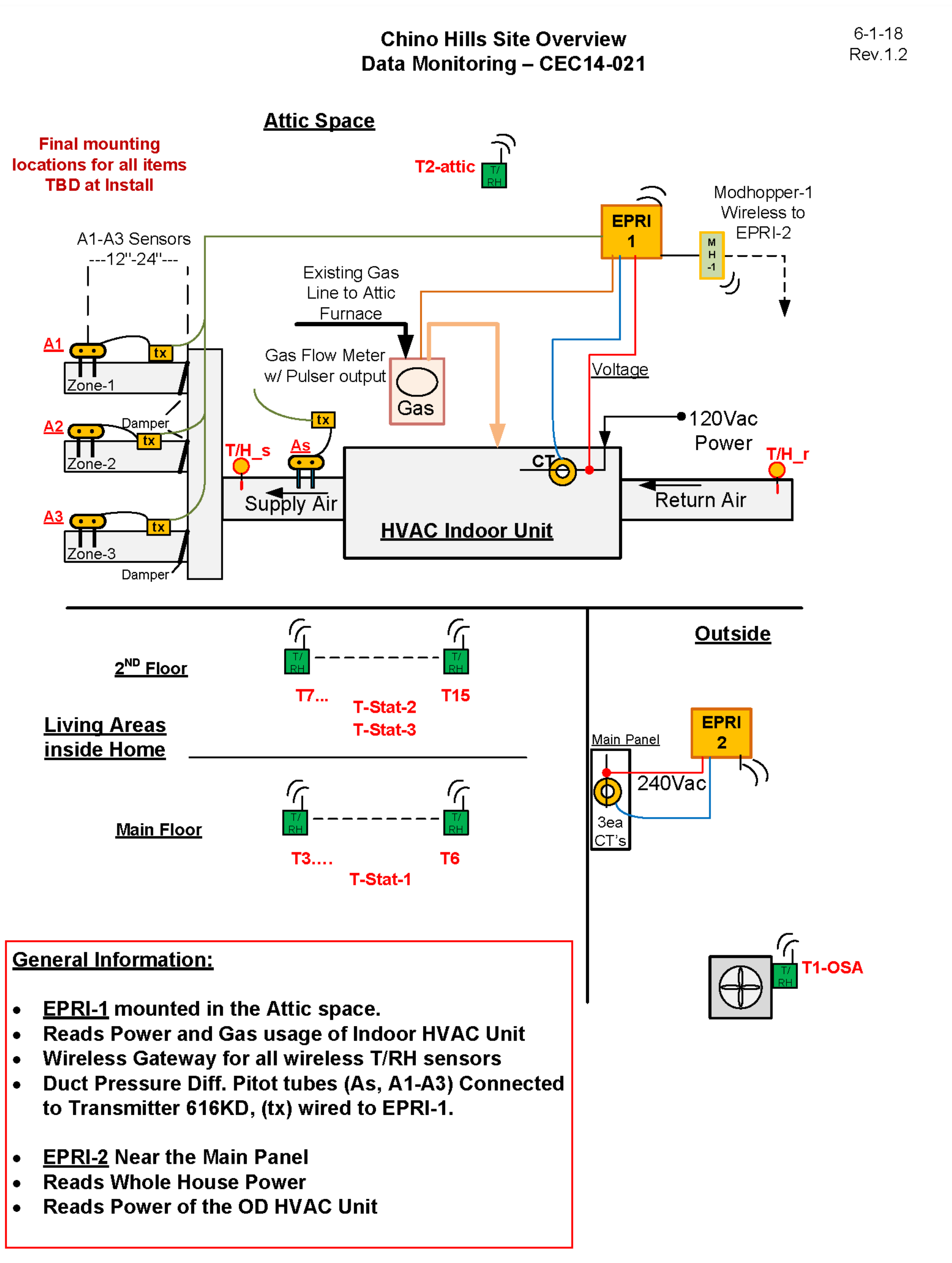
Source: EPRI

W. Sacramento CEC-Site Point List

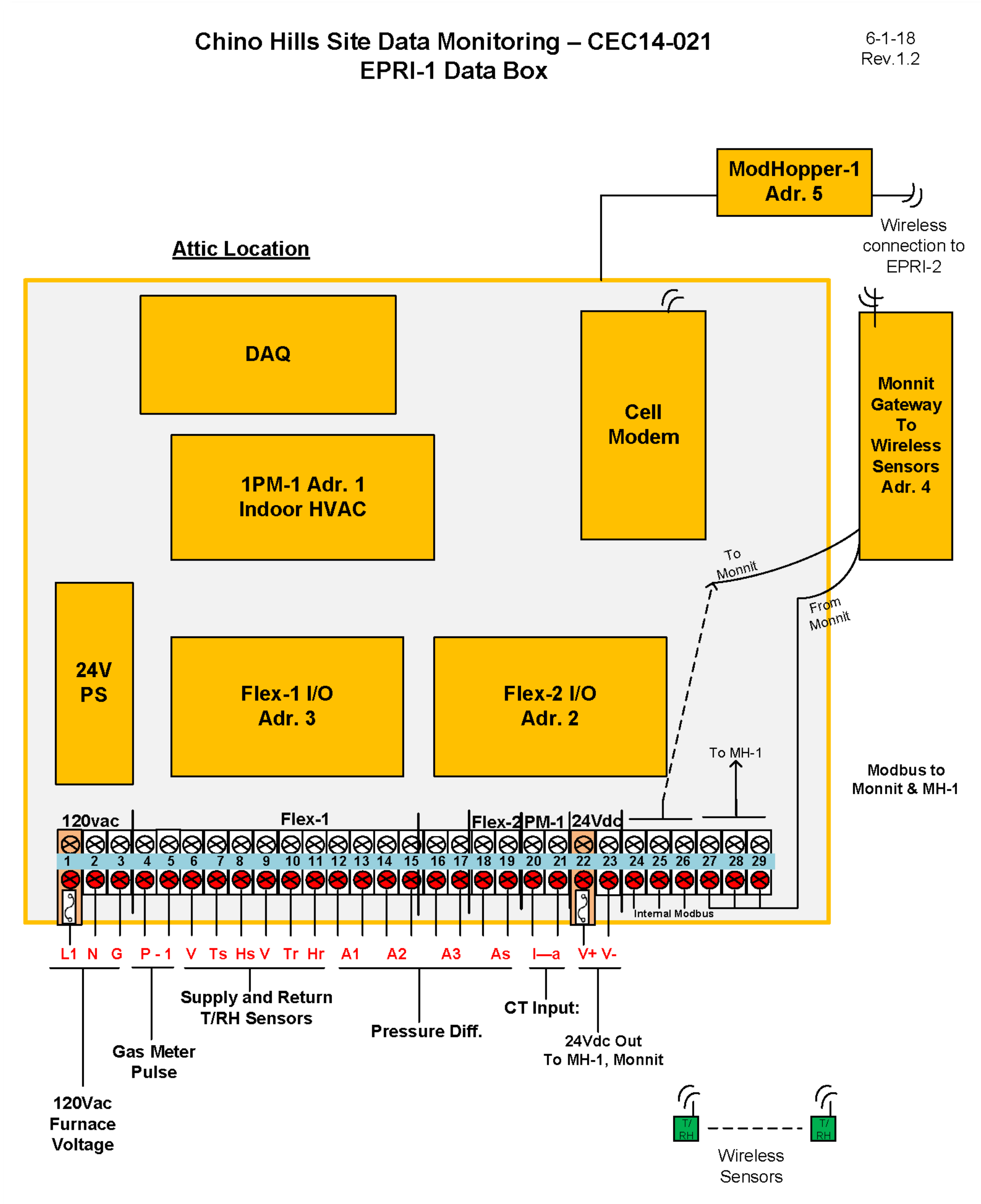
5-31-18
Rev.1.1

Item	Address-mb#	Description	Notes
			5/31/2018
Point List- Sacramento- CEC Residential HVAC Project - Sara Beaini			
Good Start Date>>			MAC: 001EC60-54A37
Item	Address-mb#	Description	Notes
			Cell IP: 166.247.148.036 / 704 996-3362
Data Box EPRI-1 Attic			
Acquisuite A8810-0 / Cell Modem		Interval Data = 1min	
Flex-1 Attic	Adr.1		
Gas Flow to Furnace	CH 1	Pulse output gas flow meter	Model# BK250 w/ Pulse output 1p/Cf
Ts = Supply Air Duct Temp	CH 2	Dwyer In-Duct; wired	
Hs = Supply Air Duct RH	CH 3	"	
Tr = Return Air Duct Temp	CH 4	"	
Hr = Return Air Duct RH	CH 5	"	
A1 = Zone-1 Supply Air Duct Press. Diff.	CH 6	Dwyer PAFS/616KD-B	Pitot Tube and 1% Transducer 4-20mA output
A2 = Zone-2 Supply Air Duct Press. Diff.	CH 7	"	"
A3 = Zone-3 Supply Air Duct Press. Diff.	CH 8	"	"
Flex-2 Attic	Adr. 2		
As = Main Supply Duct Pressure Diff.	Ch 1	Dwyer PAFS/616KD-B	Pitot Tube and 1% Transducer 4-20mA output
A4 = Zone-4 Supply Air Duct Press. Diff.	Ch 2	NA	
NA	Ch3	NA	
PM-1; Indoor HVAC Unit Power	Adr. 3	W2-M1-mA	120Vac; 1 CT MRS-075
Monnit Gateway - Alta	Adr. 4		
T1 - OSA T/RH (OD HVAC Unit) (T/RH)	Slot 0	386690, Wireless Sensor T/RH	Industrial Monnit Sensor - OSA
T2 - Attic Temp/RH - (T/RH)	Slot 1	372674	
T3 - Air Vent - Entry (T)	Slot 2	386833, First Floor	
T4 - Air Vent - Dining Area (T)	Slot 3	386863	
T5 - Air Vent - Kitchen (T)	Slot 4	386869	
T6 Air Vent - Family Room (T)	Slot 5	386870	
T7 Air Vent - Spare Room (T)	Slot 6	372706	
T8 Air Vents - Bath (T)	Slot 7	386818	
T9 Air Vent - Laundry (T)	Slot 8	386859	
T10 Air Vent - MBR (T)	Slot 9	386704, Secound Floor	
T11 Air Vent - Master Bath (T)	Slot 10	386775	
T12 Air Vent - Master Bath Tolet (T)	Slot 11	386826	
T13 Air Vent - Office (T)	Slot 12	386867	
T14 Air Vent - Hall Tolet (T)	Slot 13	386858	
T15 Air Vent - Hall Bath (T)	Slot 14	386718	
T16 Air Vent - BR-2 (T)	Slot 15	386836	
T17 Air Vent - BR-3 (T)	Slot 16	386732	
T18 Air Vent - Loft area (T)	Slot 17	386729	
T19 Return Vent - MBR (T/RH)	Slot 18	372680	
T20 Return Vent - Hall Upstairs (T/RH)	Slot 19	372687	
TStat-1 - Entry Stat (T/RH)	Slot 20	372718, 1st Floor	
TStat-2 - Hall Main Floor (T/RH)	Slot 21	372710, 1st Floor	
TStat-3 - Master BR (T/RH)	Slot 22	372696, 2nd Floor	
TStat-4 - Upstairs BR (T/RH)	Slot 23	372667, 2nd Floor	
ModHopper-1 DAQ	Adr. 5	Wireless data collection from EPRI-2	MH-1 to MH-2
Data Box EPRI-2		Outside next to Power Panel	
ModHopper-2 HVAC & House Power	Adr. 6	Communication between Box-1 and Box-2	
PM-2; Outdoor HVAC Unit	Adr. 7	W2-M1-mA	240Vac; 1 CT MRS-075
PM-3; Whole House Power	Adr. 8	"	240Vac; 2 CT's House Mains

