FINAL REPORT

High-Performance Air-Source Cold Climate Heat Pump (CCHP)

ESTCP Project EW-201721

OCTOBER 2020

Dr. Frederick Cogswell
United Technologies Research Center

Dr. Ahmad Mahmoud
Carrier Corporation

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This document has been cleared for public release

REPORT DOCUMENTATION PAGE

Form Approved
OMB No. 0704-0188

1. REPORT DATE *(10-26-2020)*

2. REPORT TYPE
ESTCP Final Report

3. DATES COVERED *(From - To)*
2/23/2017 - 04/2021

4. TITLE AND SUBTITLE
High-Performance Air-source Cold Climate Heat Pump

5a. CONTRACT NUMBER
W912HQ-17-C-0029

5b. GRANT NUMBER
NA

5c. PROGRAM ELEMENT NUMBER
NA

6. AUTHOR(S)
Dr. Frederick Cogswell (UTRC)
Dr. Ahmad Mahmoud (Carrier)

5d. PROJECT NUMBER
EW-201721

5e. TASK NUMBER
NA

5f. WORK UNIT NUMBER
NA

7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES)
United Technologies Research Center,
411 Silver Ln,
E.Hartford Ct. 06118
Maine Army National Guard,
Camp Keyes,
Augusta Me.

8. PERFORMING ORGANIZATION REPORT NUMBER
EW-201721

9. SPONSORING / MONITORING AGENCY NAME(S) AND ADDRESS(ES)

Environmental Security Technology Certification Program
4800 Mark Center Drive, Suite 16F16
Alexandria, VA 22350-3605

10. SPONSOR/MONITOR'S ACRONYM(S)

11. SPONSOR/MONITOR'S REPORT NUMBER(S) EW-201721

12. DISTRIBUTION / AVAILABILITY STATEMENT
Approved for public release; distribution is unlimited

13. SUPPLEMENTARY NOTES

14. ABSTRACT
Through this project United Technologies Research Center (UTRC) demonstrated a nominal 10TR high-efficiency cost-competitive Cold Climate air-source Heat Pump (CCHP) system at a Department of Defense installation in a relevant climate zone. The CCHP achieved >40% annual combined space heating and cooling energy savings, in ASHRAE climate zone 6A, while meeting the current pricing of state-of-the-art (SOA) mid-tier heat pumps. Demonstrations were conducted at the Maine Army National Guard headquarters at Camp Keyes in Augusta, ME and at UTRC's psychrometric chamber in E. Hartford, CT. SOA industry standard heat pumps can degrade by up to 60% in capacity and 50% in system efficiency at extreme heating conditions. Two key enabling technologies, high-efficiency high-lift compression and system-level design optimization for cold climates enabled this performance. During Phase I of this project a SOA Carrier heat pump unit was installed at the Maine field trial site, and a full year of data was obtained. During Phase II this unit was replaced by the new UTRC CCHP and another year of data was obtained. This project bought the prototype CCHP components and system to TRL8, demonstrated a > 30% decrease in annual energy consumption over state-of-the-art systems in all cold climates (Zones 4A through 7), and demonstrated site autonomous operation of the prototype CCHP.

15. SUBJECT TERMS
Cold Climate Heat Pump

16. SECURITY CLASSIFICATION OF: a. REPORT	b. ABSTRACT	c. THIS PAGE	17. LIMITATION OF ABSTRACT	18. NUMBER OF PAGES	19a. NAME OF RESPONSIBLE PERSON
UNCLASS	UNCLASS	UNCLASS	UNCLASS	67	Dr. Frederick Cogswell
					19b. TELEPHONE NUMBER *(include area code)* 860-610-1688/860-978-0261

Standard Form 298 (Rev. 8-98)
Prescribed by ANSI Std. Z39.18

FINAL REPORT
Project: #

TABLE OF CONTENTS

Page

TABLE OF CONTENTS (Continued)

Page

LIST OF FIGURES

Page

LIST OF FIGURES

Page

LIST OF TABLES

Page

ACRONYMS AND ABBREVIATIONS

BAA	Broad Agency Announcement
BMCS	Building Management and Control System
CCHP	Cold Climate Heat Pump
CFM	Cubic Feet per Minute
CO2	Carbon Dioxide
COP	Coefficient of Performance = Capacity / Power into unit
DoD	Department of Defense
da	In pyschrometric functions: Dry Air (not including mass of water)
ESTCP	Environmental Security Technology Certification Program
eGRID	Emissions & Generation Resource Integrated Database
°F/F	Degrees Fahrenheit
GHG	Green House Gas
ID	Indoor
kW	kilowatt
kWh	kilowatt-hour
O&M	operation and maintenance
OA	Outdoor Air
OAT	Outdoor Air Temperature
RA	Return Air
RAT	Return Air Temperature
RCRA	Resource Conservation and Recovery Act
RTU	Roof Top Unit.
SA	Supply Air
SAT	Supply Air Temperature
sCFM	Standard Cubic Feet per Minute
SOA	State of the Art
TC	Thermo-Couple temperature sensor
TMY3	Typical Meteorological Year weather data, version 3.
TR	Tons of Refrigeration (1TR = 12,000 Btu/hr, 3.5kW)
PPM	Parts per Million
VOC	Volatile Organic Compound
UTRC	United Technologies Research Center

ACKNOWLEDGMENTS

This material is based upon work supported by the Department of Energy under Award Number DE-EE0006108 and Department of Defense ESTCP W912HQ-17-C-0029.

This project would not have been successful without the help and support of Mr. Alan J. Ballard, Energy Director of the Maine Army National Guard, who oversaw all contracting work at the field trial site and provided invaluable feedback throughout the project.

ABSTRACT

INTRODUCTION AND OBJECTIVES

Through this project United Technologies Research Center (UTRC) demonstrated a nominal 10TR **high-efficiency cost-competitive Cold Climate air-source Heat Pump (CCHP) system** at a Department of Defense (DoD) installation in a relevant climate zone. The CCHP demonstration achieved >40% annual combined space heating and cooling energy savings, in ASHRAE climate zone 6A, while meeting the current pricing of state-of-the-art (SOA) mid-tier heat pumps. The demonstrations were conducted at the Maine Army National Guard headquarters at Camp Keyes in Augusta, ME and at UTRC's psychrometric chamber in E. Hartford, CT. This project aimed to 1) bring the prototype CCHP components and system to TRL8, (2) demonstrate > 20% decrease in annual energy consumption over state-of-the-art systems in all cold climates (Zones 4A through 7), and (3) demonstrate site autonomous operation of the prototype CCHP, operability and functionality.

TECHNOLOGY DESCRIPTION

The goal of the CCHP field-trail was to significantly improve on state-of-the-art (SOA) industry standard heat pumps that can degrade by up to 60% in capacity and 50% in system efficiency (COP) at extreme heating conditions. Two key enabling technologies, high-efficiency high-lift compression and system-level design optimization for cold climates enabled this performance. The CCHP is scalable beyond 40TR (140kW) nominal capacity, cost effective, and has no change to footprint and installation complexity from existing roof top units.

PERFORMANCE AND COST ASSESSMENT

During Phase I of this project a SOA (State of the Art) Carrier heat pump unit was used at the Maine field trial site, and a full year of data was obtained. During Phase II this unit was replaced by the new UTRC CCHP and another year of data was obtained. The new CCHP significantly outperformed the baseline unit, achieving >40% energy savings over the year while providing supply air temperature >100F thus avoiding "cold blow." The annual energy savings due to lower electric use ($3500) exceeded the incremental customer cost ($1200) and therefore meets the sales target of less than 2-year payback.

IMPLEMENTATION ISSUES

From the manufacture's perspective one of the main purposes of doing field trials is to advance the controls to TRL8. Relating to this, several modifications were successfully made to control set points to avoid nuisance trips and improve overall performance. There were several issues relating to the field trial implementation and data: a) the BSCM provided outdoor air temperature was found to be inaccurate, especially in cooling seasons, b) the temperature-averaged load between the two seasons was not consistent even though the zone was the same and the 2018 and 2019 temperatures were very similar, c) changing zone set points can have significant effect on load and the use of electric heat thus affecting unit efficiency, and d) cooling COP was lower than expected; this was due to cool return air temperatures and running at colder outdoor temperatures when the economizer option should have been utilized.

PUBLICATIONS

none.

EXECUTIVE SUMMARY

INTRODUCTION

Through this project United Technologies Research Center (UTRC) demonstrated a nominal 10TR **high-efficiency cost-competitive Cold Climate air-source Heat Pump (CCHP) system** at a Department of Defense (DoD) installation in a relevant climate zone. The CCHP demonstration achieved >40% annual combined space heating and cooling energy savings, in ASHRAE climate zone 6A, while meeting the current pricing of state-of-the-art (SOA) mid-tier heat pumps. The demonstrations were conducted at the Maine Army National Guard headquarters at Camp Keyes in Augusta, ME (***Figure 1***) and at UTRC's psychrometric chamber in E. Hartford, CT.

Figure 1. Building 39 at MEARNG Camp Keyes which Was the Primary Demonstration Site.

This project aimed to 1) bring the prototype CCHP components and system to TRL8, (2) demonstrate > 20% decrease in annual energy consumption over state-of-the-art systems in <u>all cold climates</u> (Zones 4A through 7), and (3) demonstrate site autonomous operation of the prototype CCHP, operability and functionality.

TECHNOLOGY DESCRIPTION

The goal of the CCHP field-trail was to significantly improve on state-of-the-art (SOA) industry standard heat pumps that can degrade by up to 60% in capacity and 50% in system efficiency (COP) at extreme heating conditions. Two key enabling technologies, high-efficiency high-lift compression and system-level design optimization for cold climates enabled this performance. The CCHP is scalable beyond 40TR (140kW) nominal capacity, cost effective, and has no change to footprint and installation complexity from existing roof top units.

The CCHP concept illustrated below (***Figure 2***) integrates 1) best-in-class tandem modulating scroll compressors for highly efficient part-load and high-lift full load operation, 2) a patented low-cost pre-compressor, 3) a cost-effective multi-speed indoor blower, 4) an optimized system (heat exchanger surfaces, insulation, etc.) and 5) advanced controls designed for cold ambient.

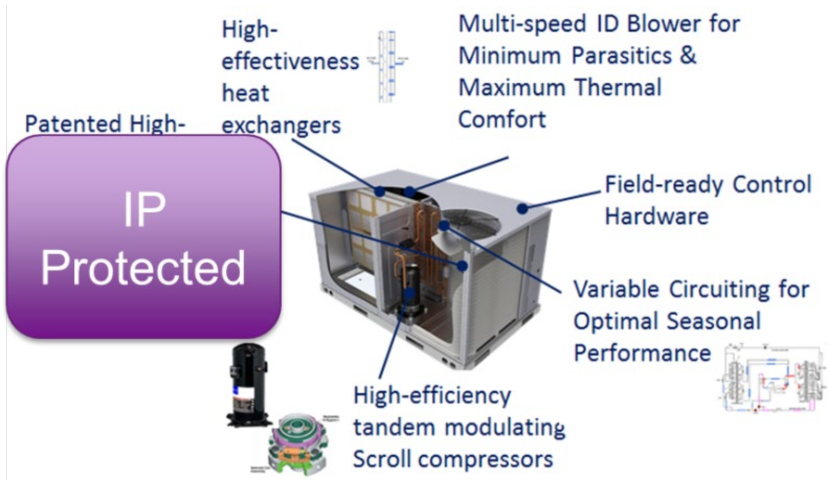


Figure 2. UTRC CCHP Packaged Rooftop System.

The performance of UTRC's CCHP compared to SOA heat pump systems is shown in ***Figure 3***, as demonstrated by a previous DOE project. The CCHP system maintains >95ºF supply air temperature to minimize thermal comfort issues and negative customer perception ("cold-blow"). This was achieved by providing extra compression capacity and through the use of a four-step variable speed indoor fan which was used to maintain discharge pressure and therefore supply air temperature within acceptable bands. The CCHP heating capacity at 0F and 17F is 93% and 48% greater than the SOA heat pump system, respectively. Delivering higher heating capacities at the price of current SOA heat pump systems would increase market penetration in climate zones where they are not currently being sold.

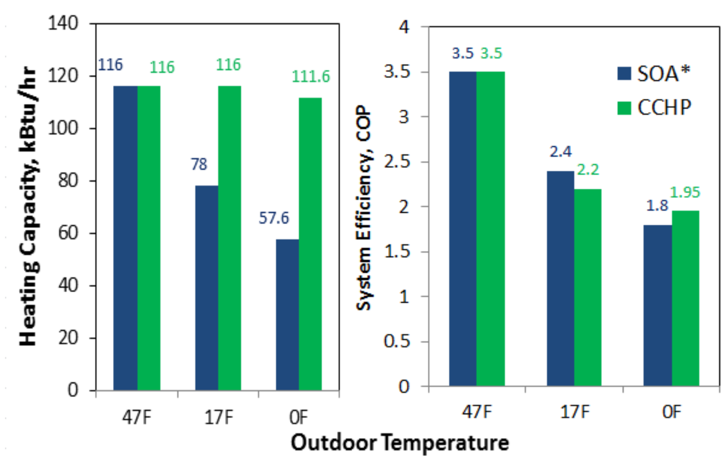


Figure 3. CCHP Laboratory Test Data vs. State-of-the-art (SOA) from Previous Projects.

OBJECTIVES

As mentioned above, the project's objectives were to 1) bring the prototype CCHP components and system to TRL8, (2) demonstrate > 20% decrease in annual energy consumption over state-of-the-art systems in all cold climates (Zones 4A through 7), and (3) demonstrate site autonomous operation of the prototype CCHP, operability and functionality.

The following table describes the quantitative and qualitative performance objectives:

Performance Objective	Metric	Data Requirements	Success Criteria	Results
Quantitative Performance Objectives				
Performance and Cost Avoidance	Annualized Energy Savings	Meter readings of power delivered to baseline and prototype systems; Price structure and calculation of cost avoidance by using the proposed technology	≥20% average annualized energy savings as tested in the field demonstrations.	Cost savings easily achieved with heating COP ~twice baseline. >30% savings calculated for all cold-climate zones. Cost premium $1200
Air Supply Temperature	ºF	Measure air supply temperature from the packaged rooftop unit; Optimize operation of ID blower to maintain desired SAT.	≥95ºF SAT maintained through controls of components, independent of load or ambient temperature.	Achieved using indoor air flow rate control. Demonstrated SAT typically >100F.
Direct Greenhouse Gas Emissions	Direct fossil fuel emissions, metric tons	Estimated release of greenhouse gases based on source of energy if technology had not been implemented	>20% reduction (indirect) in green-house gases over SOA heat pump systems corresponding to energy consumption reduction.	Achieved with COP improvement since electricity consumption corresponds to both cost and CO2 production.
Scalability Across DoD	%	Bin hour analysis and local weather data to calculate annual energy savings to determine value proposition	>56% of DoD installations reside in cold climate zones where CCHP technology can reduce dependence on fossil fuels.	Technology demonstrated can be applied to any rooftop size, and is therefore applicable to any office building with electric heat.
Qualitative Performance Objectives				
Ease of Use	Level of expertise/ skill of intended end user	Survey and/or interview results describing the difficulty of using the technology as designed	Exceeds expectations of key stakeholders with regards to maintenance and operation.	Standard Rooftop installation. No operational maintenance required for product.
Thermal Comfort	Feedback of occupants and operations staff	Survey and/or interview results describing the experience of the base personnel during the prototyping phase	Exceeds expectations of occupants with regards to thermal comfort during heating and cooling seasons	AJ. Ballard (MEARNG) reports improved customer satisfaction.

PERFORMANCE ASSESSMENT

During Phase I of this project a SOA (State of the Art) Carrier heat pump unit was used at the Maine field trial site, and a full year of data was obtained. During Phase II this unit was replaced by the new UTRC CCHP and another year of data was obtained. The new CCHP significantly outperformed the baseline unit, achieving >40% energy savings over the year while providing supply air temperature >100F thus avoiding "cold blow." ***Figure 4*** and ***Figure 5*** show an example of transient data from the baseline and CCHP units. The green lines are electrical power consumption; >40kW means the electric heater was on. For the baseline, the unit ran continuously as it cycled the electric heater on and off to meet load. For this period the electric heat was on 40% of the time, and when the electric heater was not on, the supply air temperature was below 80F (causing cold-blow). For the CCHP there was very little electric heat required. The spikes are mostly due to defrost-cycles. Also, even without electric heat the supply air temperature exceeded 100F.