Source: EPRI

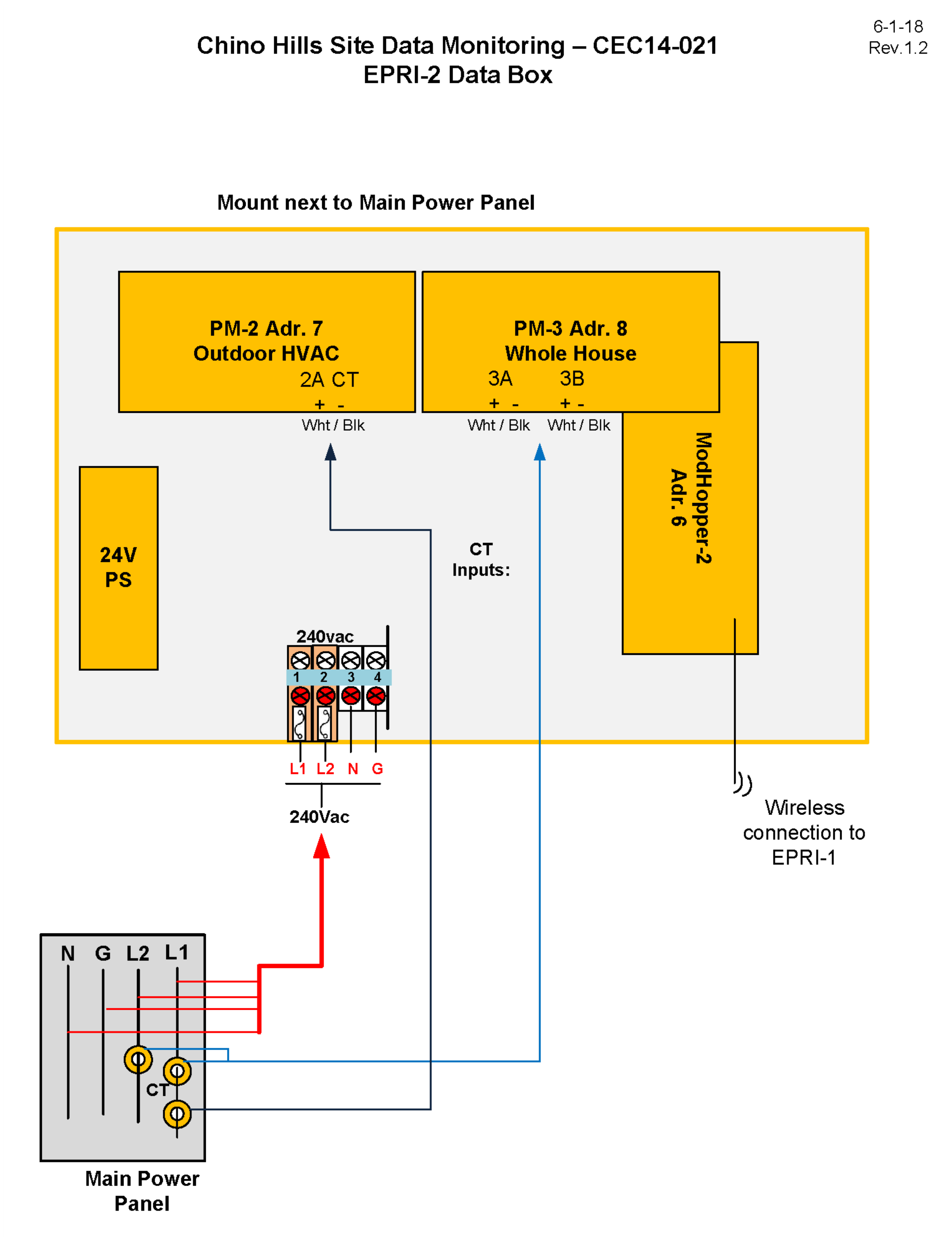

Chino Hills Site Overview
Data Monitoring – CEC14-021
6-1-18
Rev.1.2
Attic Space
Final mounting locations for all items TBD at Install
T2-attic
T/RH
EPRI 1
Modhopper-1 Wireless to EPRI-2
MH-1
A1-A3 Sensors
---12"-24"---
Existing Gas Line to Attic Furnace
Gas Flow Meter w/ Pulser output
Gas
Voltage
120Vac Power
A1
tx
Zone-1
A2
Damper
Zone-2
A3
Zone-3
Damper
T/H_s
As
Supply Air
CT
HVAC Indoor Unit
T/H_r
Return Air
2ND Floor
T7…
T-Stat-2
T-Stat-3
T15
Living Areas inside Home
Main Floor
T3….
T-Stat-1
T6
Outside
Main Panel
EPRI 2
240Vac
3ea CT's
T1-OSA
General Information:
EPRI-1 mounted in the Attic space.
Reads Power and Gas usage of Indoor HVAC Unit
Wireless Gateway for all wireless T/RH sensors
Duct Pressure Diff. Pitot tubes (As, A1-A3) Connected to Transmitter 616KD, (tx) wired to EPRI-1.
EPRI-2 Near the Main Panel
Reads Whole House Power
Reads Power of the OD HVAC Unit

Source: EPRI

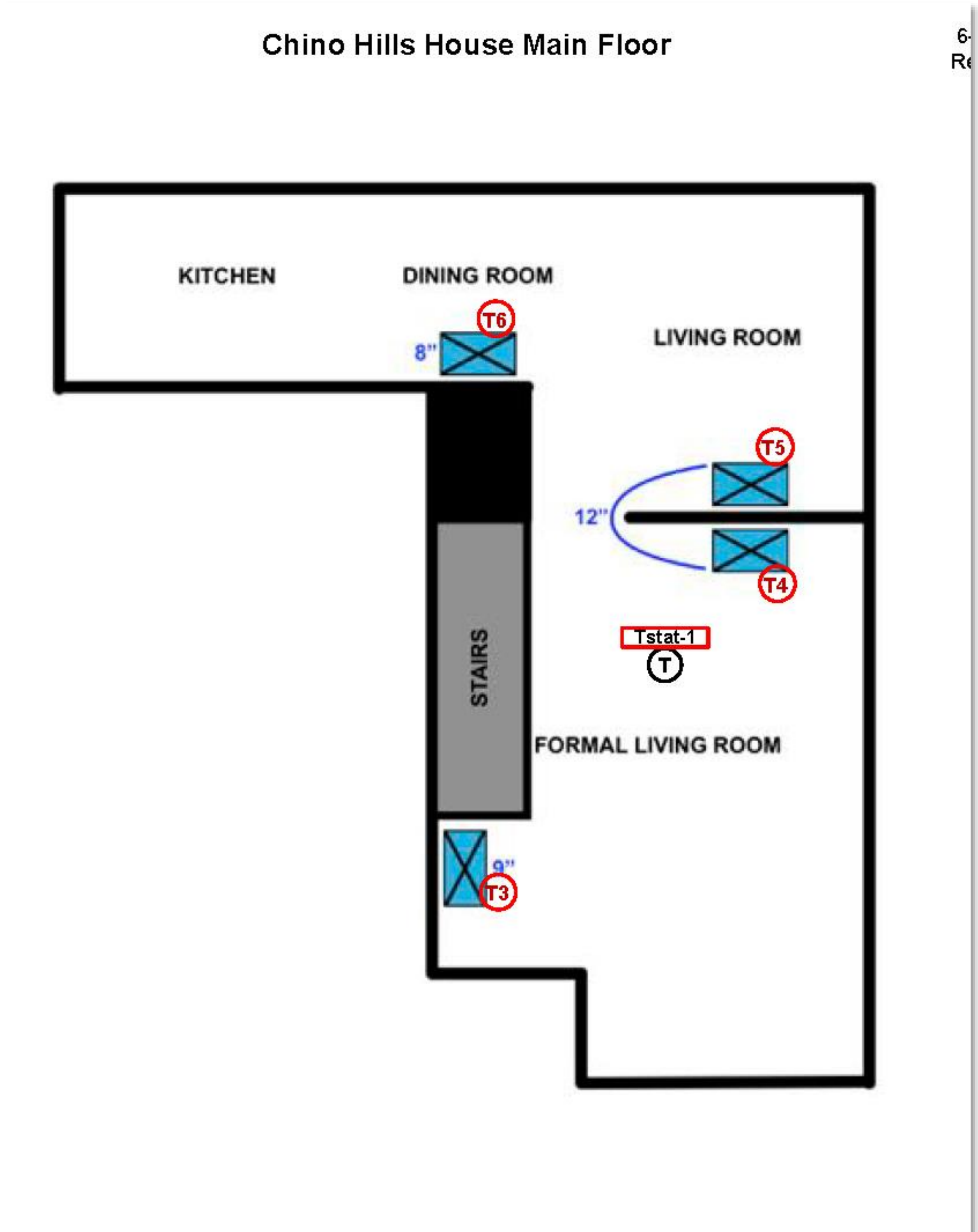


Chino Hills Site Data Monitoring – CEC14-021
EPRI-1 Data Box
6-1-18
Rev.1.2
ModHopper-1
Adr. 5
Wireless connection to EPRI-2
Attic Location
DAQ
Cell Modem
Monnit Gateway To Wireless Sensors Adr. 4
1PM-1 Adr. 1
Indoor HVAC
To Monnit
From Monnit
24V PS
Flex-1 I/O
Adr. 3
Flex-2 I/O
Adr. 2
To MH-1
Modbus to Monnit & MH-1
120vac
Flex-1
Flex-2
PM-1
24Vdc
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17 18 19 20 21 22 23 24 25 26 27 28 29
Internal Modbus
L1 N G P - 1 V Ts Hs V Tr Hr A1 A2 A3 As I—a V+ V-
Supply and Return T/RH Sensors
Pressure Diff.
CT Input:
24Vdc Out
To MH-1, Monnit
Gas Meter Pulse
120Vac Furnace Voltage
T/RH
T/RH
Wireless Sensors

Source: EPRI

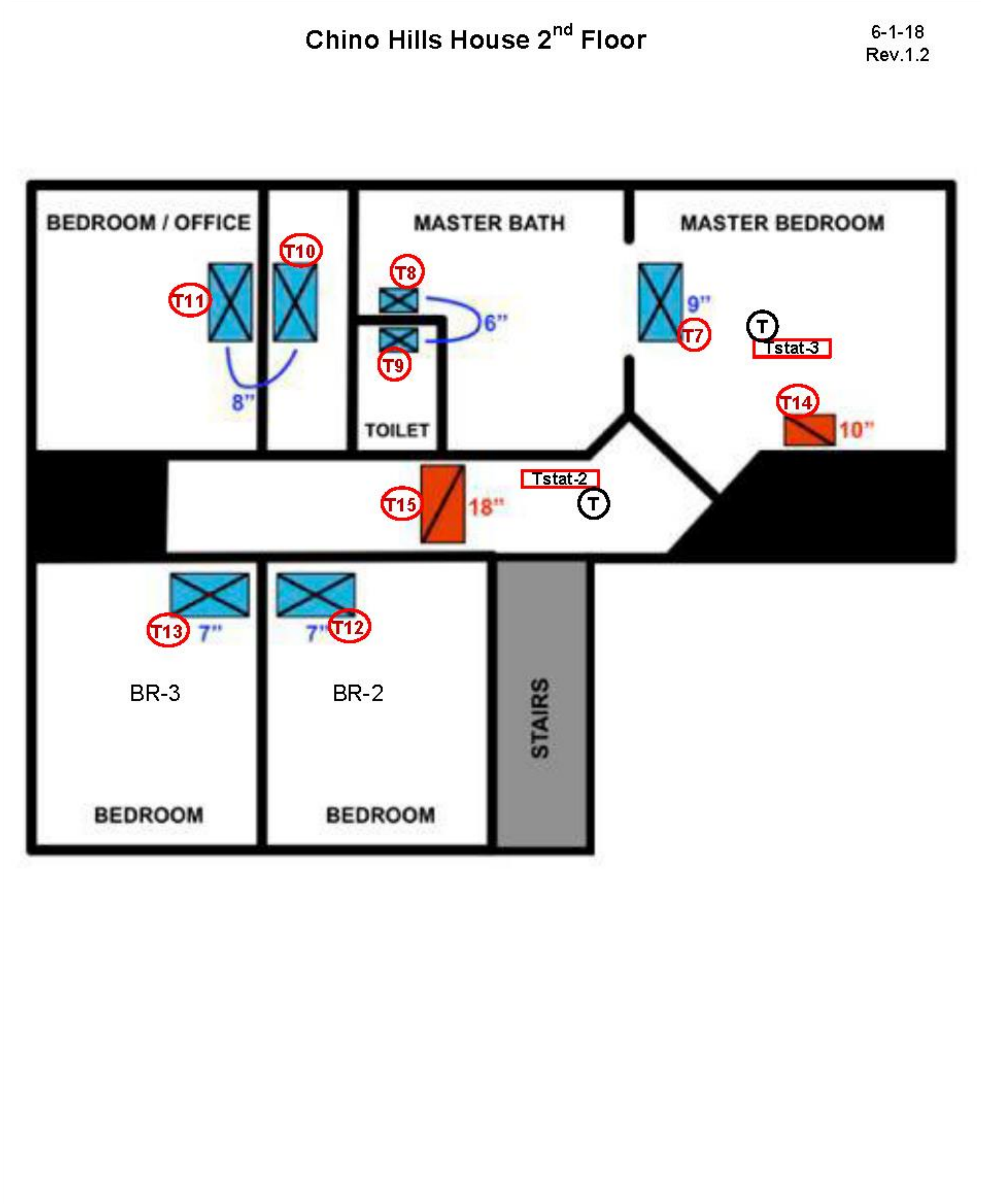

6-1-18
Rev.1.2
Chino Hills Site Data Monitoring – CEC14-021
EPRI-2 Data Box
Mount next to Main Power Panel
PM-2 Adr. 7
Outdoor HVAC
2A CT
+ -
Wht / Blk
PM-3 Adr. 8
Whole House
3A
3B
+ -
+ -
Wht / Blk
Wht / Blk
ModHopper-2
Adr. 6
24V
PS
CT
Inputs:
240vac
1
2
3
4
L1
L2
N
G
240Vac
Wireless
connection to
EPRI-1
N
G
L2
L1
CT
Main Power
Panel

Source: EPRI



Chino Hills House Main Floor
6-
Re
KITCHEN
DINING ROOM
T6
8"
LIVING ROOM
T5
12"
T4
Tstat-1
T
STAIRS
FORMAL LIVING ROOM
9"
T3

Source: EPRI


Chino Hills House 2nd Floor
6-1-18
Rev.1.2
BEDROOM / OFFICE
MASTER BATH
MASTER BEDROOM
T10
T11
T8
6"
T9
9"
T7
Tstat-3
8"
TOILET
T14
10"
Tstat-2
T15
18"
T13
7"
7"
T12
BR-3
BR-2
STAIRS
BEDROOM
BEDROOM

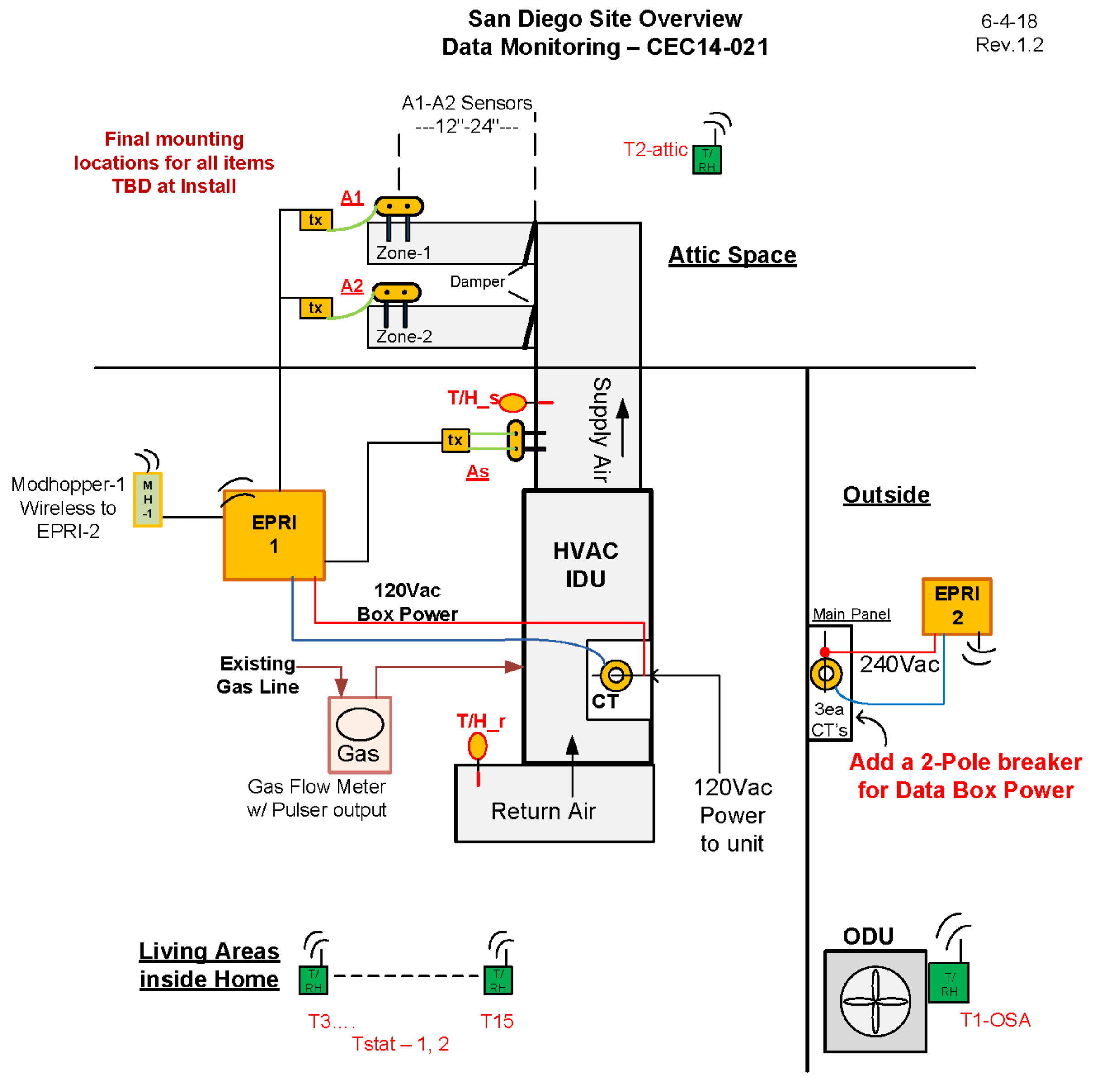
Source: EPRI

Point List

6-1-18
Rev.1.2

			6/1/2018
Point List- Chino Hills- CEC Residential HVAC Project - Sara Beaini			
Good Start Date>>		2826 Rolling Meadow Dr.; Chino Hills CA, 91709	MAC: 001EC60-54A36
Item	**Address-mb#**	**Description**	**Notes**
			Cell IP: 166.148.178.147; 704 589-3475
Data Box EPRI-1 Attic			
Acquisuite A8810-0 / Cell Modem		**Interval Data = 1min**	
PM-1; Indoor HVAC Unit Power	**Adr. 1**	**W2-M1-mA**	**120Vac; 1 CT MRS-075**
Flex-1 Attic	**Adr.2**		
Gas Flow to Furnace	**CH 1**	**Pulse output gas flow meter**	**Model# BK250 w/ Pulse output 1p/Cf**
Ts = Supply Air Duct Temp	**CH 2**	**Dwyer In-Duct; wired**	
Hs = Supply Air Duct RH	**CH 3**	"	
Tr = Return Air Duct Temp	**CH 4**	"	
Hr = Return Air Duct RH	**CH 5**	"	
A1 = Zone-1 Supply Air Duct Press. Diff.	**CH 6**	**Dwyer PAFS/616KD-LR; 0.5inW.C.**	**Pitot Tube and .25% Transducer 4-20mA output**
A2 = Zone-2 Supply Air Duct Press. Diff.	**CH 7**	"	"
A3 = Zone-3 Supply Air Duct Press. Diff.	**CH 8**	"	"
Flex-2 Attic	**Adr. 3**		
As = Main Supply Duct Pressure Diff.	**Ch 1**	**Dwyer PAFS/616KD-B; 1inW.C.**	**Pitot Tube and 1% Transducer 4-20mA output**
(Box Temp option)	**Ch 2**	**NA**	
NA	**Ch3**	**NA**	
Monnit Gateway - Alta	**Adr. 4**		
T1 - OSA T/RH (OD HVAC Unit) (T/RH)	**Slot 0**	**386691, Wireless Sensor T/RH**	**Industrial Monnit Sensor - OSA; 145sec**
T2 - Attic Temp/RH - (T/RH)	**Slot 1**	**372679**	**60sec**
T3 - Air Vent - Entry Area (T)	**Slot 2**	**386714, First Floor**	"
T4 - Air Vent - Formal Living Rm (T)	**Slot 3**	**386857**	"
T5 - Air Vent - Living Rm (T)	**Slot 4**	**386828**	
T6 Air Vent - Dining Rm (T)	**Slot 5**	**386813**	
T7 Air Vent - Master Bed Rm (T)	**Slot 6**	**386868, Secound Floor**	
T8 Air Vents - Master Bath (T)	**Slot 7**	**386735**	
T9 Air Vent - Master Toilet (T)	**Slot 8**	**386773**	
T10 Air Vent - Upstairs Bath Rm (T)	**Slot 9**	**386871**	
T11 Air Vent - Rm /Office (T)	**Slot 10**	**386770**	
T12 Air Vent - Bed Rm (T)	**Slot 11**	**386730**	
T13 Air Vent - Bed Rm (T)	**Slot 12**	**386723**	
T14 Return Vent - MBR (T/RH)	**Slot 13**	**372717**	
T15 Return Vent - Hall Upstairs (T/RH)	**Slot 14**	**372709**	
TStat-1 - Main Floor Thermostat (T/RH)	**Slot 15**	**372716**	
TStat-2 - 2nd Floor Thermostat (T/RH)	**Slot 16**	**372673**	
TStat-3 - 2nd Floor Thermostat (T/RH)	**Slot 17**	**372721**	
MH-1; ModHopper-1 DAQ	**Adr. 5, Ch 5**	**Wireless data collection from EPRI-2**	**MH-1 to MH-2**
Data Box EPRI-2		**Outside next to Power Panel**	
MH-2; ModHopper-2 ODU & Whole House	**Adr. 6, Ch 5**	**Communication between Box-1 and Box-2**	
PM-2; Outdoor HVAC Unit	**Adr. 7**	**W2-M1-mA**	**240Vac; 1 CT MRS-075**
PM-3; Whole House Power	**Adr. 8**	"	**240Vac; 2 CT's House Mains**