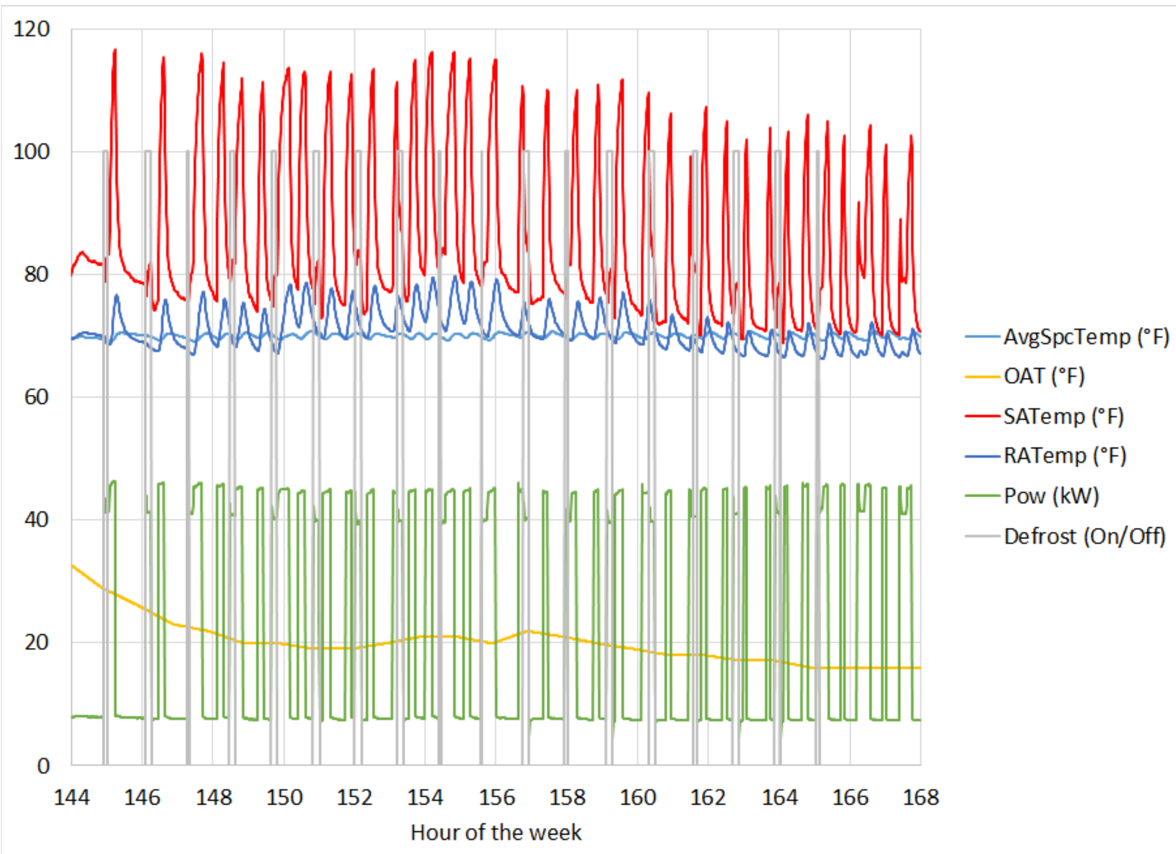


Figure 4. Baseline Transient Data from Nov. 14th, 2018.

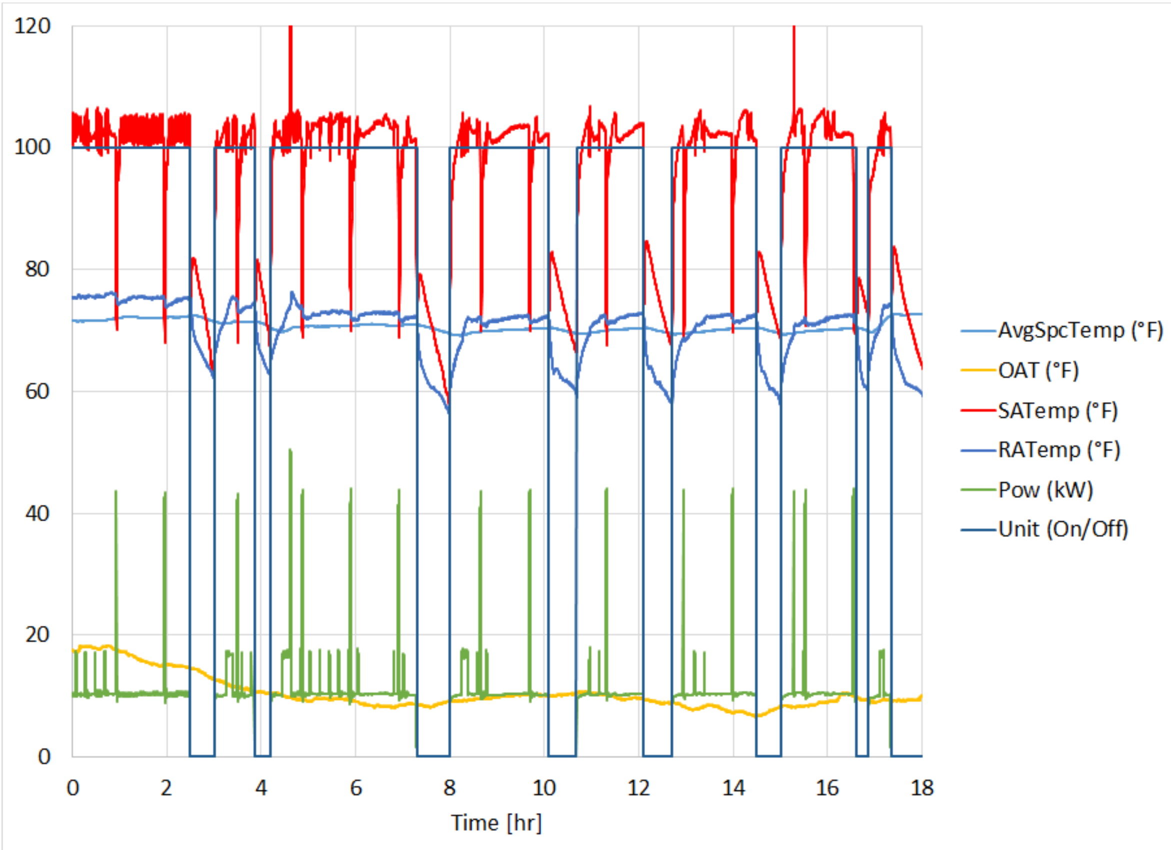
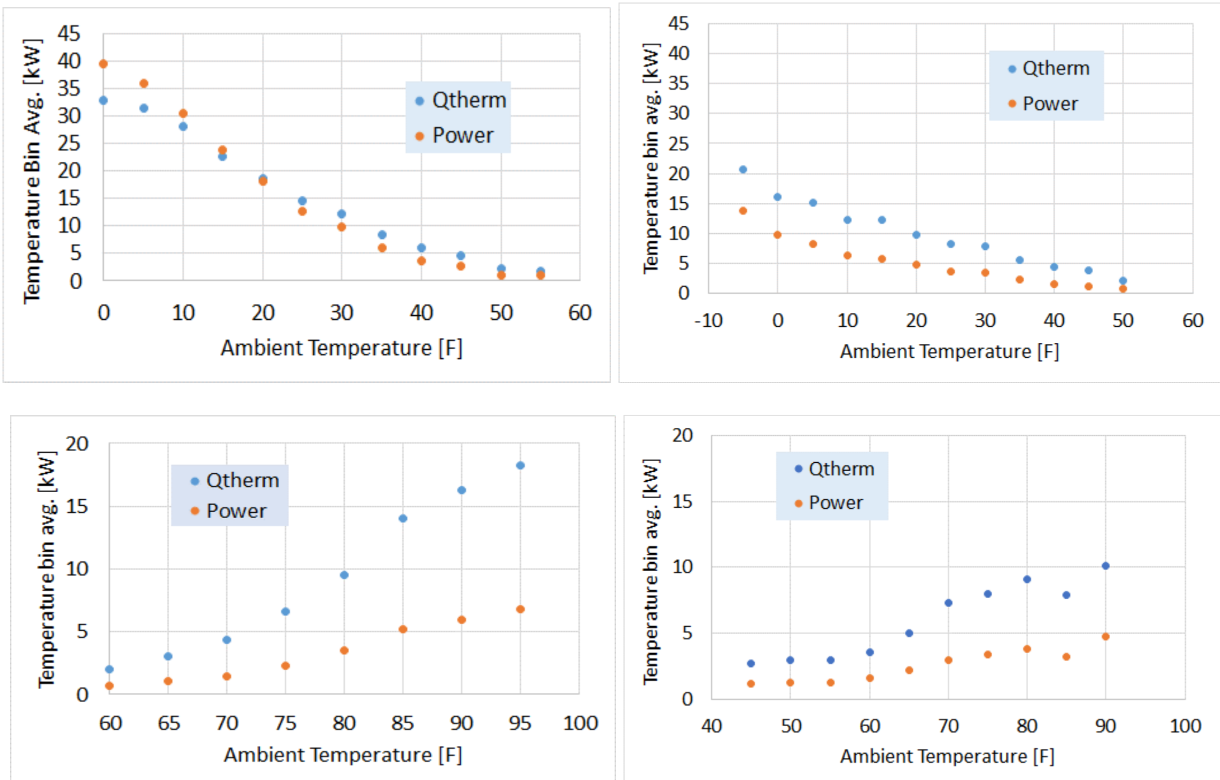


Figure 5. CCHP Transient Data from Jan. 22nd to 23rd, 2019

The transient data over each year (baseline 2018 and CCHP 2019) were integrated to determine the time-averaged load, capacity and COP (=Load/Capacity). Four hour averages were used:

$$COP = \frac{\int Q(t)dt}{\int P(t)dt}; \qquad (t_2 - t_1 = 4\text{hr})$$

These averages were temperature-binned and summed over the full year, and relations were developed for averaged load and efficiency (COP) as a function of outdoor temperature. ***Figure 6*** shows the binned-averaged results for load and power as function of ambient for both heating and cooling seasons. Note that the 2019 averaged load was lower even though the peak loads were about the same (this is discussed in the "Lessons Learned" section below). ***Figure 7*** shows the averaged COP for the baseline and CCHP for both heating and cooling. The CCHP far exceeded the baseline in heating mode, but fell short compared to the baseline in cooling mode (this is also discussed in "Lessons Learned"). ***Figure 8*** shows the required capacity and power consumed using a) the 2018 load line (higher), b) each COP function, and c) TMY3 data for Augusta Maine. It can be seen that the heating season dominates the annual power consumption.



Phase I (baseline) ***Phase II (CCHP)***

Figure 6. Baseline - left and CCHP - right: Temperature-bin-averaged Capacity and Power for Heating (Top) and Cooling (Bottom).

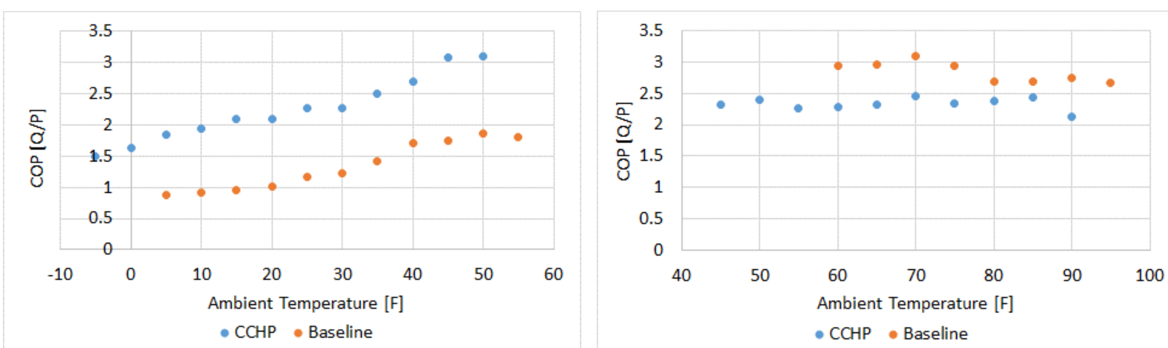


Figure 7. Temperature-bin-averaged COP.

Left: heating; Right: cooling.

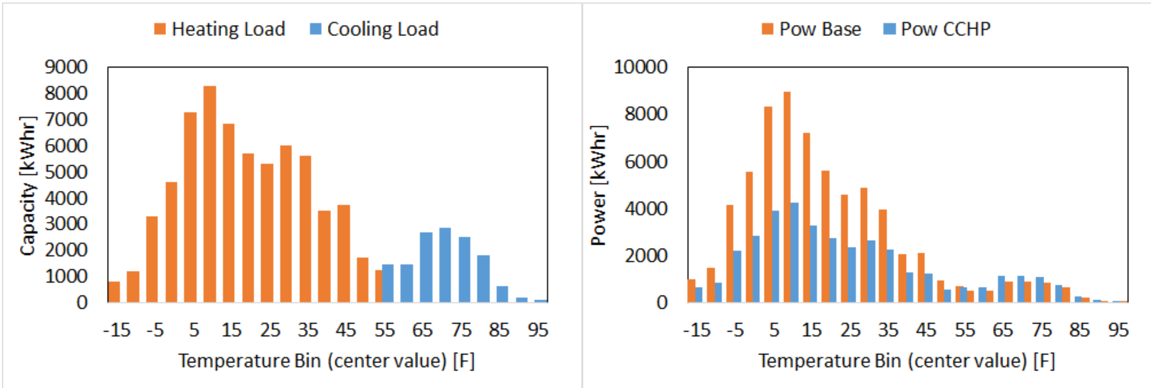


Figure 8. Left: Temperature-bin-summed Load; Right: Temperature-bin-summed Power.

The annual power consumption is determined by summing over each hour of the year. The table below shows the results for Augusta and 3 other locations covering the ASHREA zones 4A to 7A. The energy savings are based on the baseline and CCHP units' COP (Coefficient of Performance) and averaged thermal load, as determined from the field trial data, as a function of OAT. A simple value of \$0.12/kWh was used for electricity cost. It can be seen that the CCHP saves >30% for each zone.

Table 1. Energy and Cost Savings for 4 Different Climate Zones.

Fort Knox, Kentucky
ASHRAE Zone 4A

	Base Unit	CCHP	Savings	%
Annual kWh	41039	28481	12558	31%
Electric $/kWh	0.12	0.12		
Annual Cost	$ 4,925	$ 3,418	$ 1,507	

Colorado Springs, CO
ASHRAE Zone 5B

	Base Unit	CCHP	Savings	%
Annual kWh	46766	28842	17924	38%
Electric $/kWh	0.12	0.12		
Annual Cost	$ 5,612	$ 3,461	$ 2,151	

Augusta Maine,
ASHRAE Zone 6A

	Base Unit	CCHP	Savings	%
Annual kWh	66161	37226	28935	44%
Electric $/kWh	0.12	0.12		
Annual Cost	$ 7,939	$ 4,467	$ 3,472	

Houlton, Maine
ASHRAE Zone 7A

	Base Unit	CCHP	Savings	%
Annual kWh	87451	47908	39543	45%
Electric $/kWh	0.12	0.12		
Annual Cost	$ 10,494	$ 5,749	$ 4,745	

COST ASSESSMENT

There are three primary costs associated with the proposed technology

- Purchase,

- Installation, and
- Operational cost.

The technology developed by UTRC for the CCHP has been designed to meet the current pricing of state-of-the-art (SOA) mid-tier heat pumps. In addition, since standard RTU packaging has been maintained, the installation is the same as for a standard cooling/heating RTU.