Source: EPRI

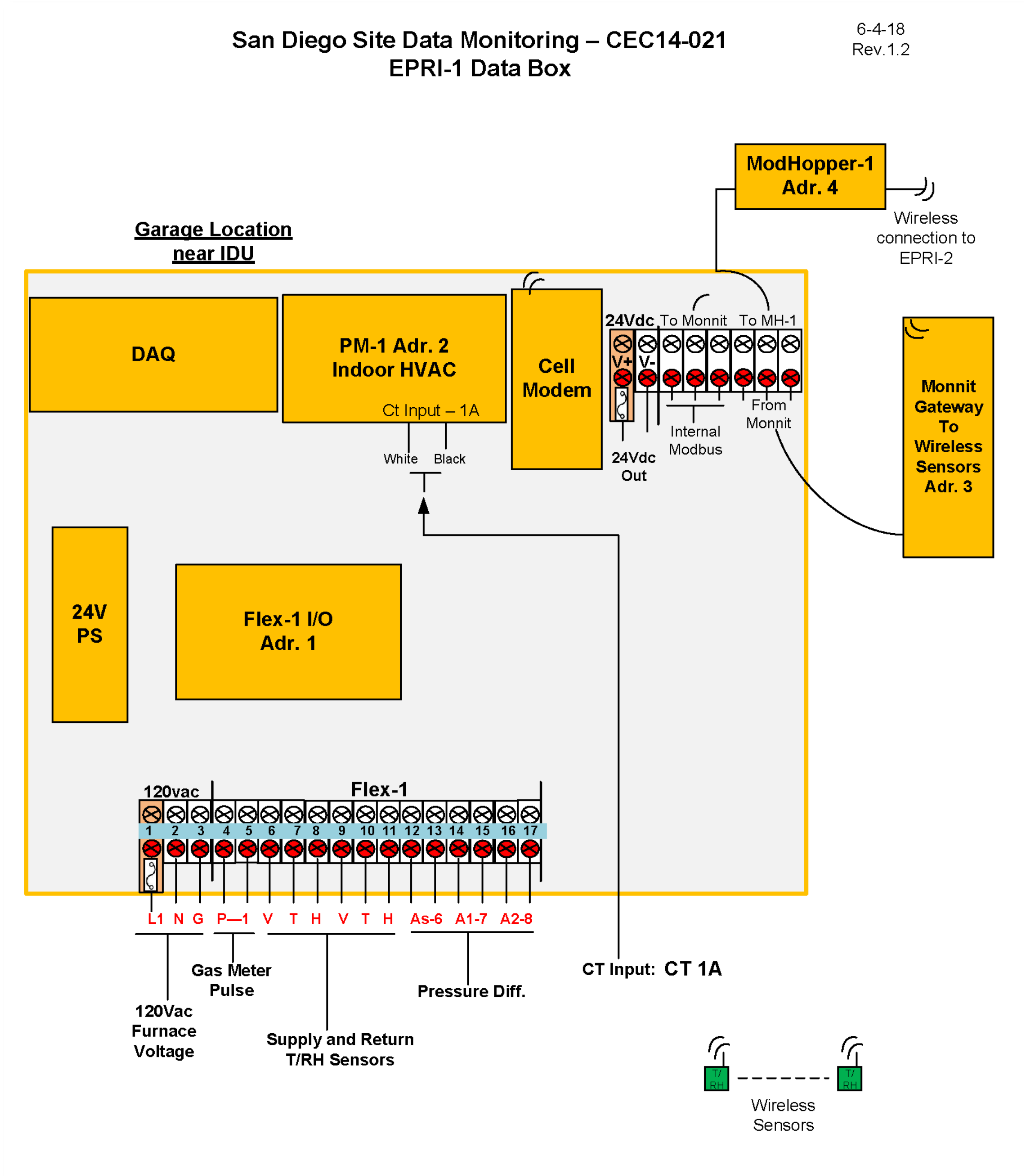


General Information:

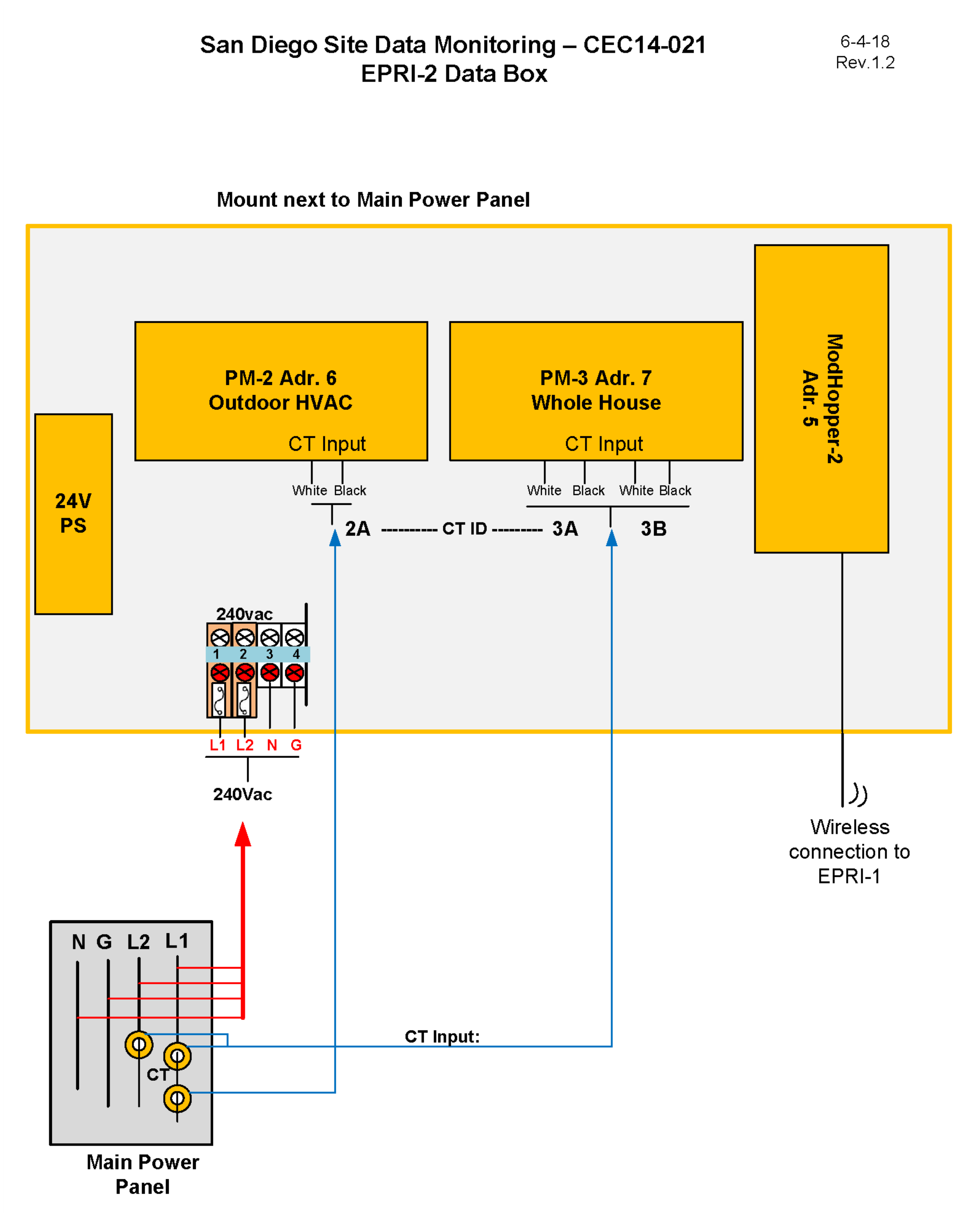
- **EPRI-1 mounted near the IDU in Garage.**
- **Reads Power and Gas usage of Indoor HVAC Unit**
- **Wireless Gateway for all wireless T/RH sensors**
- **Duct Pressure Diff. Pitot tubes (As, A1-A2) Connected to Transmitter 616KD, (tx) wired to EPRI-1.**

- **EPRI-2 Near the Main Panel**
- **Reads Whole House Power**
- **Reads Power of the OD HVAC Unit**