The added compression capacity provided by the CCHP is calculated to cost < \$700 (or approximately \$1200 cost increase to customer). The annual energy savings due to lower electric use of \$3500 exceeds the incremental unit cost and allows for a <1 year payback.

IMPLEMENTATION ISSUES

From the manufacture perspective, the purpose of doing field trials is to look for problems that did not show up in laboratory testing, and to advance the controls to TRL8. Along this line, during the field trials several modifications were successfully made to controls set points to avoid nuisance trips and to improve the overall performance. There were no product issues (such as poor oil return or unstable control) found. From the government and sales perspective, the purpose of these field trials was also to document product operational efficiency and ease of use. Along these lines several issues did come up relating to performance of the field trials:

a) The BSCM provided outdoor air temperature was found to be inaccurate, especially in cooling seasons. The outdoor air temperature is important in that it is used to calculate ventilation load and is used as the independent variable for the COP and capacity functions. Fortunately, the Augusta Airport is right next door to the field trial site, and airport data was able to be downloaded and used in the analysis.

b) The averaged load between the two years (baseline 2018 and CCHP 2019) as a function of temperature was not consistent. This appears to violate the control variable of testing the two units in exactly the same building zone over the same period of time. On the other hand, the peak loads were found to be consistent, but there was more scatter in the 2019 data. The reason for this is not known for sure, but it is likely that differences in neighboring zone set points may have caused this issue.

c) Changing zone set points can have significant effect on load and the use of electric heat. Zone occupancy scheduling is common practice in the industry, but with an electric heat pump this can actually cost energy. Many times it was noticed that CCHP electric heater was activated when the zone setting was increased. This significantly lowered the COP during these times. For most of the heating season with the CCHP, other than for during the defrost cycle, the electric heat was not needed.

d) Cooling COP was lower for the CCHP. Analysis of the field data indicated that this was due to a combination of a) running in cooling mode at low temperatures but not implemented the economizer, b) cool return air temperatures from the zone leading to cool supply air temperatures and therefore lower compressor suction pressures, and 3) insufficient air flow given the above. After the field trial, the unit was returned to UTRC's psychrometric facility for testing. Under the standard ARI conditions (Return air = 80 dry bulb, 67 wet bulb, and supply air = 56 dry bulb) the unit was found to still meet specifications.

1.0 INTRODUCTION

Through this project United Technologies Research Center (UTRC) demonstrated a nominal 10TR **high-efficiency cost-competitive Cold Climate air-source Heat Pump (CCHP) system** at a Department of Defense (DoD) installation in a relevant climate zone. The CCHP demonstration achieved >40% annual combined space heating and cooling energy savings, in ASHRAE climate zone 6A, while meeting the current pricing of state-of-the-art (SOA) mid-tier heat pumps. The demonstrations were conducted at the Maine Army National Guard headquarters at Camp Keyes in Augusta, ME and at UTRC's psychrometric chamber in E. Hartford, CT. This project aimed to 1) bring the prototype CCHP components and system to TRL8, (2) demonstrate > 20% decrease in annual energy consumption over state-of-the-art systems in all cold climates (Zones 4A through 7), and (3) demonstrate site autonomous operation of the prototype CCHP, operability and functionality.

1.1 BACKGROUND

Technology Description: The goal of the CCHP field-trail was to significantly improve on state-of-the-art (SOA) industry standard heat pumps that can degrade by up to 60% in capacity and 50% in system efficiency (COP) at extreme heating conditions. Two key enabling technologies, high-efficiency high-lift compression and system-level design optimization for cold climates enabled this performance. Previous analysis was done under a U.S. Department of Energy program to advance and demonstrate system performance over a wide range of ambient conditions, delivering heating performance at low cost while maintaining cooling performance. The CCHP is scalable beyond 40TR (140kW) nominal capacity, cost effective, and has no change to footprint and installation complexity from existing roof top units.

Expected Benefits: In addition to > 20% annual energy savings, supporting the DoD energy security and energy efficiency goals, tangible benefits from this project include the following:

- *Institutional Scalability:* The 10TR air-source CCHP that was demonstrated is unique in that it allows high efficiency and high capacity operation over a wide-range of ambient temperatures and modes of operation. This enables technology deployment in several climate zones within the U.S. UTRC projects that > 56% of DoD installations are located in areas where a CCHP is a viable heating and cooling solution. The demonstrated CCHP is easily scalable from the 5TR to 12.5TR+ sizes which are currently the highest volume packaged rooftops installed at DoD bases.
- *Remote Heating Solution:* The DoD supports a great number of facilities, across all military services, throughout the globe in strategic cold climate regions. In many instances, these facilities are in remote areas that are far removed from the natural gas grid. Logistics associated with the transport of heating oil or propane may also be challenging. Electricity is readily available across most locations. Current limitations (capacity, efficiency and supply air temperature) in SOA heat pumps are an obstacle to wide-scale cost-effective deployment of electricity based heating solutions. A cost-competitive, high-capacity and high-efficiency CCHP solution could change the logistical costs associated with heating.
- *Annual Energy Savings and Payback:* Projected savings may vary significantly depending on the age and retrofit status of the building being analyzed. As a result of increased capacity and efficiency, the demonstrated system was shown to have greater than 20% annual reduction in HVAC energy consumption across relevant climate zones.

This represents a potential annual energy savings of 14 trillion Btu and $61 million. Additional savings associated with the price volatility and dependence on hydrocarbon fuels is not factored here. An additional $2.3B annual energy cost savings could be achieved with successful widespread deployment of the CCHP for the commercial US building stock in the relevant zones (4A to 7).

1.2 OBJECTIVE OF THE DEMONSTRATION AND REGULATORY DRIVERS

This project was in response to Environmental Security Technology Certification Program (ESTCP) Broad Agency Announcement (BAA) fiscal year (FY) 2017, and in support of Energy Policy Act (EPAct) 2005 and Executive Order (EO) 13423. The primary goal of the project was to demonstrate at least 20% annual space cooling and heating energy savings, in American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) climate zones 4A to 7, while meeting the current pricing of state-of-the-art (SOA) mid-tier heat pumps. Through demonstrations that were conducted at the Maine Army National Guard headquarters at Camp Keyes in Augusta, ME and at UTRC's psychrometric facility in E. Hartford, CT. This project:

1. Brought the prototype CCHP components and system to technology readiness level 8(TRL8),
2. Demonstrated > 20% relative decrease in annual energy consumption over state-of-the-art systems, and
3. Demonstrated site autonomous operation of the prototype CCHP, operability and functionality.

2.0 TECHNOLOGY DESCRIPTION

The demonstrated system improves on state-of-the-art (SOA) industry standard heat pumps that can degrade by up to 60% in capacity and 50% in system efficiency (measured as Coefficient of Performance or COP) at extreme heating conditions. Two key enabling technologies, high-efficiency high-lift compression and system level design optimization for cold climates enables this performance. Previous analysis was done under a U.S. Department of Energy program to advance and demonstrate system performance over a wide range of ambient conditions, delivering heating performance at low cost while maintaining cooling performance. The CCHP is scalable beyond 40TR (140kW) nominal capacity, cost effective, and has no change to footprint and installation complexity from existing roof top units.

2.1 TECHNOLOGY OVERVIEW AND ADVANTAGES

The CCHP concept illustrated below (***Figure 9***) integrates 1) best-in-class tandem modulating scroll compressors for highly efficient part-load and high-lift full load operation, 2) a patented low-cost pre-compressor, 3) a cost-effective multi-speed indoor blower, 4) an optimized system (heat exchanger surfaces, insulation, etc.) and 5) advanced controls designed for cold ambient.

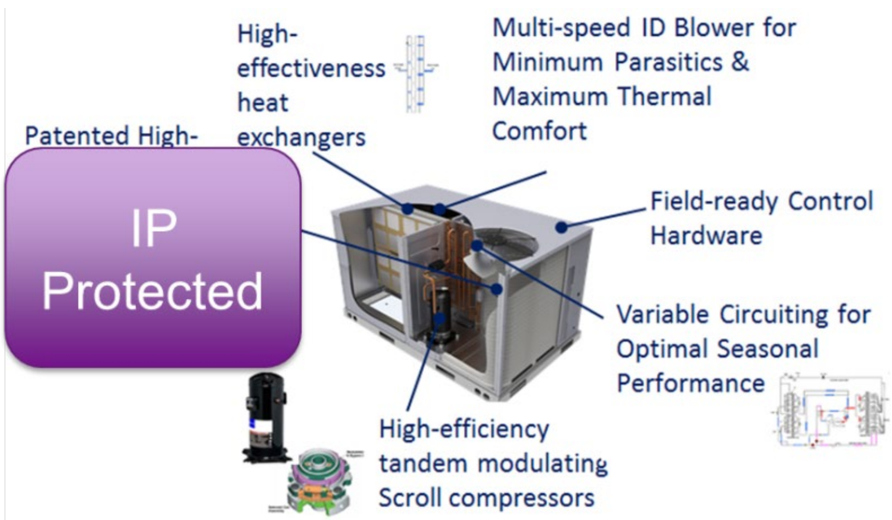


Figure 9. UTRC CCHP Packaged Rooftop System.

The performance of UTRC's CCHP compared to SOA heat pump systems is shown in ***Figure 10***, as demonstrated by the previous DOE project. The CCHP system maintains >95^{o}F supply air temperature to minimize thermal comfort issues and negative customer perception ("cold-blow"). This was achieved through a trade-off of COP and heating capacity to deliver a sufficiently high supply temperature. The CCHP heating capacity at 0F and 17F is 93% and 48% greater than the SOA heat pump system, respectively. Delivering higher heating capacities at the price of current SOA heat pump systems would increase market penetration in climate zones where they are not currently being sold.

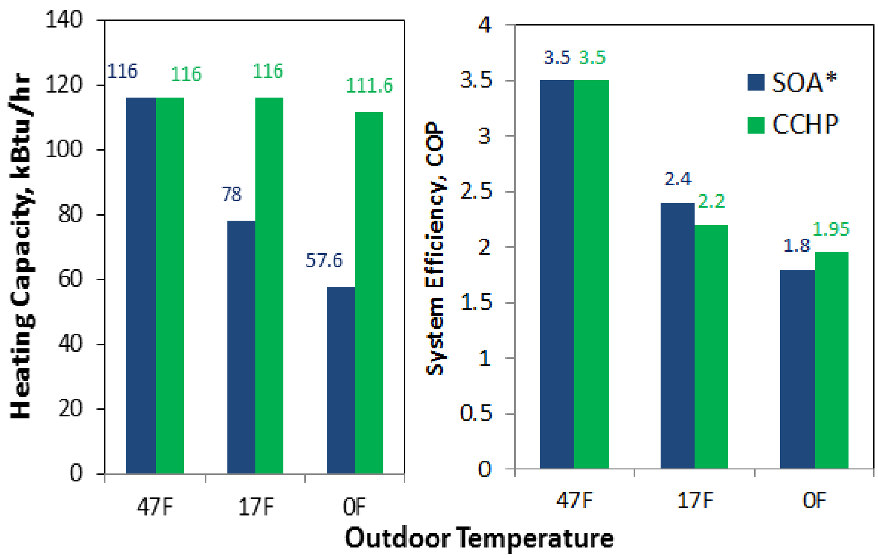


Figure 10. CCHP Laboratory Test Data vs. State-of-the-art (SOA) from Previous Projects.

The rational used to quantify the building energy savings realized can be seen below in ***Figure 11***, which shows the variation of heating load for a typical occupied commercial office building along with the capacity profile of 1) an installed SOA heat pump system (Blue) and 2) the demonstrated cost-effective CCHP system (Green) as a function of ambient temperature. The regions above the system capacity lines represent building loads where heat pump system capacity needs to be supplemented with electrical heat. The so-called balance point is the condition where the building heat load is met by the heat pump employed with no need for supplemental electrical resistive heat. In this example, the proposed CCHP system shifts the balance point by >25F delaying the need for supplemental electric heat and ultimately reducing the required amount of electric resistive heat consumed. The shift in balance point is dependent on many application specific variables such as climate zone, building type and occupancy schedule. In addition to the projected savings for DoD buildings, the DOE estimates that cold climate heat pumps at full market penetration could save 0.27 Quads of energy annually in residential homes alone. Further, UTRC estimates that $2.3 billion could be saved by utilizing high-efficiency CCHP in the domestic commercial building stocks within U.S. cold climates. To realize these heating and cooling capacity benefits, UTRC had (prior to this project) applied proprietary modeling tools to design and build several hardware variants to experimentally qualify the design and performance of the low-cost tandem compression technology and to integrate a low-cost pre-compressor to boost system efficiency.

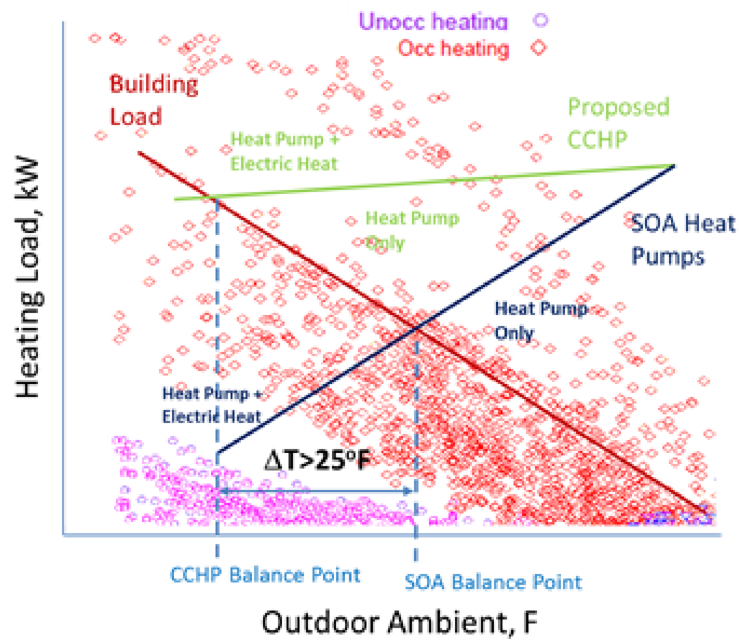


Figure 11. Occupied Building Heating Load & Capacity Schematic of SOA vs. Proposed CCHP System Capacity Profiles.

(Typical small office building in a cold climate).

During the cooling season the CCHP heat pump must also maintain or exceed a SOA unit's cooling performance. Extra compressor capacity has been added to the CCHP in order to meet low-ambient heating needs. The challenge facing the industry when it comes to CCHP technology, is to do so in a cost-effective manner. UTRC has done this by using tandem modulating compressors and a unique chassis and heat exchanger combination. Other SOA systems such as VRF (Variable Refrigerant Flow) are significantly more expensive than the packaged roof-top market that the prototype CCHP will disrupt. The control logic, developed for this project, runs only one compressor at a time for cooling, and therefore the cooling capacity is not oversized. Also, since the two compressors share a single circuit, the unit runs efficiently with properly sized heat exchangers.

2.2 TECHNOLOGY DEVELOPMENT

UTRC began development of this product under a DOE BTO contract that began back in 2013. That project successfully met performance objectives down to -13F ambient conditions.[1] ***Figure 10*** above presents results from that project. In 2016 a Carrier funded program cost-optimized the CCHP design and verified that performance and cost targets could be met for both heating and cooling. This design was further optimized and applied for this field trial.

3.0 PERFORMANCE OBJECTIVES

3.1 SUMMARY OF PERFORMANCE OBJECTIVES

The following table describes quantitative and qualitative performance objectives at a high level. Section 3.2 breaks down the performance objectives into finer detail.

Table 2. Performance Objectives.

Performance Objective	Metric	Data Requirements	Success Criteria	Results
Quantitative Performance Objectives				
Performance and Cost Avoidance	Annualized Energy Savings	Meter readings of power delivered to baseline and prototype systems; Price structure and calculation of cost avoidance by using the proposed technology	≥20% average annualized energy savings as tested in the field demonstrations.	Cost savings easily achieved with heating COP ~twice baseline. >30% savings calculated for all cold-climate zones. Cost premium $1200
Air Supply Temperature	ºF	Measure air supply temperature from the packaged rooftop unit; Optimize operation of ID blower to maintain desired SAT.	≥95ºF SAT maintained through controls of components, independent of load or ambient temperature.	Achieved using indoor air flow rate control. Demonstrated SAT typically >100F.
Direct Greenhouse Gas Emissions	Direct fossil fuel emissions, metric tons	Estimated release of greenhouse gases based on source of energy if technology had not been implemented	>20% reduction (indirect) in green-house gases over SOA heat pump systems corresponding to energy consumption reduction.	Easily achieved with COP improvement since electricity consumption corresponds to both cost and CO2 production.
Scalability Across DoD	%	Bin hour analysis and local weather data to calculate annual energy savings to determine value proposition	>56% of DoD installations reside in cold climate zones where CCHP technology can reduce dependence on fossil fuels.	Technology demonstrated can be applied to any rooftop size.
Qualitative Performance Objectives				
Ease of Use	Level of expertise/ skill of intended end user	Survey and/or interview results describing the difficulty of using the technology as designed	Exceeds expectations of key stakeholders with regards to maintenance and operation.	Standard Rooftop installation. No operational maintenance required for product.
Thermal Comfort	Feedback of occupants and operations staff	Survey and/or interview results describing the experience of the base personnel during the prototyping phase	Exceeds expectations of occupants with regards to thermal comfort during heating and cooling seasons	AJ. Ballard (MEARNG) reports improved customer satisfaction.

3.2 PERFORMANCE OBJECTIVES DESCRIPTIONS

The following sub-sections detail the quantitative and qualitative performance objectives highlighted in the Table 2.

1. Performance and Cost Avoidance

- Purpose: Quantify the energy consumed by the baseline and prototype systems to deliver capacity and comfort to the demonstration building during heating and cooling seasons, and quantify the cost savings associated with the improved performance. In addition, quantify cost avoidance that may be achieved through a) lower kW demand charge, b) lower hardware cost due to reduced or eliminated electrical heater requirements, and c) avoidance of oversized building wires, breakers, and associated installation costs.

- Metric: The difference in energy consumption and capacity delivered relative to the baseline. Associated electrical cost required/avoided by using the alternative systems to satisfy the building load and maintain thermal comfort. Difference in installed cost due to reduction in required kW.

- Data: Electrical power consumption data for the entire unit (fans, defrost, electric heat) in kW used. Data logging enabled the calculation of the integrated energy consumption needed for heat pump operation in kWh. Magnitude of kW demand spikes were identified. Cost avoidance was the difference between the integrated power required for the baseline and prototype.

- Analytical Methodology: Data extracted from the field demonstrations was used to develop an effective COP as a function of ambient temperature for both the baseline unit (tested in Phase I) and the CCHP (testing in Phase II). The effective COP included all transient effects, frost and defrost, and actual operating conditions, and was therefore lower than the steady-COPs measured in the lab. This performance function was then applied to various buildings and climate zones. TMY3 data was used for different locations. Since the two units (baseline and CCHP) were demonstrated in the same building zone, and each for nearly a year of operation, the two COP functions are directly comparable.

- Success Criteria: It was projected that the cold climate heat pump prototype would show at least a >20% annual energy savings over the baseline Carrier system. This takes into account the supplemental electrical heat, defrost operation, heating mode and cooling mode. Cost avoidance is primarily proportional to energy savings.

2. Air Supply Temperature

- Purpose: Quantify the supply air temperature delivered to the space in order to determine the extent of cold blow into the space during the heating season and verification of baseline performance in the cooling mode.

- Metric: The prototype system delivered greater than 95F supply air temperature in the heating mode over a range of outdoor air temperatures from 47F to 0F. The SOA system delivered ~95F at 47F and ~75F at 0F. Due to this limitation heat pumps have a very low penetration in cold climates due to the delivery of cold air into the space.

- Data: Calibrated dry bulb and dew-point temperatures were installed in both the supply and return air ducts. The outdoor air temperature and dew-point was also captured in the vicinity of the units being tested.
- Analytical Methodology: Data extracted from the field demonstration was used to 1) demonstrate the extent of cold blow for the baseline SOA system and 2) demonstrate that the supply air temperature was >95F over the entire heating mode.
- Success Criteria: >95F supply air temperature delivered to the space. This was also reaffirmed though the qualitative thermal comfort surveys of the occupants.

3. **Direct Greenhouse Gas Emissions (GHG)**

- Purpose: To calculate the amount of displaced greenhouse gases that corresponds to the use of a high-performance air-source heat pump in cold climates.
- Metric: Heat pumps that do not maintain capacity and efficiency at very low outdoor air temperatures require the use of supplemental heating methods, e.g. electric heat, to maintain thermal comfort in the occupied space. The kWh offset between the field demonstrated prototype and a state-of-the-art system is needed.
- Data: The power consumption data collected for performance objective 1 was used to calculate the energy savings associated with the use of a cold climate heat pump system. See performance object 1 for details.
- Analytical Methodology: The kWh saved was determined from energy savings from objective 1 above. Since the RTUs deployed used only electricity, the percent CO_2 savings is the same as the percent kWh savings. The total amount of CO_2 savings can be calculated from an estimate of product penetration, the value for US average CO_2 generation per kWh generated, and the value for average transmission losses.
- Success Criteria: Similar to Performance Objective 1, it was projected that the cold climate heat pump prototype would show at least a >20% annual energy savings over the baseline Carrier system which translates to >20% reduction of CO_2 GHG. This takes into account the supplemental electrical heat, defrost operation, heating mode and cooling mode.

4. **Scalability Across the DoD**

- Purpose: To develop a means to evaluate the applicability and payback of deploying air-source cold climate heat pumps relative to other sources of heating at DoD installations.
- Metric: Annual cost savings due to reduced energy consumption by switching to air-source cold climate heat pump technology.
- Data: The field trial demonstrated a nominal 10TR capacity CCHP system. The highest volume packaged rooftops currently installed at DoD bases ranges from 5TR to 12.5TR. The components deployed in the field trial unit are available to cover at least this size range. The technology is shown to work well over the targeted CCHP climate zones from 4A to 7.
- Analytical Methodology: Models developed for Performance Objective 1 were used to determine the annual cost savings of utilizing the prototype cold climate heat pump at several DoD installations in different geographic regions. The field trial site was located in ASHRAE climate zone 6A (humid and cold).

- Success Criteria: The proposal estimates indicated that 56% of DoD buildings are in regions (ASHREA climate zone 4 and higher) where air-source heat pumps could provide significant cost savings (>20% annual heating and cooling) for buildings currently using electric heat.

5. **Ease of Use**

- Purpose: To qualitatively assess the ease of installation and use of the prototype cold climate heat pump system with particular focus on integration with the building management control system and troubleshooting.

- Metric: No significant down-time or unscheduled maintenance or repairs relative to the state-of-the-art Carrier system. No significant issues related to the integration of the prototype controls with the building management control system.

- Data: Alan J. Ballard (Energy Director of the Maine National Guard) was the person who oversaw all contracting work for the Maine field trial unit. Various contractors involved included installation, controls and facility work. He provided feedback relating to installation and operation of the CCHP unit throughout the project.

- Analytical Methodology: UTRC addressed Mr. Ballard's concerns and used the prototype test for ongoing data collection and improvement, when possible.

- Success Criteria: No significant down-time or unscheduled maintenance or repairs relative to the state-of-the-art Carrier system. No significant issues related to the integration of the prototype controls with the building management control system.

6. **Thermal Comfort**

- Purpose: To qualitatively assess the performance of the system, namely air discharge temperature and avoidance of cold blow in peak winter operation, by occupants of the building.

- Metric: The prototype system delivered greater than 95F supply air temperature in the heating mode over a range of outdoor air temperatures from 47F to 0F. The SOA system delivered ~95F at 47F and ~75F at 0F. Due to this limitation heat pumps have a very low penetration in cold climates due to the delivery of cold air into the space.

- Data: A.J. Ballard (Energy Director of the Maine National Guard) conducted surveys of space occupants periodically to ensure thermal comfort in both cooling and heating seasons. This included areas of 1) temperature control, 2) humidity control, 3) air velocity or cold-blow, and 4) general comfort.

- Analytical Methodology: UTRC reviewed the results of the surveys as reported by Mr. Ballard, and addressed concerns and used the prototype test for ongoing data collection and improvement, when possible.

- Success Criteria: Occupants report that they are comfortable. Data shows that the zone temperature is held at set point. No cold blow issues during heating reported.

4.0 FACILITY/SITE DESCRIPTION

The two field trial units were tested at 1) office building 39 at Camp Keyes, Headquarters of the Maine Army National Guard, in Augusta, ME and 2) at the psychrometric facility at UTRC, E. Hartford, CT.

Camp Keyes, Maine Army National Guard, Augusta, ME 04333,
Mr. Alan J. Ballard CEM, Energy Director, Phone 207-430-5679,
E-mail: alan.j.ballard.hfg@mail.mil

UTRC, 411 Silver Ln, E. Hartford, CT 06118
Dr. Frederick Cogswell, Principal Investigator, Phone 860-610-1688,
E-mail: cogswefj@utrc.utc.com

4.1 DEMONSTRATION FACILITY/SITE LOCATION AND OPERATIONS

The Augusta site is part of the Maine National Guard. It is located at Camp Keyes at the Augusta Airport. This field trial site and UTRC's psychrometric facility are discussed below.

Unit 1: Maine Army National Guard – Camp Keyes

Camp Keyes serves as the headquarters for the Maine Army National Guard of which Building 39 was utilized for the demonstration (Figures 12, 13 and 14). Building 39 is located at 194 Winthrop St., Augusta, ME 04330. Prior to this project two packaged rooftop units served Building 39's cooling and heating needs (***Figure 14***). Hot water, produced by natural gas fueled condensing boilers, was sent to duct heating coils to heat the air. Building 39 is a small office building with a nominal heating load of approximately 116 kBtuh. The capacity of this building was ideal for the nominal 10TR (120kBtuh) demonstration units that were installed and tested. The building has a BMCS (Honeywell Controller Model 64385ILC) which controlled the existing HVAC systems. It was used to control the new Phase I RTU, and to monitor both the Phase I and Phase II units.

Figure 12. Building 39 at MEARNG Camp Keyes Which Was the Primary Demonstration Site.

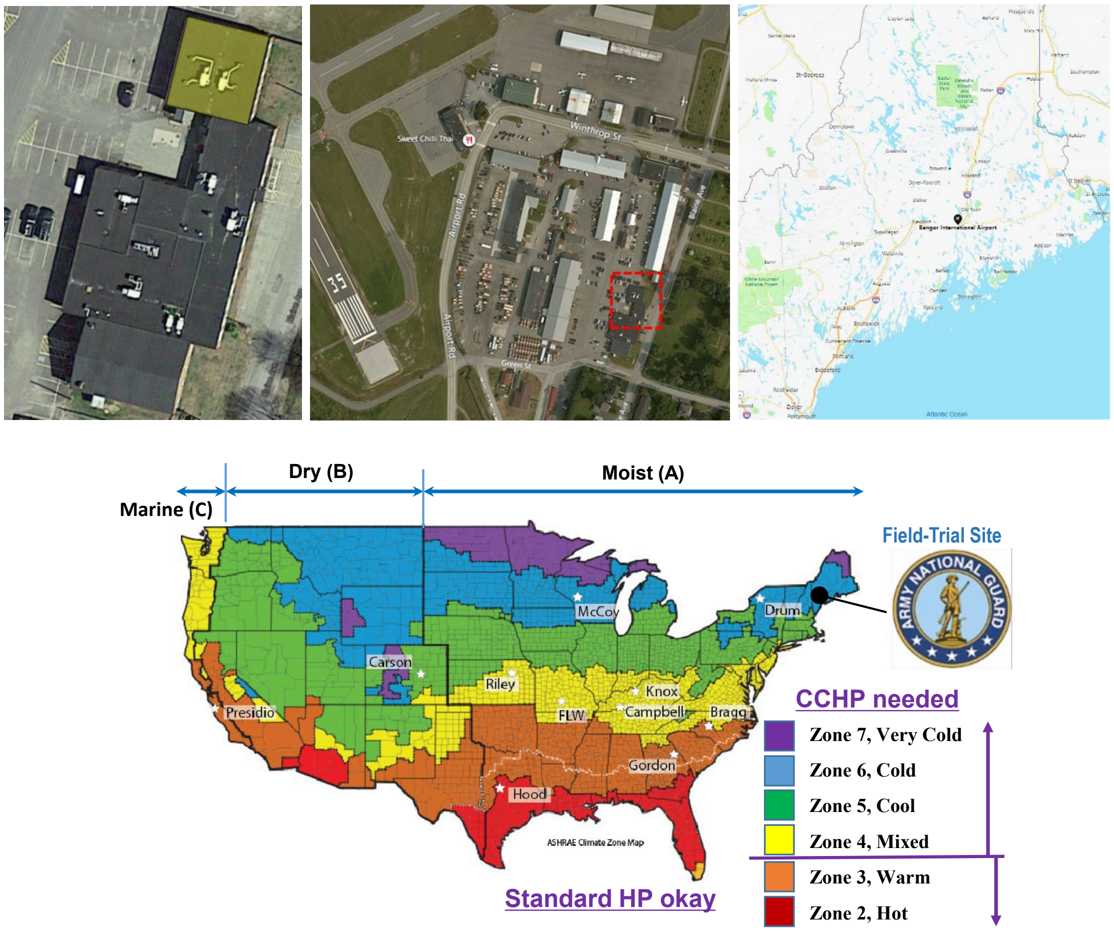


Figure 13. Location Camp Keys Field Trial Site.

Top-Left: Ariel view of Building 39 (shaded) that was used as the main demonstration site; Top-Center: view showing location relative to Augusta State Airport; Top-Right: Map of Maine showing Bangor airport and Camp Keys location; Bottom: US map showing ASHRAE climate zones and location of site in zone 6A.

Figure 14. Conceptual Drawing of Demonstration Scope to Improve Efficiency of Building 39.

(Removal of 2 existing air-handling units and 4 condensing units, and replacing with one centralized packaged roof-top system)

Unit 2: UTRC Psychrometric Facility

The second unit was tested in UTRC's psychrometric facility (***Figure 15***) which has the following capabilities:

- Indoor room: 45 to 145F, 30 to 95%RH,
- Outdoor room: -10 to 145F, 20 to 95%RH,
- Up to 20 TR (240kbtuh) heating and/or cooling.

The second CCHP unit can be seen installed in the outdoor chamber (***Figure 15***-right), where both heating and cooling performance can be tested at any combination of outdoor and indoor air conditions.

Figure 15. UTRC Psychrometric Facility and Unit#2.

Left: Outdoor chamber front, indoor chamber rear. Right: CCHP Unit#2 in outdoor chamber, supply and return ducts connected on right to Indoor chamber.

4.2 GENERAL FACILITY/SITE SELECTION CRITERIA

The field trial site at August Maine is located in ASHRAE climate zone 6A (humid and cold). ***Figure 16*** shows the TMY3 data from that location. This climate zone has a significant number of hours below 30F where traditional heat pumps do not perform well. In addition, since this site is humid, the detrimental effects of frosting and defrosting are well represented.

The Camp Keyes site is an office building which is well representative of the DoD building stock. The heating and cooling set points vary based on an occupancy schedule. Set-back is used at night and during the weekends.