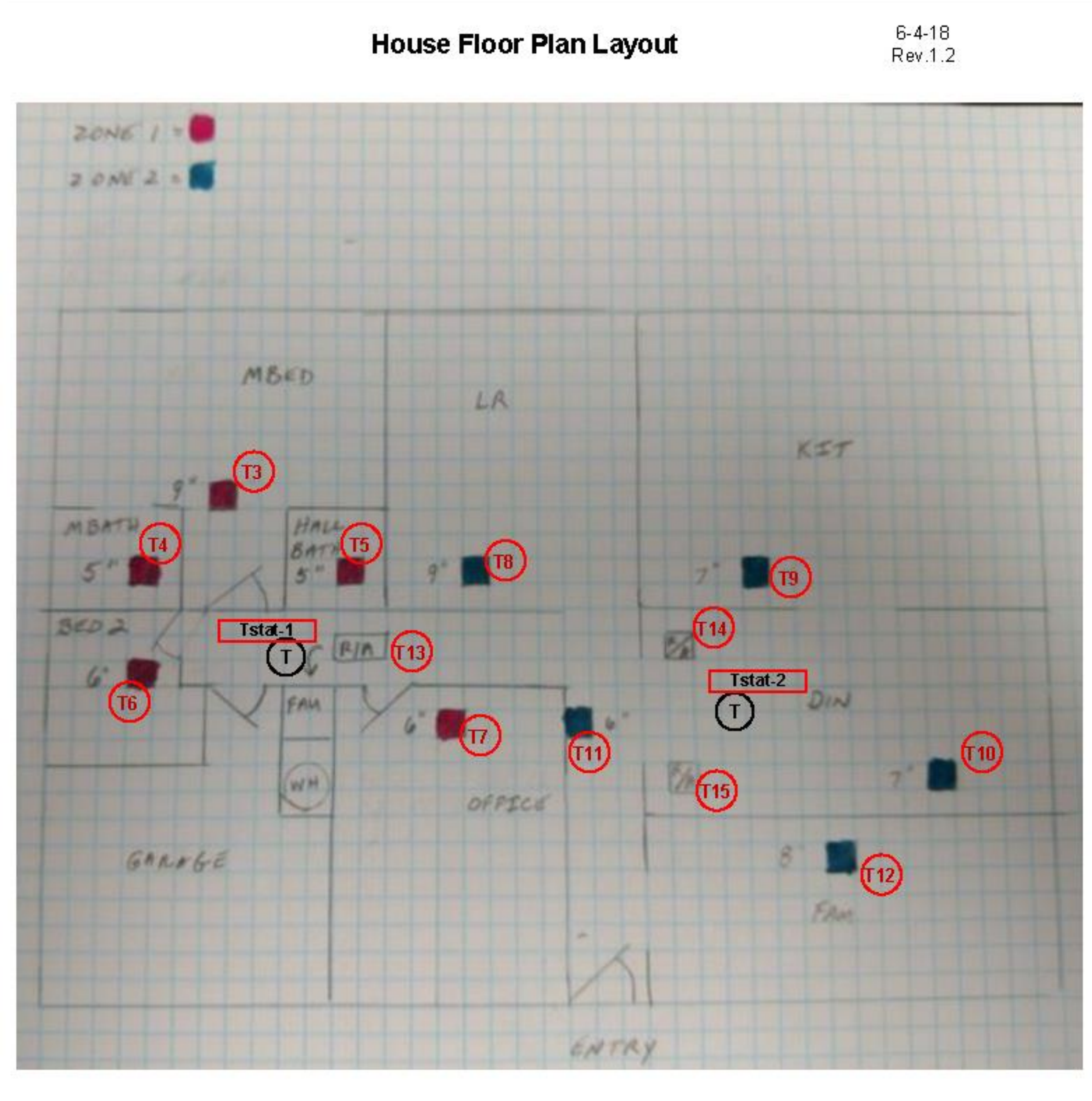
Source: EPRI


San Diego Site Data Monitoring – CEC14-021
EPRI-1 Data Box
6-4-18
Rev.1.2
ModHopper-1
Adr. 4
Wireless connection to EPRI-2
Garage Location near IDU
DAQ
PM-1 Adr. 2
Indoor HVAC
Ct Input – 1A
White
Black
Cell Modem
24Vdc
To Monnit
To MH-1
V+
V-
From Monnit
Internal Modbus
24Vdc Out
Monnit Gateway To Wireless Sensors Adr. 3
24V PS
Flex-1 I/O
Adr. 1
120vac
Flex-1
1 2 3 4 5 6 7 8 9 10 11 12 13 14 15 16 17
L1 N G P—1 V T H V T H As-6 A1-7 A2-8
Gas Meter Pulse
120Vac Furnace Voltage
Supply and Return T/RH Sensors
Pressure Diff.
CT Input: CT 1A
T/RH
T/RH
Wireless Sensors

Source: EPRI


San Diego Site Data Monitoring – CEC14-021
EPRI-2 Data Box
6-4-18
Rev.1.2
Mount next to Main Power Panel
24V PS
PM-2 Adr. 6
Outdoor HVAC
CT Input
White Black
PM-3 Adr. 7
Whole House
CT Input
White Black White Black
2A ---------- CT ID --------- 3A 3B
ModHopper-2
Adr. 5
240vac
1 2 3 4
L1 L2 N G
240Vac
Wireless connection to EPRI-1
N G L2 L1
CT
CT Input:
Main Power Panel

Source: EPRI


House Floor Plan Layout
6-4-18
Rev.1.2
ZONE 1 =
ZONE 2 =
MBED
LA
KIT
MBATH
HALL BATH
BED 2
Tstat-1
Tstat-2
T3
T4
T5
T6
T7
T8
T9
T10
T11
T12
T13
T14
T15
9"
5"
6"
7"
8
DIN
FAM
WH
OFFICE
GARAGE
FAM
ENTRY

Source: EPRI

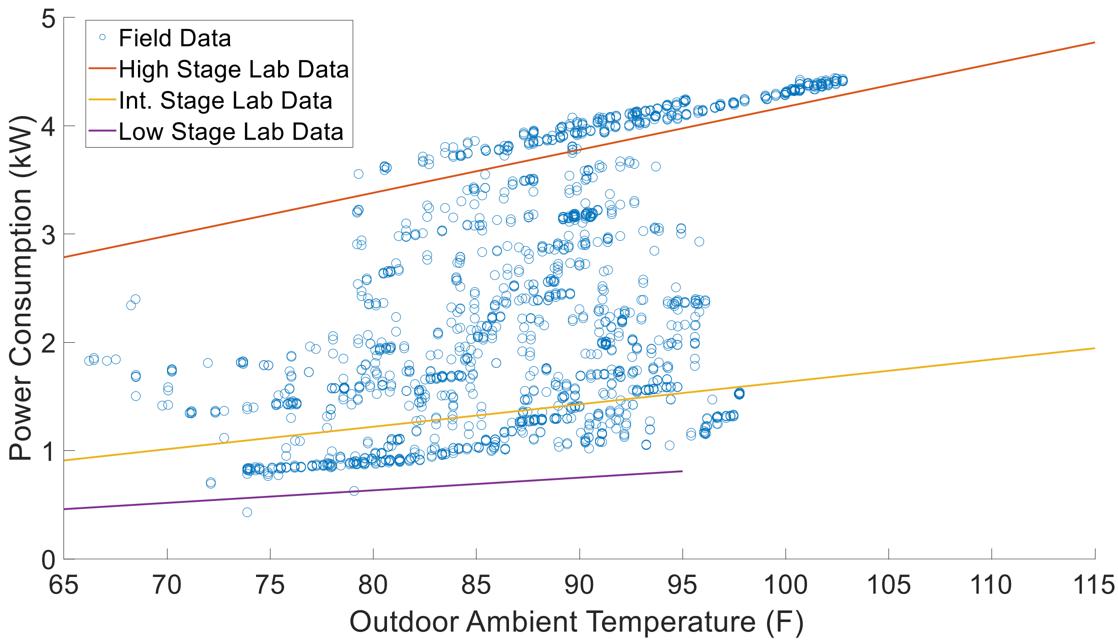
Point List

6-4-18
Rev.1.2

			6/4/2018
Point List- San Diego - CEC Residential HVAC Project - Sara Beaini			
Good Start Date>>			MAC: 001EC60-54B3B
Item	**Address-mb#**	**Description**	**Notes**
			Cell IP: 166.155.228.137; 704 589-5820
Data Box EPRI-1 Garage		**Box Near IDU**	
Acquisuite A8810-0 / Cell Modem		**Interval Data = 1min**	
PM-1; Indoor HVAC Unit Power	**Adr. 1**	**W2-M1-mA**	**120Vac; 1 CT MRS-075**
Flex-1	**Adr. 2**		
Gas Flow to Furnace	**CH 1**	**Pulse output gas flow meter**	**Model# BK250 w/ Pulse output 1p/Cf**
Ts = Supply Air Duct Temp	**CH 2**	**Dwyer In-Duct; wired**	
Hs = Supply Air Duct RH	**CH 3**	"	
Tr = Return Air Duct Temp	**CH 4**	"	
Hr = Return Air Duct RH	**CH 5**	"	
As = Main Supply Duct Pressure Diff.	**CH 6**	**Dwyer PAFS/616KD-B; 1inW.C.**	**Pitot Tube and 1% Transducer 4-20mA output**
A1 = Zone-1 Supply Air Duct Press. Diff.	**CH 7**	**Dwyer PAFS/616KD-LR; 0.5inW.C.**	**Pitot Tube and .25% Transducer 4-20mA output**
A2 = Zone-2 Supply Air Duct Press. Diff.	**CH 8**	"	"
Monnit Gateway - Alta	**Adr. 3**	**Wireless Sensors**	
T1 - OSA T/RH (OD HVAC Unit) (T/RH)	**Slot 0**	**386694**	**Industrial Monnit Sensor - OSA**
T2 - Attic Temp/RH - (T/RH)	**Slot 1**	**392073**	
T3 - Air Vent - Master Bd Rm. (T)	**Slot 2**	**386803, Zone One**	
T4 - Air Vent - Master Bath Rm (T)	**Slot 3**	**386702**	
T5 - Air Vent - Hall Bath Rm (T)	**Slot 4**	**386872**	
T6 Air Vent - Bed Rm 2 (T)	**Slot 5**	**386800**	
T7 Air Vent - Office (T)	**Slot 6**	**386781**	
T8 Air Vents - Living Rm (T)	**Slot 7**	**386705, Zone Two**	
T9 Air Vent - Kitchen (T)	**Slot 8**	**386703**	
T10 Air Vent - Dinning Rm (T)	**Slot 9**	**386725**	
T11 Air Vent - Entry Hall (T)	**Slot 10**	**386701**	
T12 Air Vent - Family Rm (T)	**Slot 11**	**386744**	
T13 Return Vent - Main Hall (T/RH)	**Slot 12**	**372694**	
T14 Return Vent 1 - Dinning Rm (T/RH)	**Slot 13**	**372662**	
T15 Return Vent 2 - Dinning Rm (T/RH)	**Slot 14**	**372706**	
T16 Garage Temp/RH (T/RH)	**Slot 15**	**372715**	
TStat-1 - Zone -1 Main Hall (T/RH)	**Slot 16**	**372697**	
TStat-2 - Zone -2 Din (T/RH)	**Slot 17**	**372713**	
ModHopper-1 DAQ	**Adr. 4, Ch 4**	**Wireless data collection from EPRI-2**	**MH-1 to MH-2**
Data Box EPRI-2		**Outside next to Power Panel**	
ModHopper-2 HVAC & House Power	**Adr. 5, Ch 4**	**Communication between Box-1 and Box-2**	
PM-2; Outdoor HVAC Unit	**Adr. 6**	**W2-M1-mA**	**240Vac; 1 CT MRS-075**
PM-3; Whole House Power	**Adr. 7**	"	**240Vac; 2 CT's House Mains**

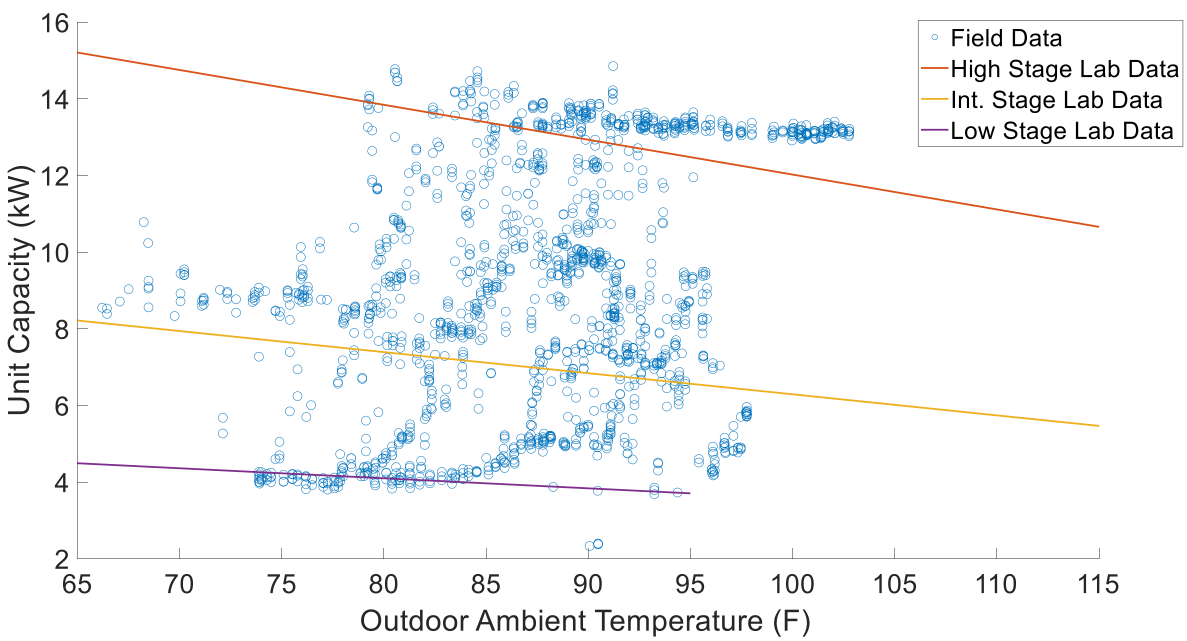
Source: EPRI

Figure G-1: Cooling Power Consumption as a Function of Ambient Temperature Compared to Scaled Lab Data (PG&E Site)