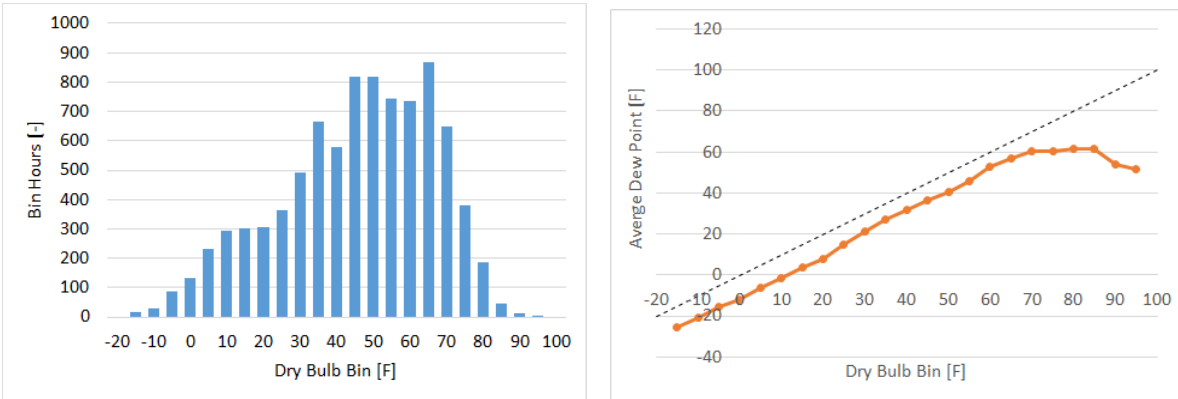


Figure 16. TMY3 Data for Augusta Maine Airport.

The second field trial unit was tested in UTRC's Psychrometric Laboratory where various ambient conditions (temperature and humidity) can be controlled. This unit was used to advance the design of the CCHP. Specific areas of advancement included:

- Alternative component designs and control features of the novel components deployed in the CCHP system in order to reduce product cost and/or improve performance.
- Improved defrost logic (as compared to current field performance) and the associated control architecture in order to increase overall system efficiency and/or reduce need for electrical heating.
- And, general improvement of the performance of the entire control framework based on evaluation of test data from the field.

Data from the winter heating season at Camp Keys was analyzed and code improvements were made and tested in UTRC's psychrometric labs, and then deployed at the Camp Keys site for the fall heating season.

5.0 TEST DESIGN

5.1 CONCEPTUAL TEST DESIGN

Traditional SOA heat pumps do not perform well in cold climates due to:

1. Inadequate heating capacity when sized properly for cooling mode, requiring additional electric heat,
2. Poor COP due to large temperature difference between outdoor and indoor temperatures,
3. Poor COP due to the requirement to continuously defrost during which time it is effectively cooling the zone it is supposed to be heating, and
4. Poor comfort due to low supply air temperatures resulting in draftiness.

The Cold Climate Heat Pump (CCHP) developed by UTRC and demonstrated during this project solves these problems using two key enabling technologies, high-efficiency high-lift compression and system level design optimization for cold climates.

The independent variables for this project are which RTU was used to meet the building loads. During Phase I a SOA carrier heat pump unit was used at the Camp Keyes site, and a full year of data was obtained. During Phase II this unit was replaced by the new UTRC CCHP and another year of data was obtained.

The dependent variables include the building load (required capacity in both heating and cooling modes) and the unit power draw. These variables are affected by:

- Ambient temperature, solar load and number of occupants,
- Rate of frost build up on the outdoor evaporator (heating only),
- Latent load (cooling only),
- Transient On/Off cycling,
- The use of zone occupancy scheduling, and
- Possibly the set-points of the neighboring zone in the same building.

The goal was to use the data obtained to develop:

A. A relation for COP as a function of ambient temperature for both the baseline and CCHP units, and
B. An average load curve for the particular building test site, also as a function of ambient temperature.

These relations can then be applied to TMY data at various sites to determine the expected energy savings provided by the new technology.

Other dependent variables include the supply air temperature (which affects occupant comfort) and occupant survey results.

The controlled variables between Phase I and Phase II at Camp Keyes were that the building zone being served by the two units (baseline and CCHP) were the same; with the same BMCS system control points and zone temperatures, and with the same duct distribution system. Also, by running each unit over multiple months of heating and cooling operation, normal variations in load and therefore performance should be captured in the analysis.

The test phases are described in Section 5.4, Operational Testing.

5.2 BASELINE CHARACTERIZATION

Figure 17 shows the Carrier Heat Pump unit model 50HCQD12 installed on the roof of Camp Keyes Building 39. This unit is a state-of-the-art high tier unit currently sold for heat-pump applications and provided the "baseline" data for this study. The supply air duct is visible on the right-front, and the return air duct is behind it. The supply air sensor station (temperature, humidity and flow) is visible on the first straight section of the duct. The 50HCQD12 has nominal 10TR cooling capacity. It has two compressors and two separate refrigeration circuits. In cooling mode the two circuits were run independently as the load required. In heating mode they were run together. The unit was supplied with a 36kW electric heater which was used for additional heating as well as during the defrost cycles.

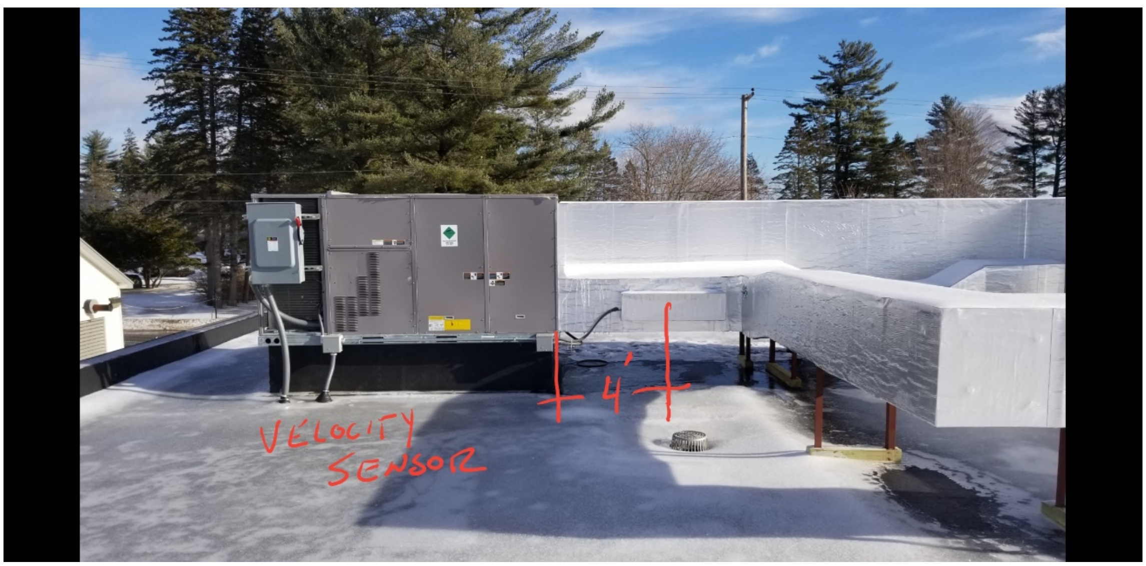


Figure 17. Carrier Heat Pump 50HCQD12 Installed at Camp Keyes.

Supply Air flow station visible on duct to right of unit.

The existing Camp Keyes BMCS (Honeywell Model 64385ILC) was used to log 1-minute data from the RTU and applicable zone. The data set is shown below in Section 5.5, Table 4. ***Figure 18*** shows the Siemens power meter used for the data in rows 31 through 37.

Figure 18. Siemens Power Meter.

5.3 DESIGN AND LAYOUT OF TECHNOLOGY COMPONENTS

The Phase I testing at Camp Keyes was with a currently available SOA RTU. It was controlled via the existing BMCS system. ***Figure 19*** shows the control diagram used.

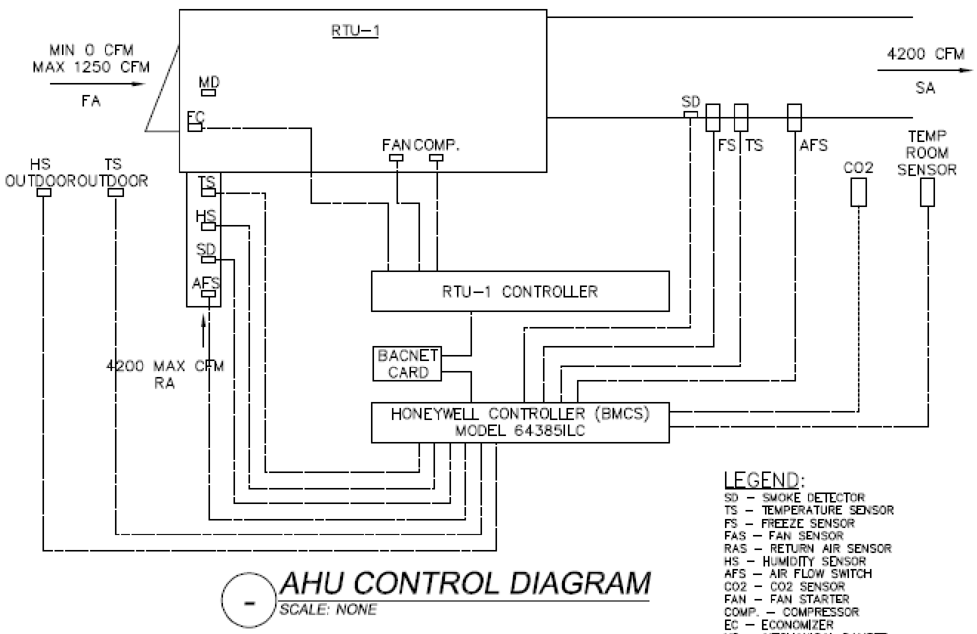


Figure 19. Control Diagram for BMCS System, Phase I.

The Phase II CCHP developed by UTRC was designed to fit into the same packaging as currently sold RTUs, and installation of the unit was identical. Most components within the CCHP are commercially available today. The field-trial controls were deployed in a special real-time simulation and testing machine (***Figure 20***) so that modifications could be made during Phase II if necessary. Upon product development the code may be transferred to standard product control boards.

Figure 20. Phase II Control Instrumentation.

Left: SpeedGoat, mobile real-time target machine, Right: control panel including SpeedGoat, relays, transformers and DC power supplies.

The Phase II controls were developed in-house at UTRC. They drew on Carrier commercial RTU controls, but modified them as necessary for the new product. Some examples of logic that needed to be modified are the compressor staging logic and indoor fan speed logic.

Another example is frost protection. In the existing product Carrier uses temperature sensors on the outdoor coil to determine when frosting may occur. It then performs a defrost cycle after a specified time period has elapsed. The customer may change this time period. For Phase I it was set at 1 hour. A defrost cycle consists of switching the unit from heating mode to cooling mode (activating the reversing valve) until the outdoor coil sensor indicates that defrost is complete, or until a maximum time is reached. During this time, in order to prevent cold-blow (since the unit is now running in cooling mode) the electric heaters are activated. During the first winter season for Phase II at the Camp Keyes site, the standard Carrier defrost logic was deployed in the CCHP. A new logic was designed and verified at the UTRC psychrometric facility during the summer of 2019, and this logic was then deployed at the Maine site for the second heating season. The goal was to minimize unnecessary defrost cycles, minimize the time it takes to defrost, and potentially eliminate the need for electric heating during defrost.

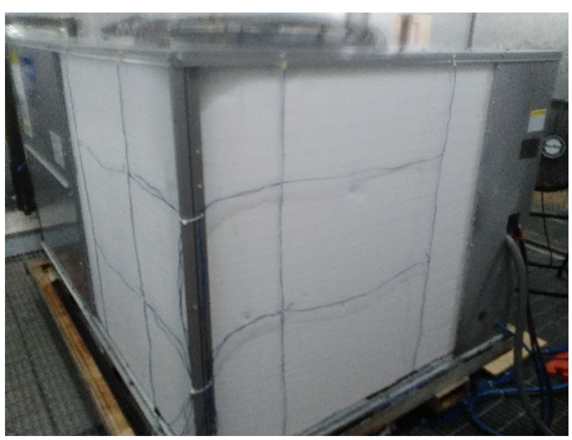


Figure 21. Frosting Test at UTRC Psychrometric Laboratory.

5.4 OPERATIONAL TESTING

There were three primary phases of field testing as shown in ***Figure 22***. Phase I consisted of benchmark testing of a SOA Carrier heat pump at the Camp Keyes Field site. Phase II testing (with the UTRC prototype CCHP) was done at both the Camp Keyes site and at the UTRC psychrometric laboratory. Phase III consisted of replacement of the field-trial unit with a high-efficiency Carrier gas-fired unit at the Camp Keys site. Initial control development was done at UTRC overlapping the Phase I testing in the field. Cost and performance analysis was ongoing throughout the project. More details of each phase are discussed below.

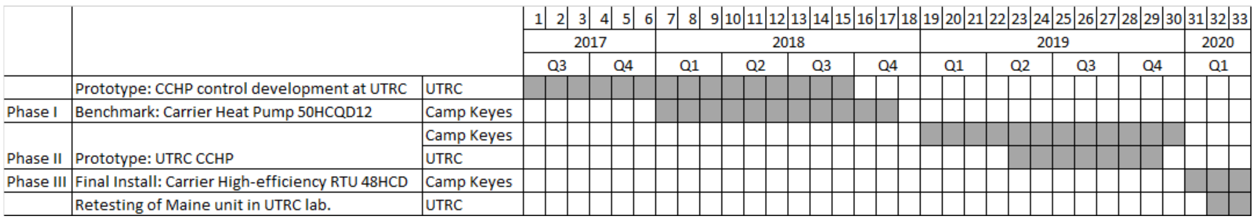


Figure 22. Gantt Chart for Testing Activities.

<u>Phase I:</u> Consisted of the demolition of existing legacy system and retrofitting with a 120 kBtuh cooling capacity and 116 kBtuh heating capacity (47F rating) SOA Carrier heat pump for baseline testing. The controls of this Carrier system were through the current building management system. Standard product commissioning was performed after the installation was complete.

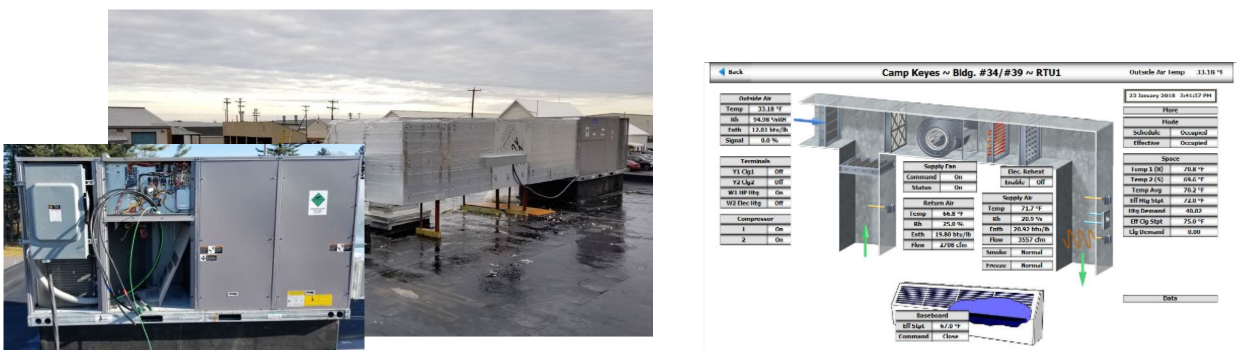


Figure 23. Phase I Baseline Unit and Controls.

<u>Phase II:</u> Consisted of 1) replacing the Carrier SOA system in Camp Keyes with an UTRC prototype CCHP system with a similar cooling and heating capacity, and 2) deploying a second CCHP system in the UTRC psychrometric facility to advance the product with respect to components and controls. Each activity is described in greater detail below.

<u>1st CCHP Unit</u>: There was no change in the nominal capacity rating between the two Camp Keyes units. Both the baseline unit and the CCHP were nominally 10TR (35kW) cooling units. The baseline unit was able to maintain 35kW of electric heat over all outdoor temperatures by utilizing supplemental electric heat, while the CCHP was designed to provide this capacity in heat pump mode alone. The objective was to demonstrate the capacity and system efficiency improvement at the colder outdoor ambient temperatures. The controls of the CCHP system were through proprietary controls deployed at the site through a prototype system controller.

A separate new commercial hard-wired cable internet connection was installed so that UTRC could communicate with the RTU controller without going through the DoD installation network. This eliminated all cyber-security concerns and associated activities. Standard product commissioning was performed after installation was complete and was then followed by monitoring by UTRC to verify that control parameters were properly set for uninterrupted operation.

2nd CCHP Unit: The second CCHP unit was tested in the UTRC psychrometric facility. The goal was to further optimize the system with respect to performance and cost. Alternate component designs and control features of the novel components deployed in the system were examined. The design of the key enabling CCHP component technologies were evaluated through modeling and analysis of testing results. New improved defrost logic (as compared to current field performance) and associated control and hardware architecture was developed and then verified on the Camp Keyes unit for the fall heating season.

Figure 24. Phase II Advanced CCHP Unit and Controls.

Phase III: Consisted of replacing the UTRC prototype system at Camp Keyes with a 90 kBtuh (7.5TR) cooling capacity Carrier SOA system with natural gas fired heating that will remain in place after this project. Standard product commissioning was performed. Note that field trial data indicated that only 7.5TR of cooling was required for this site.

Figure 25. Phase III 7.5TR Cooling and Gas Fired Heating Unit.

5.5 SAMPLING PROTOCOL

Table 3 below shows sources of data throughout this project. UTRC was responsible for the LabView and Control data, while the site contractors were responsible for the BMCS data. During control development at the UTRC psychrometric facility, data was recorded both through a facility LabView data acquisition system at 1 sample per second, as well as through the SpeedGoat control system at a rate of 10 samples per second. The BMCS data sets are different for each of the three phases. During Phase I the BMCS system was controlling the RTU and the recorded data includes control variables such as the heating and cooling demand levels, whereas during Phase II UTRC supplied the module that controlled the unit, and the BMCS system provided building data as shown in Table 5. Phase III consists of standard product BMCS data and controls.

Table 3. Data Sources for Each Phase and Location.

	UTRC Lab	Camp Keyes
Control Development	LabView 1s, Control 0.1s	
Phase I: Benchmark		BMCS 1min
Phase II: CCHP	Control 0.1s	BMCS 1min & Control 0.1s
Phase III: Gas		BMCS 1min

The BMCS data was stored in the existing systems which have the capacity to hold many months' worth of 1-minute data. The CCHP control module stored data locally which was periodically transferred first to a local laptop computer, and then through a dedicated Ethernet connection which could be accessed from UTRC via Windows Remote Desktop Connection. Data transfer was done at least once per month to avoid data loss.

The Phase I and Phase II data sets are presented below:

- Table 4: Phase I BMCS
- Table 5: Phase II BMCS
- Table 6: Phase II RTU Controls (note: this data set is considered UTC/Carrier proprietary and has been removed from this public document.)

Table 4. Phase I 1-minute Data Logged by BMCS System, Camp Keyes.

	Variable type	Variable Name	Comments
1		?timestamp	
2	Zone data	AvgSpcTemp (°F)	Combined zone temperature, control point.
3	Control variables	EffHtgStpt (°F)	Typically set to 72°F during heating season.
4		EffClgStpt (°F)	Typically set to 73°F during cooling season, 75°F during heating.
5		Mode_0Htg_100Clg	
6		FanCmd	Fan request ON. [0/100]
7		FanSts	Indicates when fan is on. [0/100]
8		Comp1Sts	Indicates when compressor 1 is ON. [0/100]
9		Comp2Sts	Indicates when compressor 2 is ON. [0/100]
10		HtgDmnd	Control variable that determines heating mode [0-100]
11		HtPmpHEAT	Indicates that HP is requested [0/100]
12		ElecHEAT	Indicates that Electric heat is requested [0/100]
13		Baseboard	Indicates that baseboard heat is requested [0/100]
14		ClgDmnd	Control variable that determines cooling mode [0-100]
15		HtPmpCOOL1	
16		HtPmpCOOL2	
17	Added measurement stations near the RTU	OATemp (°F)	Enthalpy is calculated from temp and RH, Dew point is calculated from temp and RH.
18		OAHumidity (%RH)	
19		OAEnthalpy (BTU/lb)	
20		OADewPoint (°F)	
21		RATemp (°F)	Station on return air duct just before RTU, Enthalpy is calculated from temp and RH.
22		RAHumidity (%)	
23		RAEnthalpy (btu/lb)	
24		RAFlow (cfm)	
25		SATemp (°F)	Station in supply air duct just after RTU, Enthalpy is calculated from temp and RH.
26		SAHumidity (%)	
27		SAEnthalpy (btu/lb)	
28		SAFlow (cfm)	
29	Zone data	CO2_1 (ppm)	Ventilation-only may be requested by the BMS if the zone CO2 exceeds a preset value.
30		CO2_2 (ppm)	
31	Power	kW	Values measured and calculated from the Siemens power analyzer that feeds 230Vac to the RTU
32		kWh	
33		Volts (V)	
34		Amps (A)	
35		kVA (kVA)	
36		kVAh	
37		kVAR (kvar)	

Table 5. Phase II 1-minute Data Logged by BMCS System, Camp Keyes.

	Variable type	Variable Name	Comments
1		?timestamp	
2	Zone data	AvgSpcTemp (°F)	Combined zone temperature, control point.
3	Control variables	EffHtgStpt (°F)	Typically set to 72°F during occupied heating season.
4		EffClgStpt (°F)	Typically set to 73°F during cooling season, 75°F during heating.
5	Added measurement stations near the RTU	OATemp (°F)	Enthalpy is calculated from temp and RH, Dew point is calculated from temp and RH.
6		OAHumidity (%RH)	
7		OAEnthalpy (BTU/lb)	
8		OADewPoint (°F)	
9		RATemp (°F)	Station on return air duct just before RTU, Enthalpy is calculated from temp and RH.
10		RAHumidity (%)	
11		RAEnthalpy (btu/lb)	
12		RAFlow (cfm)	
13		SATemp (°F)	Station in supply air duct just after RTU, Enthalpy is calculated from temp and RH.
14		SAHumidity (%)	
15		SAEnthalpy (btu/lb)	
16		SAFlow (cfm)	
17	Zone data	CO2_1 (ppm)	
18		CO2_2 (ppm)	
19	Power	kW	Values measured and calculated from the Siemens power analyzer that feeds 230Vac to the RTU
20		kWh	
21		Volts (V)	
22		Amps (A)	
23		kVA (kVA)	
24		kVAh	
25		kVAR (kvar)	

Table 6. Phase II 10Hz Data Logged by RTU Controller, Both Sites.

Note: this table is considered UTC/Carrier proprietary, and has been removed from this public document. It included control decision variables, actuators and sensed values as well as other installed sensors that were used for monitoring and diagnostics.

5.6 EQUIPMENT CALIBRATION AND DATA QUALITY ISSUES

The data sets for each phase of testing were presented in Section 5.5. The key data for determining the units' performance are values from the measurement stations (BMCS data set) installed next to the RTUs, and the power meter. Other data are recorded for diagnostic purposes but do not directly enter the capacity and COP calculations. The supply air flow measurements were calibrated during the commissioning process by measuring and summing the flow at each of the supply registers for different indoor fan speeds. It should be noted that the humidity measurements have little impact on the heating mode calculations but have significant affect the cooling mode calculations.

For Phase II, in addition to the BMCS data, there was also the CCHP Control data which includes air temperature sensors in the supply and return ducts, and in the outdoor air damper. These sensors were used to validate the BMCS temperatures (Supply Air temperature, Return Air Temperature, and Outdoor Air Temperature). Since they were bead-style thermocouples with no protective sheath, they had a quicker response than the flow station temperature sensors.

A sampling frequency of 1 minute was used for the BMCS data, which was sufficient to capture most transients. A sampling frequency of faster than 1 second was used for the Phase II control data which was necessary to diagnose control related transients/decisions, but the key performance parameters (temperature and flow) do not change this quickly.

The nature of building load profiles is that they are not solely a function of outdoor air temperature, but vary significantly due to various factors such:

- Solar Radiation,
- Humidity (cooling mode),
- Office/building activity in the zone served by the test RTU,
- Additional transient effects brought on by set-point scheduling, and
- Activity and set point changes in the neighboring zones served by other HVAC equipment.

For example, a zone set-point change to a lower value may lead to many hours of non-operation as the zone temperature drifts down. In order to reduce these effects in the data processing, extensive data is taken over many months and averaged. The process used was:

- Integrate 4-hr averages of capacity and power draw as a function of average outdoor temperature over the 4-hr period.
- Sum up all the values for each temperature bin (5ºF bins; +/- 2.5ºF around each nominal value). Determine average load and power for each temperature bin.
- Determine COP = bin-averaged capacity / bin-averaged power as a function of outdoor temperature.
- Apply these relations to TMY data to get expected energy savings.

It should be noted that it is not proper to analyze the occupied data separately from the un-occupied. For example, if there is a large drop in set point in heating mode, then there are many hours with no operation at all as the load is ultimately shifted to the occupied time during setback recovery. For this project all data was averaged together.

5.7 SAMPLING RESULTS

5.7.1 Measurements and Equations

The primary measurements used for both Phase I and Phase II to determine the instantaneous capacity are from the two flow stations at the supply and return ducts next to the units; each measured temperature, humidity and volumetric flow.

The COP is defined as the cooling or heating capacity divided by the electrical power into the unit. The power was determined directly from power meter measurements. The capacity is determined from the temperature, humidity and flow stations located next to the RTU. The enthalpy and density of the air can be calculated directly from the measured temperature and humidity via psychrometric functions:

$$\rho = f(T,\ RH\ or\ Tdew) \quad \text{[kg-da/m^3]}$$

$$h = f(T,\ RH\ or\ Tdew)\ \text{[kJ/kg-da]}$$

[Note: it is standard for psychrometric calculations to be in terms of mass of dry-air, "da", rather than include the mass of the water vapor.] The density together with the measured volumetric flows determine the mass flow,

$$\dot{m} = CFM_{bms}/\rho \ \ \text{[kg/s]}$$

and the outdoor air flow (ventilation air) was determined from the difference of the supply air and return air.

$$\dot{m}_{OA} = \dot{m}_{SA} - \dot{m}_{RA}$$

where *SA* is supply air, *RA* return air, and *OA* outdoor air.

The instantaneous heating (positive) or cooling (negative) capacity is:

$$\dot{Q} = \dot{m}_{SA}h_{SA} - \dot{m}_{RA}h_{RA} - \dot{m}_{OA}h_{OA} \ \ \text{[kW]}$$

The outdoor air humidity and temperature was provided by the BMCS system, but as is discussed below was found to be inaccurate especially during cooling season.

- Therefore, outdoor air temperature and dew point data were downloaded from the next-door August Airport, and used instead of the BCMS data in the analysis.

Of the three values measured from each flow station (temperature, humidity, and flow), the humidity was perhaps the least accurate, and has the longest time constant to respond to changes. Yet for heating, there should be no change in net water flow through unit (and therefore neither load associated with vaporizing nor condensing water). It was found that if humidity was used in the heating calculations, then there was a significant implied load due to latent effects, which could not be real.

❖ For this reason all heating calculations were done assuming sensible only enthalpy changes.

As mentioned above, during the Phase II testing the CCHP had many additional sensors. The Supply air and return air temperature of the flow-stations were validated against the CCHP sensors and found to be accurate. The flow readings were calibrated by contractors on site after each installation.

The instantaneous COP is:

$$COP = Q/P$$

Where the capacity *Q* and power *P* are both in kW.

It should be noted that the instantaneous COP varies greatly during operation, especially during ON/OFF and defrost cycles, and that the average of the instantaneous COP over a time period is not equal to the average COP over that same time period. The proper averaged COP is defined as

$$COP = \frac{\int Q(t)dt}{\int P(t)dt}$$

A suitable time period for integration is one that is long enough so that on/off cycle variations are smoothed out, but the outdoor air temperature has not changed significantly. For the purpose of this study 4-hr integration was used.

5.7.2 Presentation of data

The baseline unit was installed in January of 2018 and removed the first week of December 2018. BMCS data was collected continuously through most of this time period. ***Figure 26*** shows the OAT (green) and calculated capacity for heating (orange) and cooling (blue) seasons over the year. The capacity shown is averaged over 4hours; therefore, there are 6 data points for each day. Shoulder seasons (where there is little load, and there could be heating and cooling within a single day) are not shown. It can be seen that the baseline (Phase I) unit data covers most of 2018. The peak 4hr-averaged capacity is about 40kW for heating and 22kW for cooling.

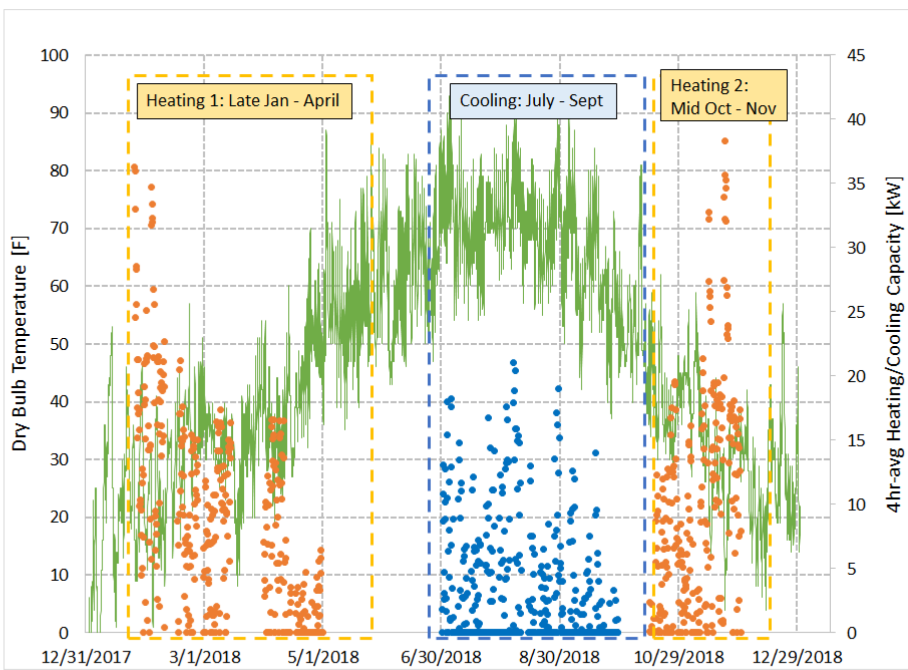


Figure 26. Processed Phase I Data Showing Ambient Temperature and 4-hr Averaged Capacity.

The Phase II CCHP unit was installed and operational in January of 2019 and was removed the first week of December 2019. Both BMCS and higher resolution SpeedGoat (unit real time controller) data was collected continuously through most of this time period. ***Figure 27*** shows the OAT and calculated capacity for heating and cooling seasons over the year.

The capacity shown is averaged over 4hours; therefore, there are 6 data points for each day. The peak 4hr-averaged capacity is about 41kW for heating and 17.5kW for cooling.