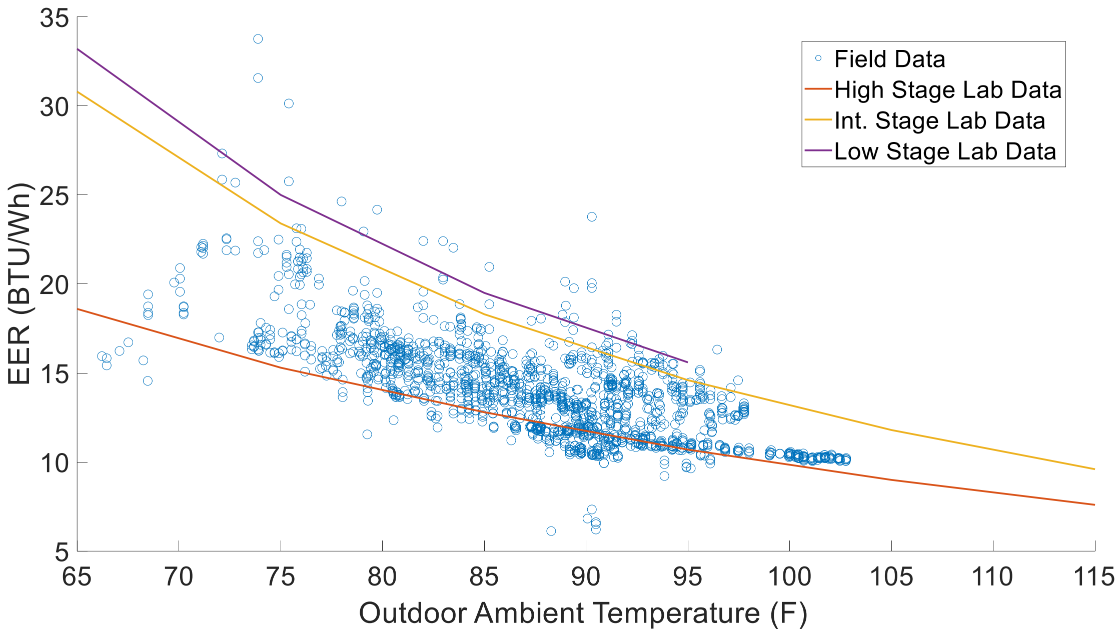
Source: EPRI

Figure G-2: Unit Cooling Capacity as a Function of Ambient Temperature Compared to Scaled Lab Data (PG&E Site)



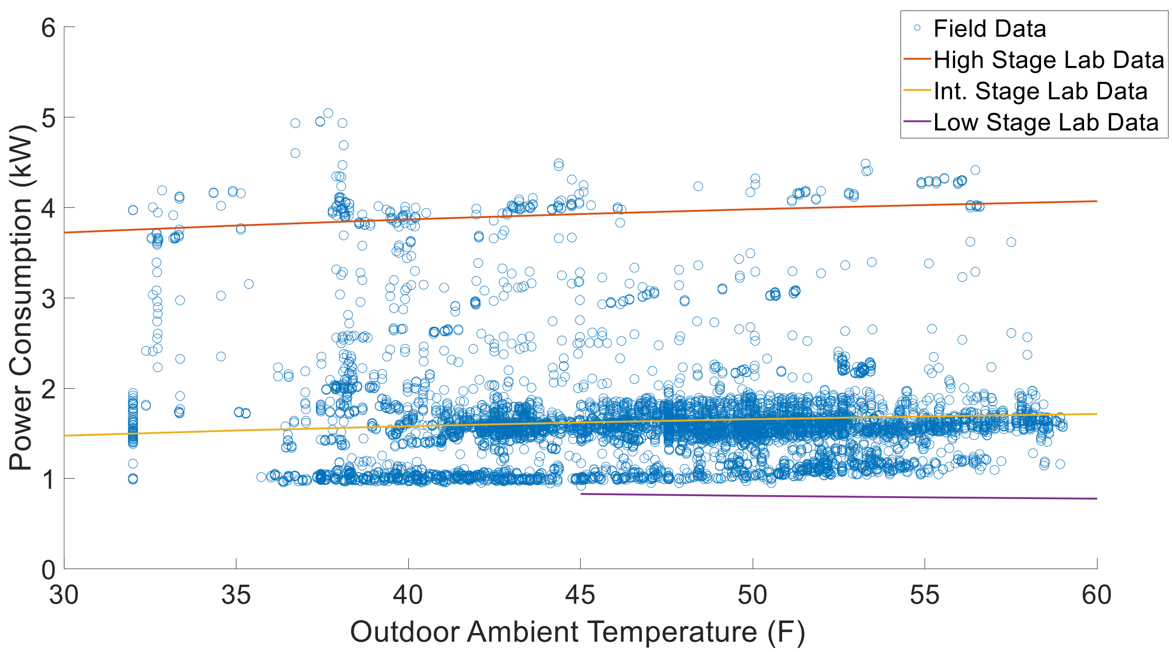
Source: EPRI

Figure G-3: Unit Cooling Efficiency as a Function of Ambient Temperature Compared to Scaled Lab Data (PG&E Site)



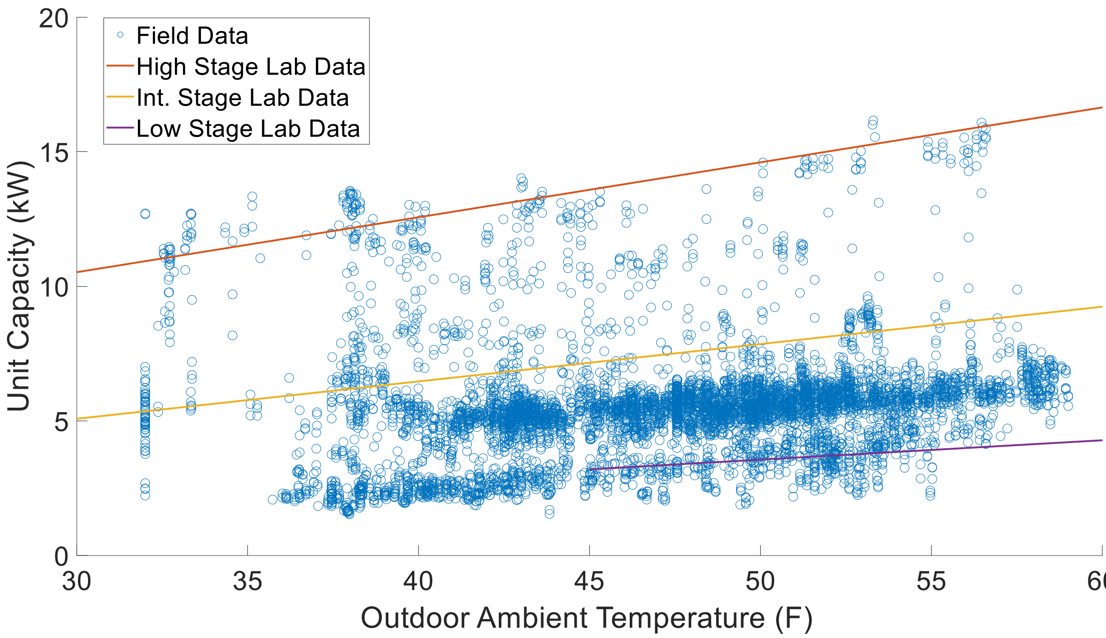
Source: EPRI

Figure G-4: Heating Power Consumption as a Function of Ambient Temperature Compared to Scaled Lab Data (PG&E Site)



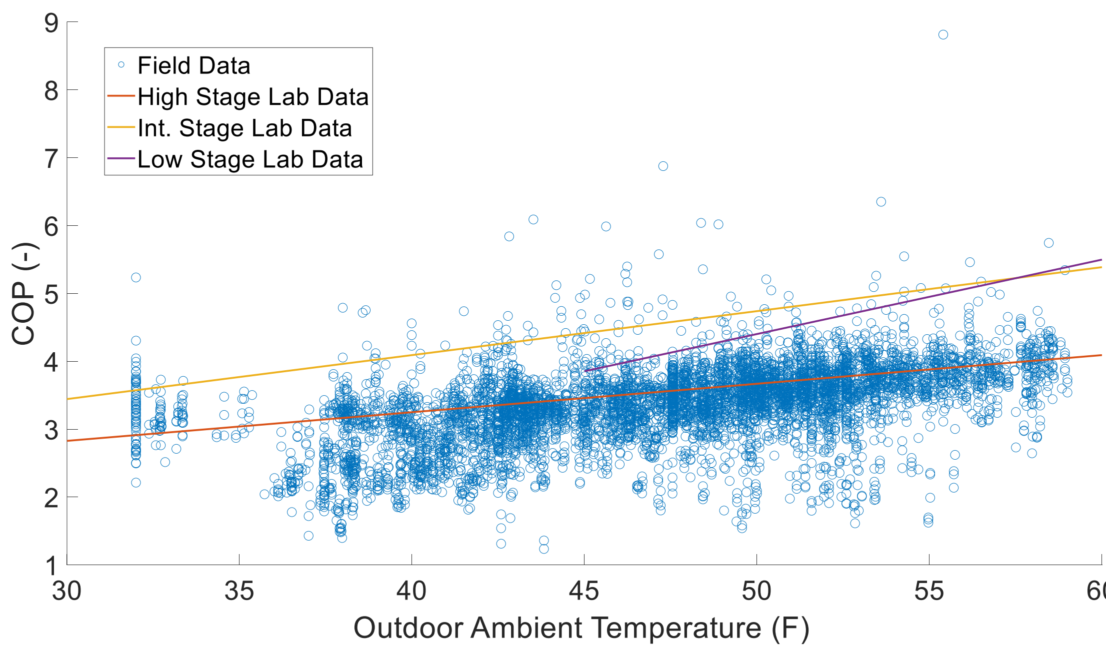
Source: EPRI

Figure G-5: Unit Heating Capacity as a Function of Ambient Temperature Compared to Scaled Lab Data (PG&E Site)