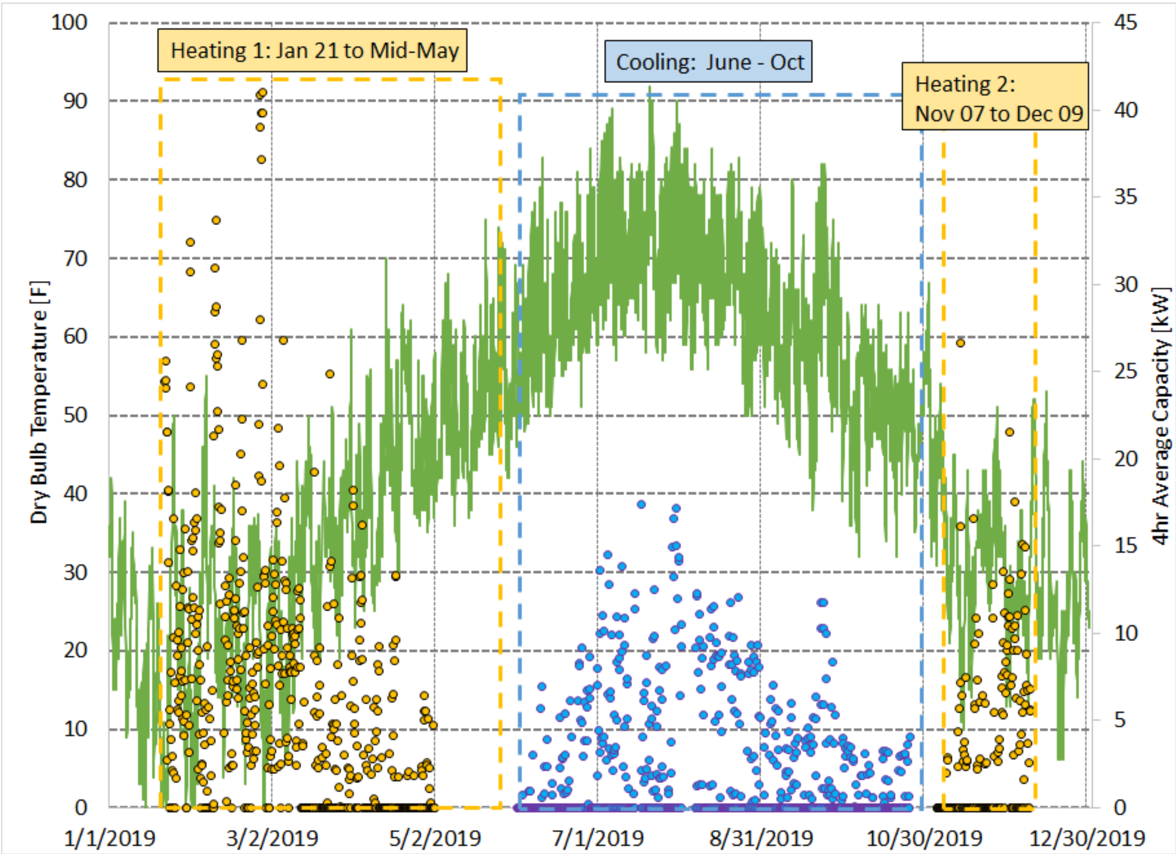


Figure 27. Processed Phase II Data Showing Ambient Temperature and 4-hr Averaged Capacity.

Comparing Phase I to Phase II, it can be seen that ambient temperatures between 2018 and 2019 were similar, as were the peak 4-hr loads. However, as will be shown below, the averaged heat load over the year for a given temperature bin was lower for 2019, which may be due to differences in set points for the neighboring zone in the building. In Section 6.0 these values are summed over temperature bins (sum of Q and sum of P separately), and then divided to find average COP as a function of temperature. Using these COP values annual capacity and power draw can be determined for different locations using TMY3 data. The difference in annual power draw directly relates to savings in cost and CO2 generation.

The same data as shown above can be plotted with outdoor temperature on the x-axis and capacity and power on the y-axis. ***Figure 28*** shows the results for heating, and ***Figure 29*** for cooling. As shown ***Figure 27*** above, the CCHP "*Heating1*" data spans from January to May 2019, and "*Heating2*" includes November and December 2019.

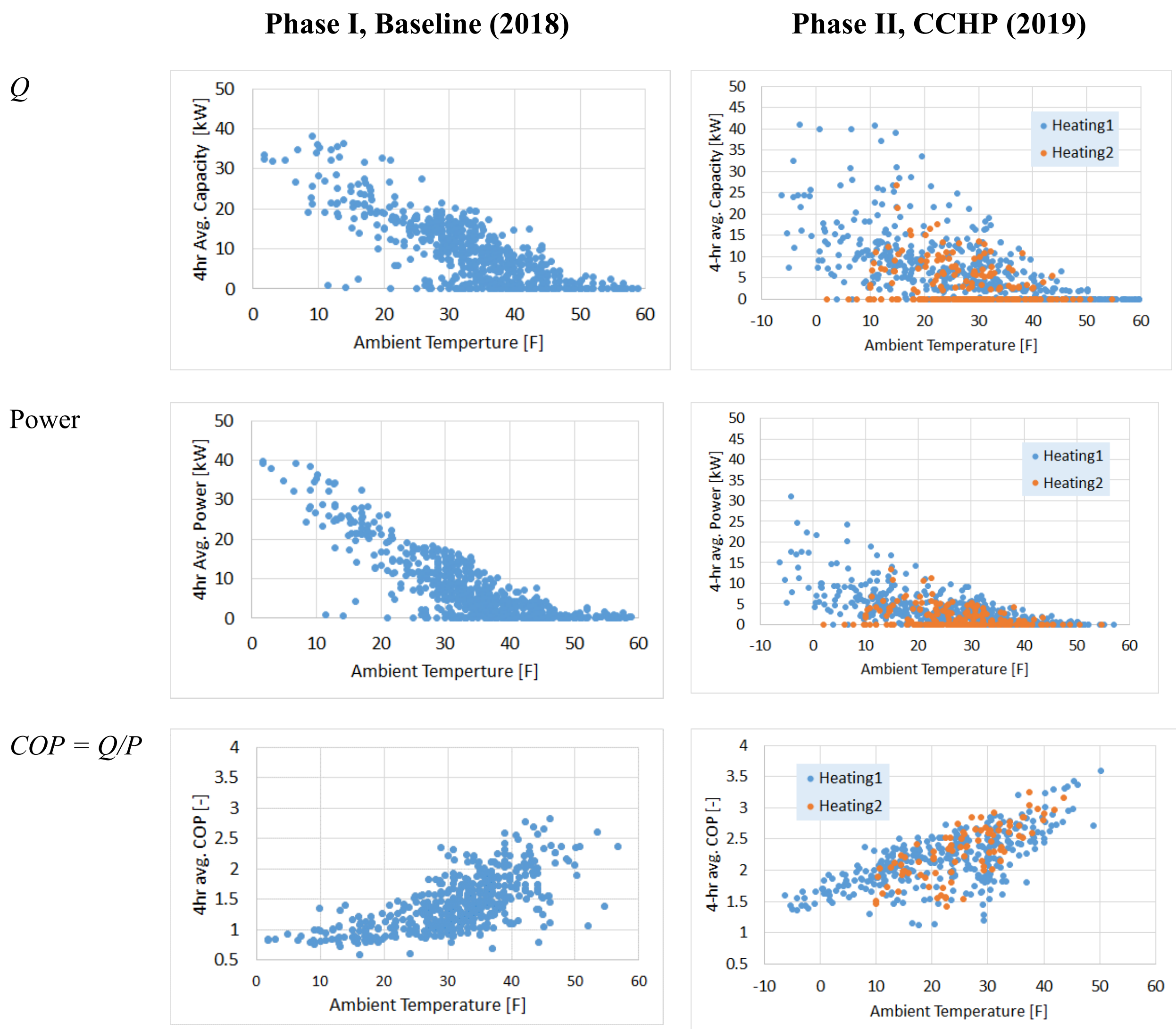


Figure 28. 4-hr Averaged Capacity (Q), Electrical Power (P), and COP for Heating Mode.

For heating mode, for both the baseline and CCHP units, the power and capacity generally increase with decreasing temperature, and there is little or no load above 50F. Both have a large scatter at a given temperature, but the CCHP data has more scatter and is on average lower. The capacity data can be compared to the "typical" commercial building plot show back in ***Figure 11***. A certain amount of scatter is expected and can be caused by zone-set-back, frosting/defrosting and cycling On/Off, but the generally lower values for Phase II were unexpected. The peak values, however, are similar at about 40kW = 11.4TR. For the baseline unit the COP can be very low, even less than one. This was due to the amount of frosting/defrosting occurring within the 4-hr average. For the baseline unit there was so little capacity provided by the heat pump that any frosting/defrosting can wipe out the advantages of even having a heat pump.

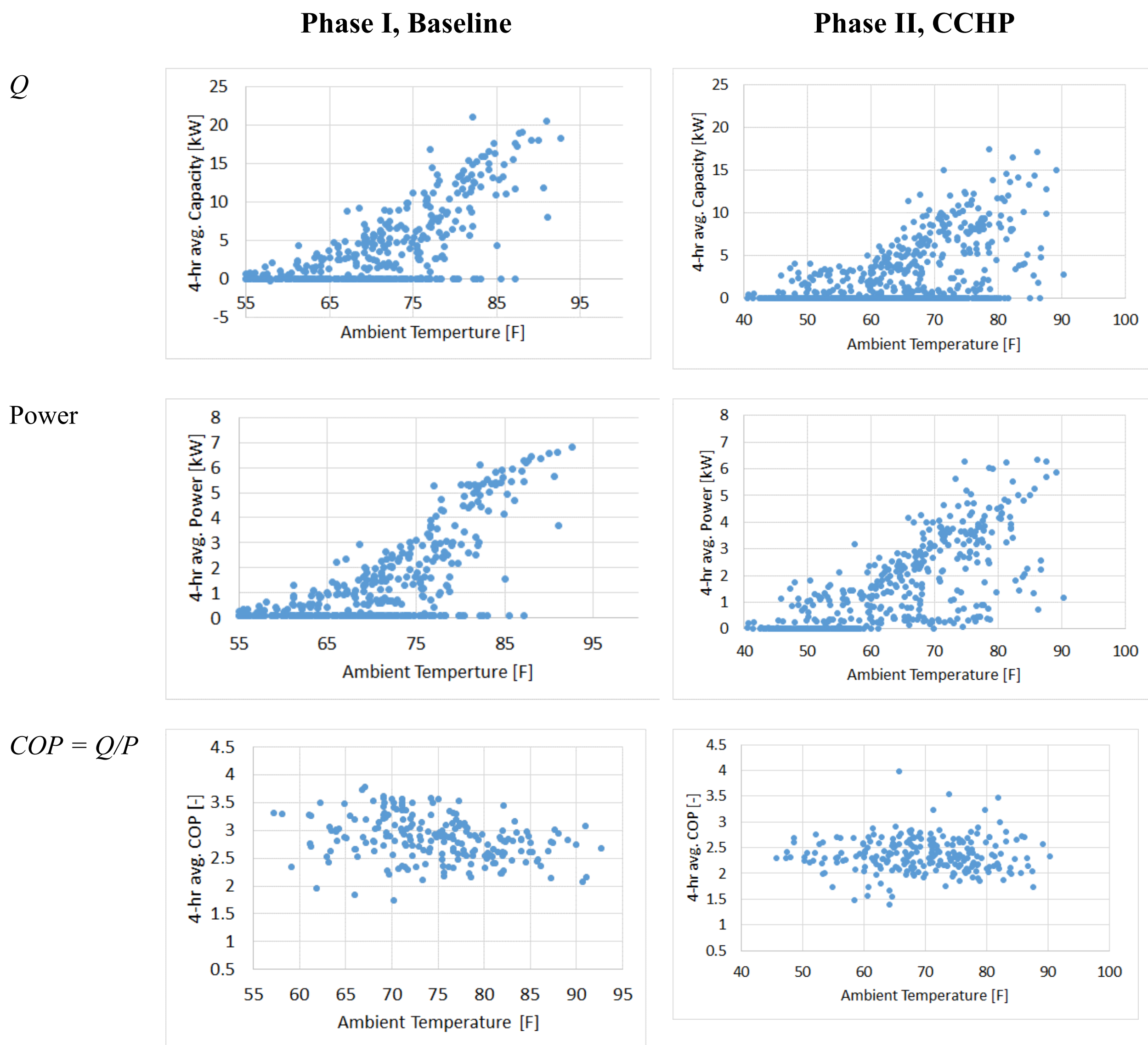


Figure 29. 4-hr Averaged Capacity (Q), Electrical Power (P), and COP for Cooling Mode.

For cooling mode, and for each unit, the power and capacity generally increase with increasing temperature. Again, there is significant scatter at a given temperature with more scatter for Phase II. The COP for both units was relatively flat with temperature, with the baseline being slightly higher. The peak cooling load seen for either unit was 21kW = 6TR which is about half the peak heating load. This demonstrates why sizing a traditional unit based on the required cooling load comes up way short for heating capacity in a cold climate. The final thing to note is that there was significant cooling load down to 45F outdoor temperature only for Phase II (CCHP). As mentioned above, this was likely due to differing zone temperature settings in the neighboring zones (see Section 8.2)

5.7.3 A Look at Transient Data

Some typical heating transient performance data is presented below in the following figures. ***Figure 30*** shows the baseline unit operating on November 14th 2018 when the outdoor temperature was about 20F. The averages over this day were:

- OAT_avg = 20.1F
- *Q*_avg = 28.4kW
- Pow_avg = 21.9kW
- COP_avg = 1.3

The green line in the plot is power to the unit. The compressors and fans take <10kW. The spikes up over 40kW occur when the 36kW electric heater is turned on. The unit runs continuously, and the electric heater was on for 40% of the time. The supply air temperature (red) spikes up over 100F whenever the heater is on, but is often less than 80F when the electric heater is off. For the second half of this plot it appears to be doing virtually no capacity without electric heat. Also worth noting is that with electric heat the return air temperature also rises above the zone temperature indicating thermal stratification. For this site the supply and return vents were in the ceiling. Defrost mode occurred 18 times during this period (grey line = 100) for a total of 140 minutes. An individual defrost cycle is limited to maximum of 10 minutes; half of these reached this limit.

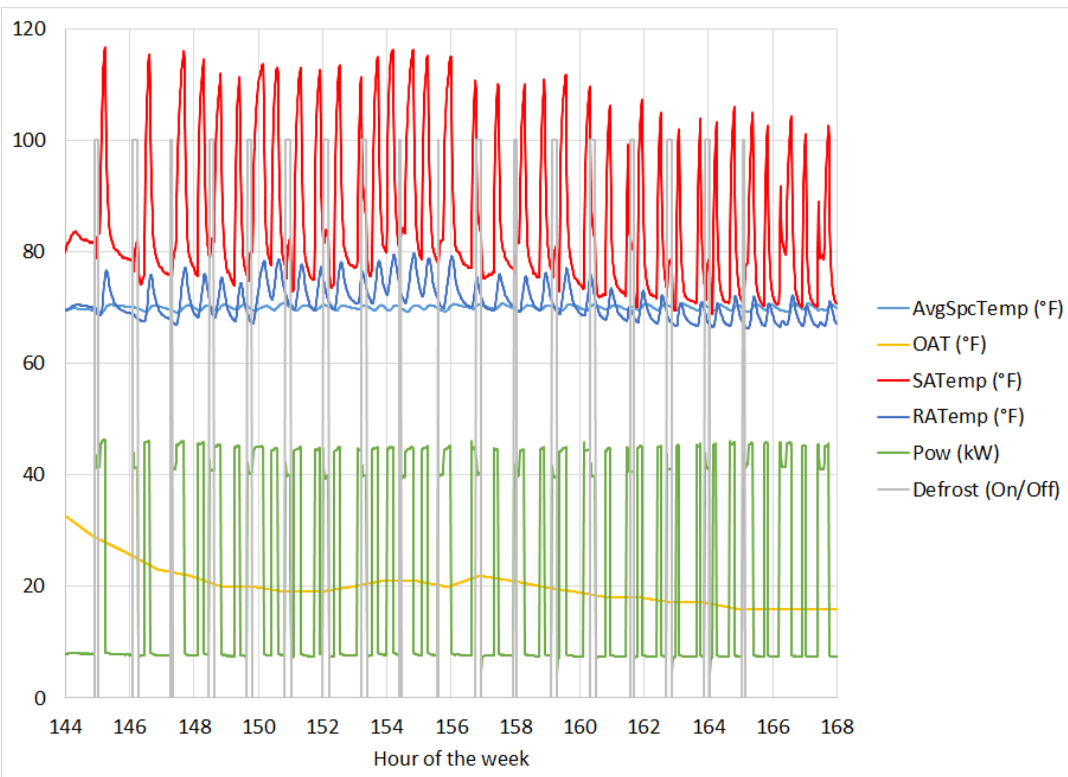


Figure 30. Baseline Transient Data from Nov. 14th, 2018.

Figure 31 shows typical CCHP heating mode operation on a slightly cooler day, but with less required load. The operation occurred from Jan 22-23rd 2019. The averages over this 18hr period are:

- OAT_avg = 10.4F
- *Q*_avg = 16.4kW
- Pow_avg = 8.9kW
- COP_avg = 1.84

Looking at the power (green) it can be seen that the lower heating stage drew about 10kW for the fans and compressors, and the higher about 17kW. Most of the time the unit was running at the lower stage. Brief electric heater spikes occur during defrost operation, which typically lasted less than 2 minutes. The supply air temperature was over 100F except during defrost when it spiked down towards the zone temperature. This indicates that the added 33kW of electric heat just cancels the cooling done to the building air during the defrost operation. There were two times that the electric heat was on briefly (2 min and 1 min) not during defrost, and the Supply Air Temperature (SAT) spiked to almost 130F. (This electric heat usage could be avoided by making the control logic less aggressive.) The unit cycled off eight times meeting the load.