Source: EPRI

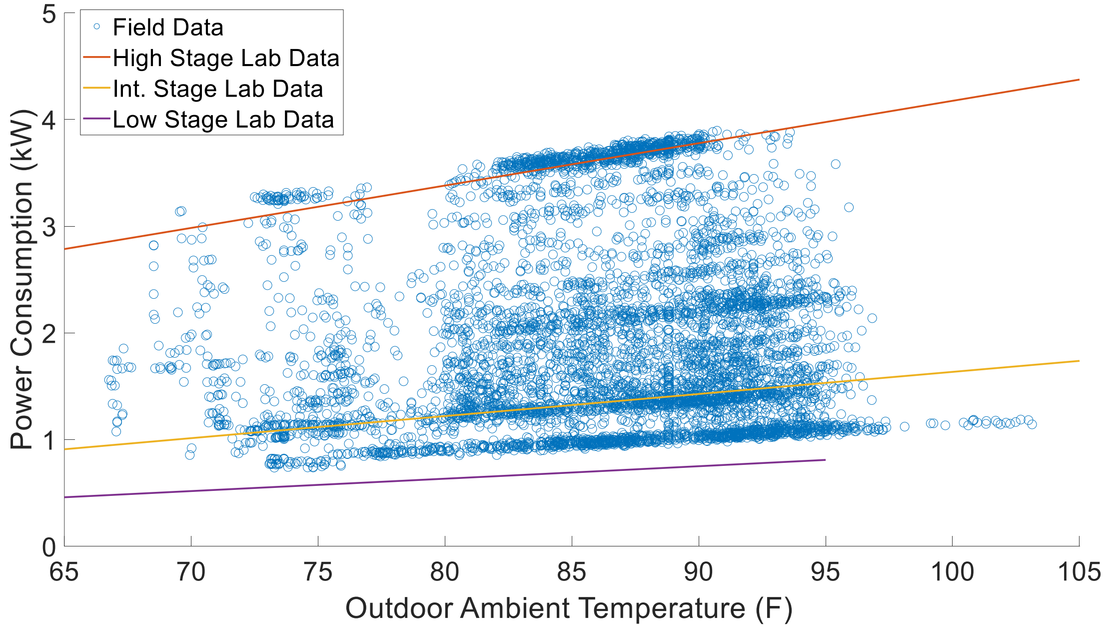
Figure G-6: Unit Heating Efficiency as a Function of Ambient Temperature Compared to Scaled Lab Data (PG&E Site)



Source: EPRI

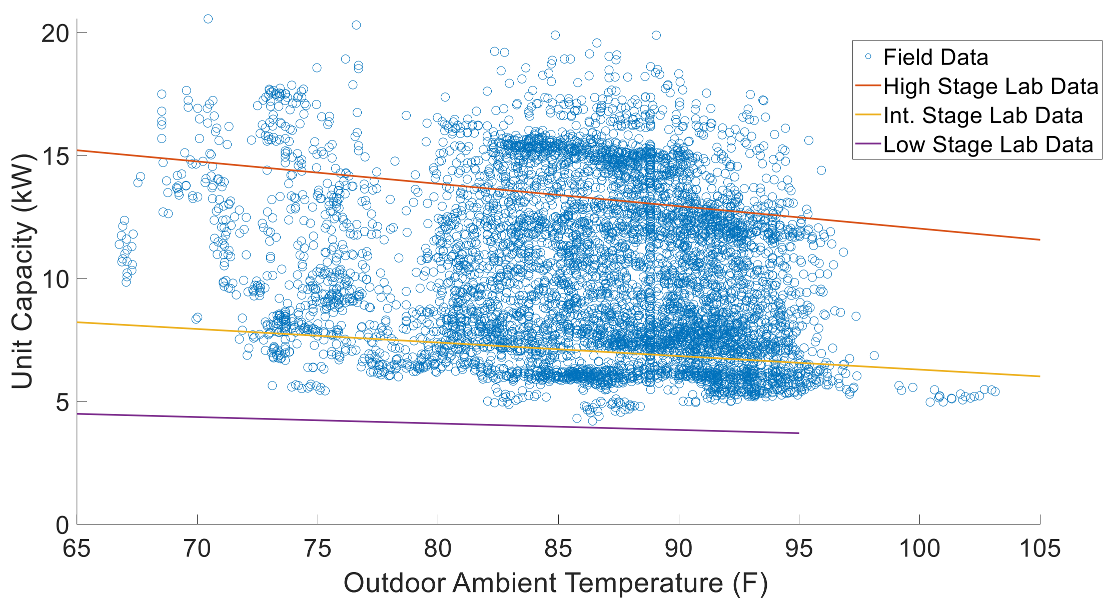
SDG&E PLOTS

Figure G-7: Cooling Power Consumption as a Function of Ambient Temperature Compared to Scaled Lab Data (SDG&E Site)



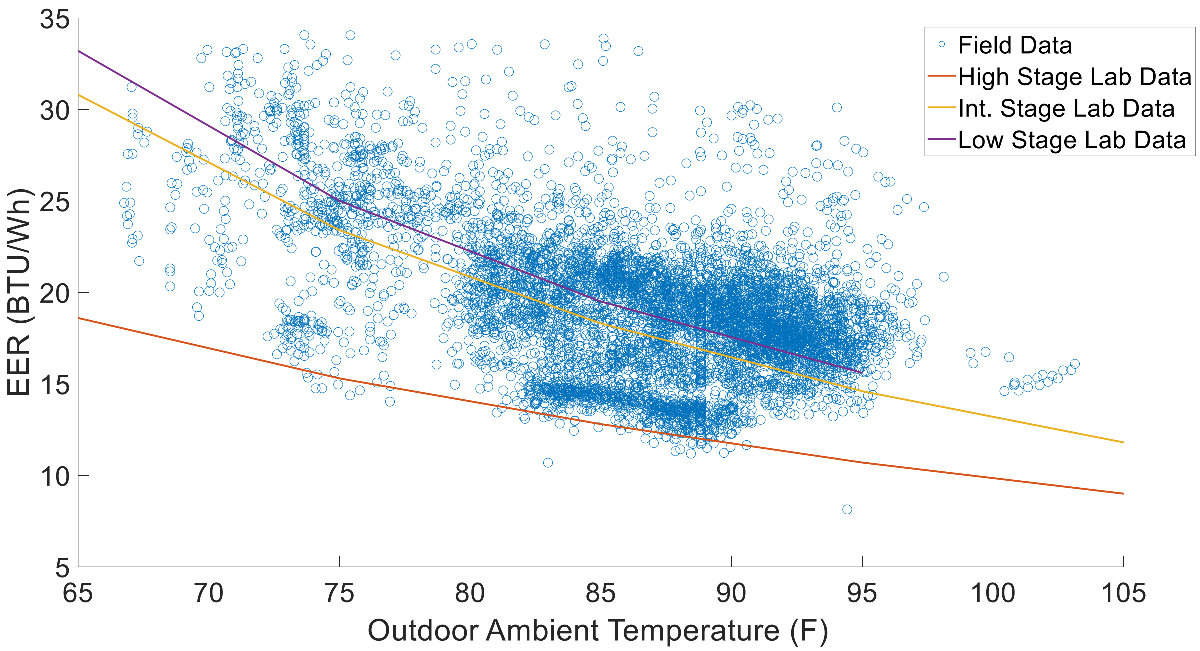
Source: EPRI

Figure G-8: Unit Cooling Capacity as a Function of Ambient Temperature Compared to Scaled Lab Data (SDG&E Site)



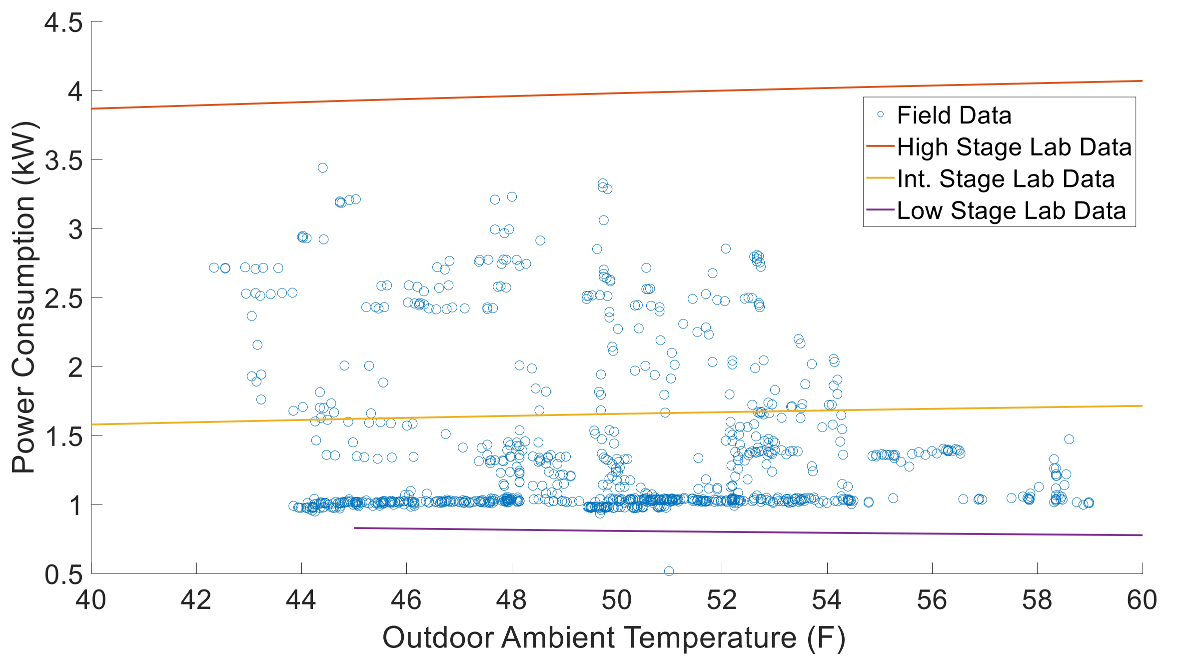
Source: EPRI

Figure G-9: Unit Cooling Efficiency as a Function of Ambient Temperature Compared to Scaled Lab Data (SDG&E Site)



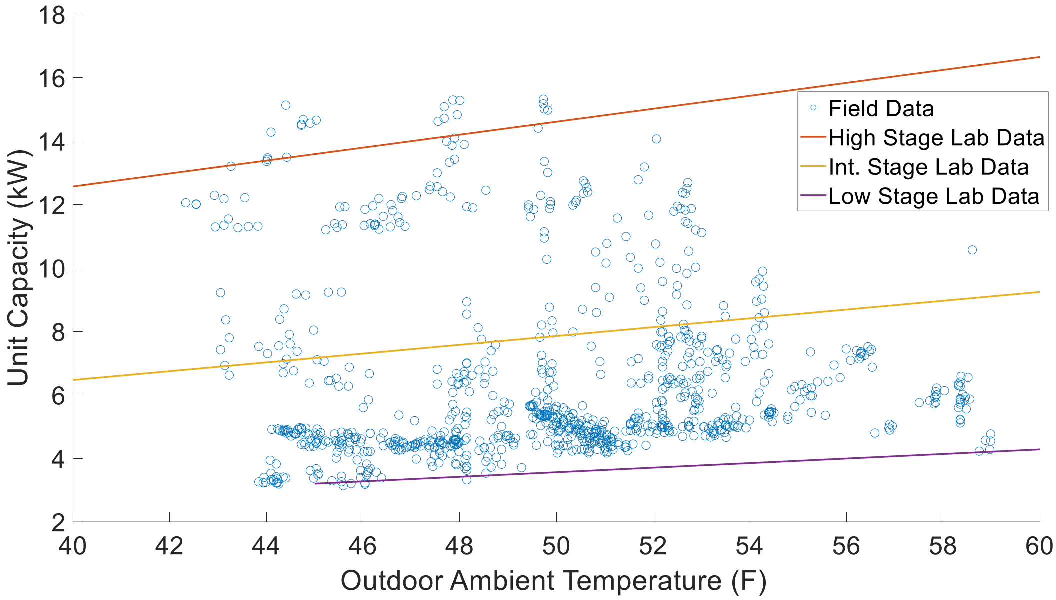
Source: EPRI

Figure G-10: Heating Power Consumption as a Function of Ambient Temperature Compared to Scaled Lab Data (SDG&E Site)



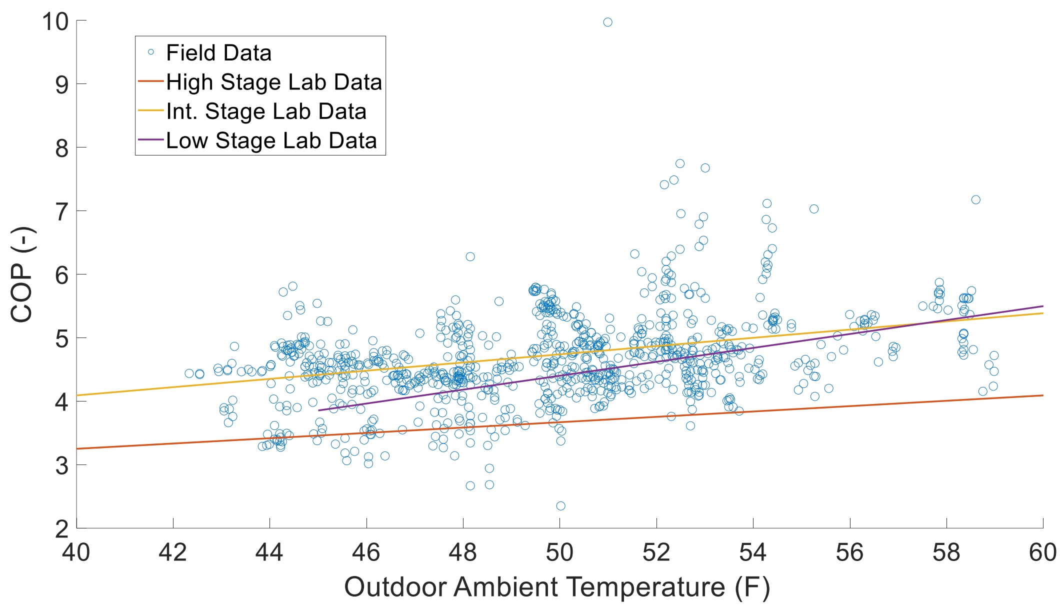
Source: EPRI

Figure G-11: Unit Heating Capacity as a Function of Ambient Temperature Compared to Scaled Lab Data (SDG&E Site)



Source: EPRI

Figure G-12: Unit Heating Efficiency as a Function of Ambient Temperature Compared to Scaled Lab Data (SDG&E Site)



Source: EPRI

Systems Adjustments at Each Host Site

Table G-1: PG&E Home

	Setting prior	Adjustment date
Zones weighting	Zone 1 (down front): Open run Zone 2 (down back): 34% Zone 3 (MBR): 33% Zone 4 (kids bedroom): 33%	On 11/14/18 Zone 1 (down front): Open run Zone 2 (down back): 40% Zone 3 (MBR): 30% Zone 4 (kids bedroom): 30% On 1/16/19 Zone 1 (down front): Open run Zone 2 (down back): 54% Zone 3 (MBR): 33% Zone 4 (kids bedroom): 33%
Airflow adjustments	–15% in low speed, 0 mid, 0 hi	
Balance point	Was set to 32F;	Changed to 38F on Nov 14, 2018; Changed to 40F on Feb 12, 2019 after furnace repair
Furnace	Heater began faulting, where it would not meet setpoint in morning. It was confirmed on 1/14/19 exhaust flue was not draining properly so the system would shut down the gas heat due to low combustion flow. Thus, locked out furnace operation until contractor repaired heater.	Jan 14, 2019 Changed control to avoid gas heat since system not heating home effectively Heater repaired on Feb 12, 2019
Pressure measurements	Dynamic pressure	Static pressure mode: Sept 28, 2018 to present. Only the main pressure was switched back to pitot on Nov 10, 2018 (system not used ~Oct 2018

Source: EPRI

Table G-2: SCE Home

	Prior to Jan 9, 2019	On and after Jan 9, 2019
Changed weighting of the zones using the EWC control board	Zone 1 (downstairs): 60% Zone 2 (kids bedroom): 35% Zone 3 (MBR): 5%	Zone 1 (downstairs): 60% Zone 2 (kids bedroom): 25% Zone 3 (MBR): 15% (sq.ft. area smaller than zone 2)
Airflow adjustments	15% on the low speed setting to help counter the overflow on the master bedroom (Zone 3)	Previously, was set to +15% for all speed settings (hi, inter, low)

Source: EPRI

Table G-3: SDG&E home

	Prior to Jan 10, 2019	On and after Jan 10, 2019
PID: allows EWC board to modulate capacity to meet supply setpoints	PID enabled	PID disabled
Changed weighting of the zones using the EWC control board	- Zone 1 weight (MBR and bedrooms; 12" ducting/ damper): 60% - Zone 2 weight (Kitchen, dining, living room; 14" ducting/ damper): 40%	- Zone 1 weight (MBR and bedrooms; 12" ducting/ damper): 40% - Zone 2 weight (Kitchen, dining, living room; 14" ducting/ damper): 60%
W-2 threshold	79% (switch to gas if total capacity needed is 79%+5%)	94% (switch to gas if total capacity needed is 94%+5%)
Airflow adjustments	15% on the low speed setting to help counter the overflow on the master bedroom (Zone 3)	Previously, was set to +15% for all speed settings (hi, inter, low)
Balance Point	20F	40F Dip switch change on indoor unit: Changed heating tab on furnace default from B to A. Kept cooling default at D

Source: EPRI

Stakeholders Webinar Q&A Summary

- Without backup electric heating strips? Gas heat as backup? What backup is being avoided?
  - The next-gen system has a natural gas furnace as part of the indoor air handling unit, but the system is also able to provide heat with the heat pump.
- What were the envelope insulation levels? to T24 code? Which code?
  - Envelope insulation based on T24; Host site specs provided in upcoming slides [see Table 38 in Chapter 4]
- Are these load reductions fixed (reduce to 50, reduce to 30% power) or relative (reduce by 50%, reduce by 70% from current operation)? At 100% output as the baseline, the answer is the same. I'd like to understand the controls capabilities, and potentially extrapolate impacts, under other baseline conditions (starting at 80% power).
  - The load reductions are relative, the DR testing started with the next-gen system operating at 90% of maximum power.
- What about the heating COP penalty?
  - The heating COP is generally lower than the cooling COP in heat pumps due to more focus on cooling design considerations.
- What is the added cost of all these upgrades to the baseline equipment?
  - It will depend on the manufacturer of the equipment.
- How did you calculate cost effectiveness? Did you include the cost of the heat pump and improvement costs (refrigerant, zone ducts etc.)?
  - The cost effectiveness will depend on the manufacturer of the equipment. This project evaluated the efficiency improvement of the system under each advanced feature.
- Could you say who did the ductwork study?
  - Western Cooling Efficiency Center
- Which report talks about 55% reduction/improvement, please? (FDD)
  - List of references are provided in Chapter 6 (section 6.3.4 for Fault Detection and Diagnostics)
- What was the size of the indoor coil?
  - The indoor unit was sized at 2 tons for the laboratory tests in Phase 1 and 2, and sized at 4 tons for the field tests in Phase 3, based on each home's energy load calculation done by the HVAC contractors and HERS rating personnel.
- Were the systems package units or split systems where the refrigerant travels throughout the home?
  - These are split system heat pumps, with refrigerant line imbedded in the home envelope, based on code specifications.
- Can you explain what you mean by the 90% annual heating load without backup? Does that mean that 90% of the load is provided by electricity rather than natural gas?

    - Yes, the system was able to meet 90% of the annual heating load of California in the lab modeling.
- If the same charge system was used for R-32 and R-410A, would the cooling performance be improved further? Or in other words, can you tease out the effect of the charge and cooling performance?
    - Refrigerant charge amount required is dependent on the refrigerant type/chemical product and the unit capacity. Reducing the amount of refrigerant charge (due to leakage) compromises the system performance.
- If all of these enhancements were added, what would be additive efficiency of this system compared to the baseline?
    - It will depend on the specific climate zone where the system is installed as well as the ducting installation, but on average it would be greater than 50%.
- You can have zone control with single-speed. Did you split out the advantages?
    - No, this project scope tested variable capacity heat pump system (VCHP) and added features and modeled the effect on VCH relative to single speed SEER 14 unit. Zoning with single speed would increase delivery effectiveness but there would also be an increase in fan power. The project researchers did not look at this, but in the lab testing did find a degradation in COP at a certain point when increasing fan speed. Single speed fans would also behave different than the inverter driven.
- Sara – You mentioned that the participants enjoyed less noise. Are there long-term data on variable speed less noise issues? In particular, it seems sometimes start up sounds differ.
    - Start-up sound with variable capacity heat pump is quieter than single speed AC units.
- Can the recommended variable speed SMART systems be turned off easily completely by the homeowner, with an on/off switch? For example, if the homeowner wants to have the windows open at night? Also, is the integrated ventilation idea relative to that concept? California does have some nice temperatures certain times of year that may not require a system to be on.
    - Yes, the homeowner can switch on/off the system or set different temperature thresholds using the Smart Thermostat as an interface.
- What are the HRV savings relative to? Is the baseline case vented with a simple exhaust device at the same airflow? Etc.?
    - The HRV savings are relative to baseline system that is a SEER 14 variable capacity air conditioner with forced air ventilation and a natural gas 80% AFUE furnace.
- Where were dampers installed in the ducts in conditioned zone? At the outlet register, or at the air handling unit outlet?
    - They were branches off the air handling unit outlet in the attic.
- In a single speed system, were heat buildup during the system cycling off at part load evaluated?

    - No, data analysis was only for steady state operation. The researchers tried to avoid transients in both field and lab when testing zone control
- Did the app that customers had allow them to modify the weighting on the zone control board themselves when they noticed the increased airflow in an individual duct or was that something that had to be done by a contractor?
    - The HVAC contractor can make the system control adjustments on the Zonal Control Board by the indoor unit (located in the attic or garage).
- Did any of the DR strategies include shutting down all but one zone?
    - No, the DR testing did not shut down particular zones. The project tested the DR event when all zones were open.
- What was assumed for efficiency when calculating the economic switchover point? I assume the heat pump COP as a function of outdoor temperature as applied.
    - The heating COP of the heat pump obtained from lab testing was used.
- In many areas in Southern California, the humidity varies by season -- Santa Ana conditions have single digit RH levels, but at other times of the year, wet bulb temperatures can be in the low to mid-70s. Can these systems address this seasonal variability?
    - No. The homeowner can set the temperature setpoint on the smart thermostat.
- My understanding is that the Title 24 Mandatory minimum duct insulation is R-4.2 while the Prescriptive duct insulation requirements vary by climate zone. Should the Mandatory minimum duct insulation requirements be increased? Or should the focus be put on higher Prescriptive duct insulation requirements?
    - There may be an argument for added insulation with variable capacity heat pumps, but the project researchers think zoning would provide a larger benefit
- Any thoughts how R-454B (A2L, GWP 466) might work in this type of system?
    - Further tests would need to be done to quantify the system performance.
- Have you considered evaluating R-454B or other similar HFC/HFO blends in addition to R-32?
    - No, the project scope, only considered R-32 as a drop-in refrigerant for R-410A.
- Did you consider ducts fully buried in attic insulation?

Only in the modeling (when looking at R-12). The project researchers are not sure what R-value buried ducts can achieve.