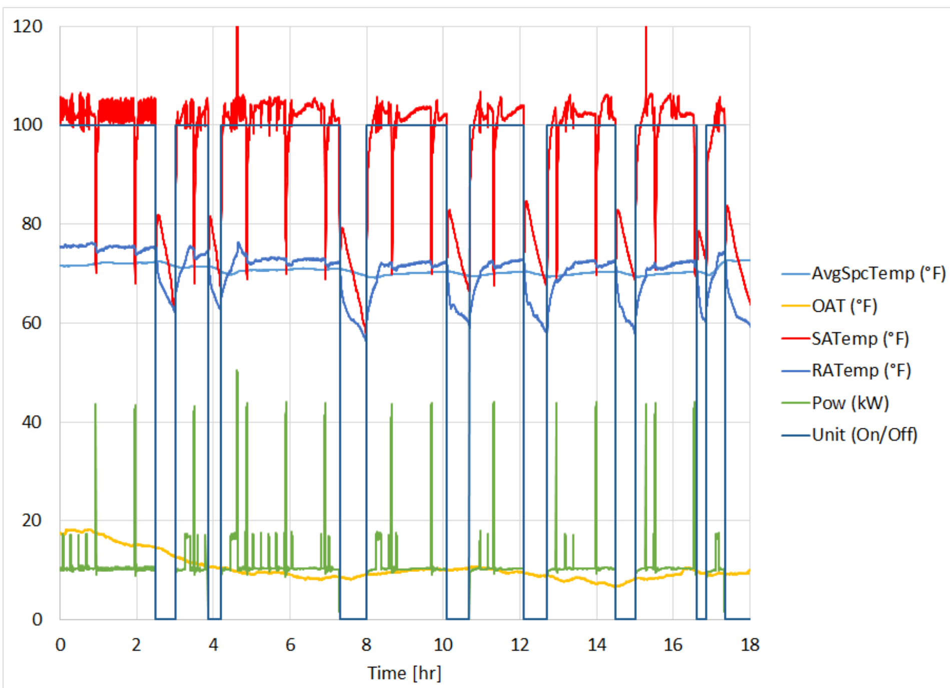


Figure 31. CCHP transient data from Jan. 22nd to 23rd, 2019.

Figure 32 shows data from February 26-27th, 2019, when the CCHP experienced its peak required capacity. The 4-hr averaged capacity over this 24hr period are: [40, 41, 24, 9, 13, and 13kW]. Since the unit cannot provide more than ~10TR (35kW) at 0F with just compressors, electric heat was required for the first 8hrs to meet load. As in the previous graph, the SAT was >100F except during defrost (which occurred after every hour of cumulative operation). There were three extended periods of electric heat at 2hr, 6.2hr, and 11.5hr. For each of these stratification is building and the Return Air Temperature (RAT) is increasing.

Figure 33 shows an expanded view of the zone temperature along with the cooling and heating set-points. It can be seen that zone temperature was well controlled. The third extended electric heat occurs during set-back-recovery (change from unoccupied to occupied in the morning). The other two could probably have been avoided by making the control logic less aggressive. The drop in required capacity from night to day is fairly dramatic, far greater than the change in ambient temperature would suggest. This was likely due to the following factors:

- Extra internal loads during occupancy, and
- Neighboring zone effects. (there was no wall between our zone and the rest of the building)

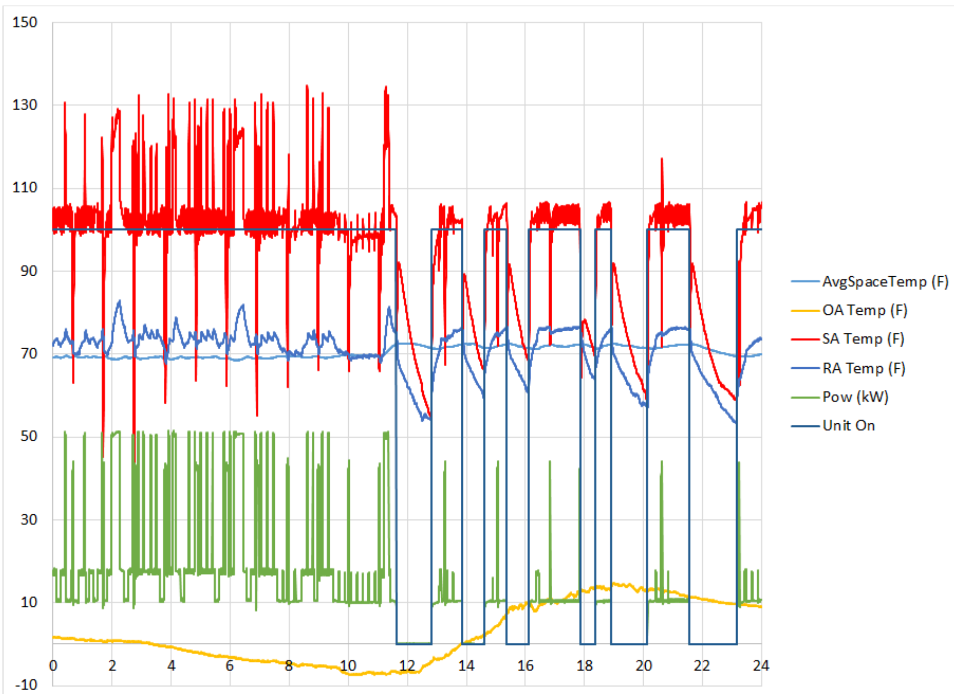


Figure 32. CCHP Transient Data from Feb 26-27th, 2019.

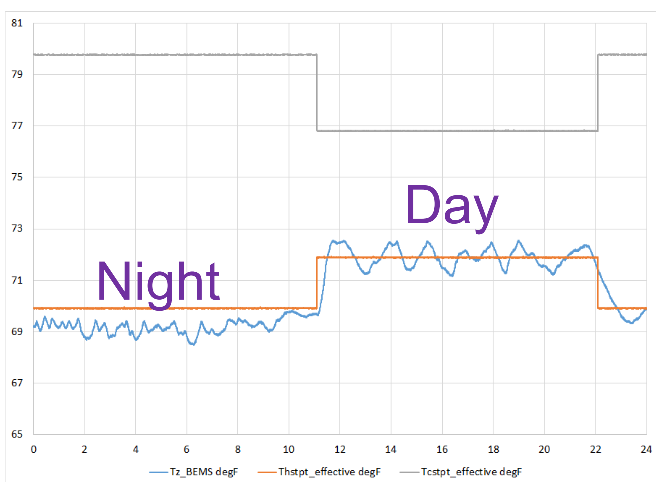


Figure 33. Corresponding Zone Control Points and Zone Temperature from Feb 26-27th, 2019.

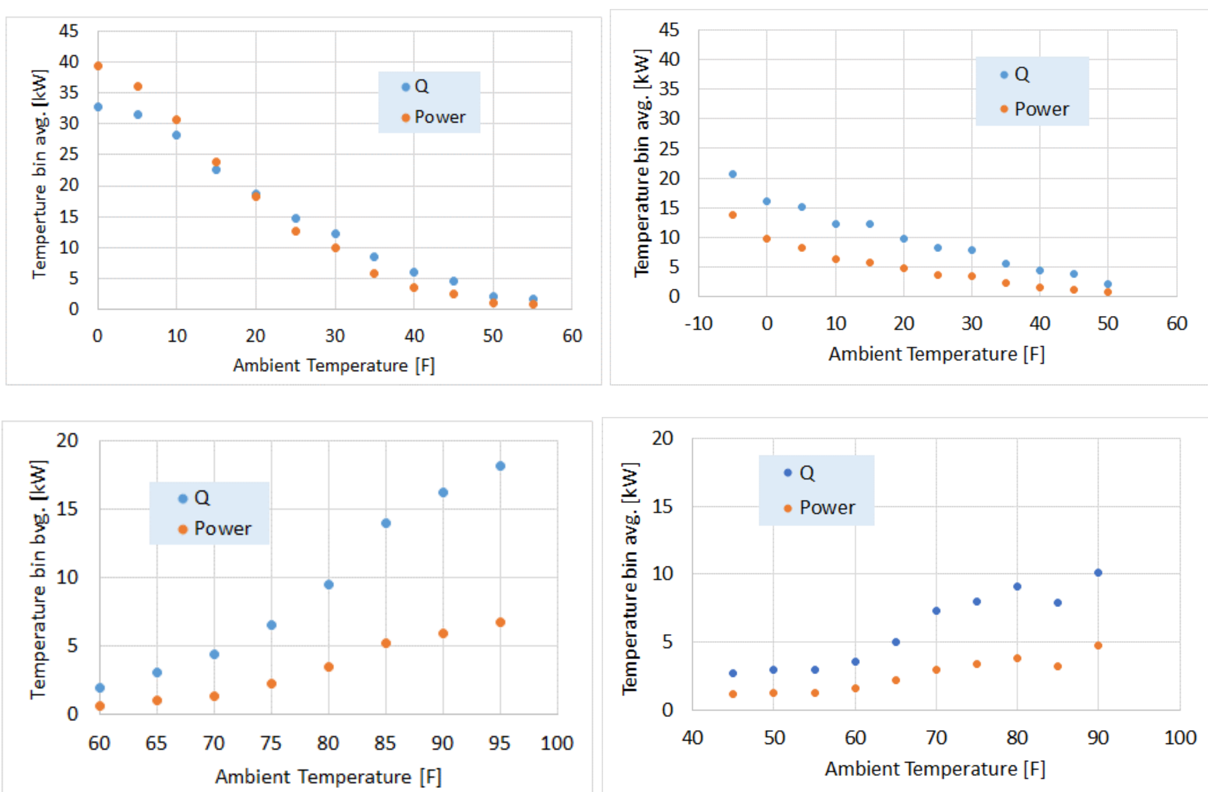
6.0 PERFORMANCE ASSESSMENT

The methods used to demonstrate each of the 6 performance objectives, described in Table 2 above, are discussed below. The primary objective was to reduce the annual energy consumption by > 20% when compared to a current SOA heat pump. Most of the data presented below has to do with that target and is presented in sub-section 1.

1. Performance and Cost Avoidance:

A primary goal of the UTRC CCHP is to demonstrate > 20% relative decrease in annual energy consumption over state-of-the-art systems. During Phase I a SOA baseline system was tested, and during Phase II the novel UTRC CCHP was tested. To compare each unit's performance an extensive set of data was obtained for each unit (see Sections 5.5 and 5.6), with the ultimate goal of determining effective COP as a function of outdoor air temperature and required load. The variables and equations used to determine COP were presented in Section 5.7.

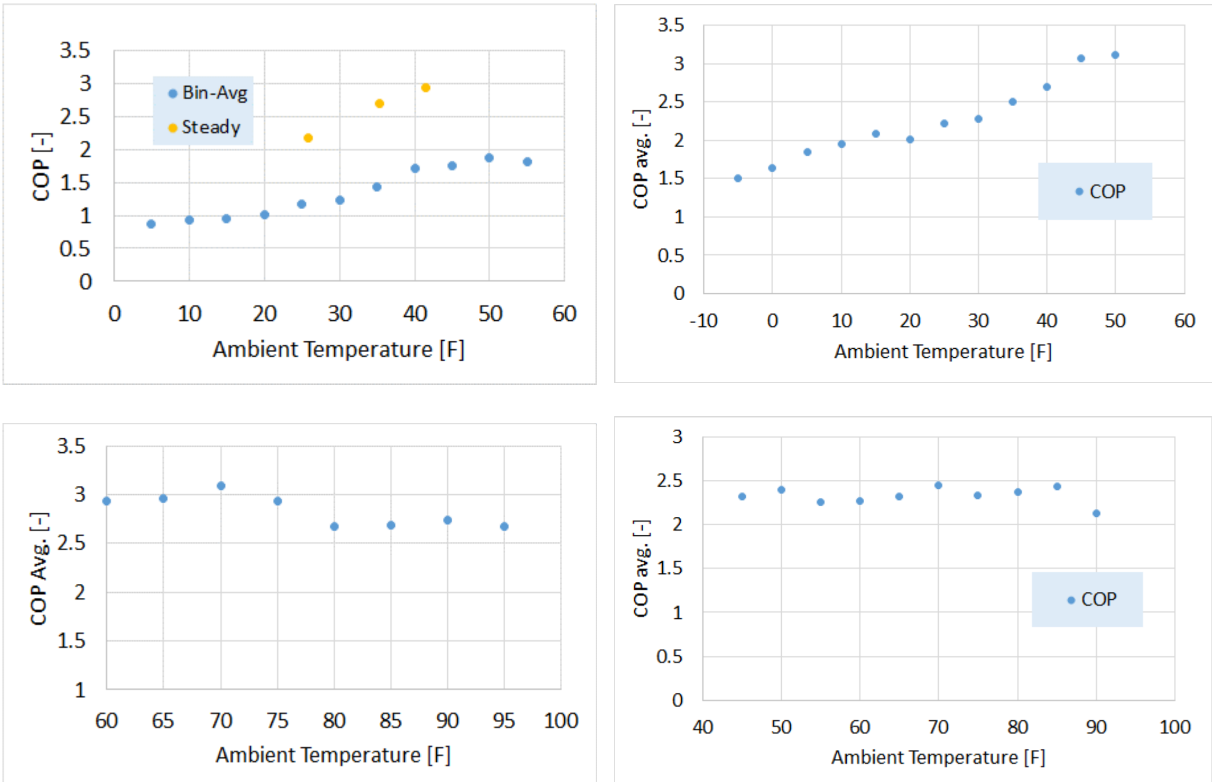
The measured values for Q and Power (presented above) are summed over 5R temperature bins (ie: the 0F bin goes from -2.5F to +2.5F), and then divided by the total number of 4-hr averages to occur at that ambient over the data set. ***Figure 34*** presents the average capacity and power at each temperature bin for the baseline unit (2018) and the CCHP (2019).



Phase I (baseline) *Phase II (CCHP)*

Figure 34. Baseline - left and CCHP - right: Temperature-bin-averaged Capacity and Power for Heating (Top) and Cooling (Bottom).

For heating it can be seen that there is an increase in load with decreasing outdoor air temperature (OAT), and for cooling the load increases with OAT. The averaged COP is determined by dividing the data in the above plots (Qavg/Pavg). The results are shown in ***Figure 35***.



Phase I (baseline) ***Phase II (CCHP)***

Figure 35. Baseline - Left and CCHP - Right: Temperature-bin-averaged Data for Heating (Top) and Cooling (Bottom).

For heating the baseline COP was very low, even below 1 for OAT < 20F. Three steady operation points (instantaneous COP) were found which had no sign of frosting, and plotted on the same graph. It can be seen that the unit is fine, and produces the expected performance when operating steadily. The dramatic loss in averaged COP is due primarily to frost/defrost cycling, and to the fact that the capacity produced by the heat pump when it is working is small and gets washed out by the other effects. For the CCHP the COP is much higher. There is a slight dip in the values from 25F to 40F, where frosting produces the greatest negative effect.

For cooling the baseline COP was fairly flat with temperature, and also lower than rated values. This was due to A) cycling, and B) the fact that the return air temperature was lower than the rated condition of 80F drybulb, 67F wetbulb. This results in lower evaporator temperatures and therefore lower COP. (This effect is discussed in Section 8.4)

The CCHP heating data shown above includes both the winter and fall heating seasons. Since the code was modified over the summer, it is useful to view the two heating seasons separately.

Figure 36-left shows the winter season in blue, and the fall season in orange. Some of the fall data is higher than the winter, especially during the heavy frosting range of 25-40F, but other points were significantly lower. These points were examined and it was found that many of the points in the temperature bins with low values had significant electrical heat, and all occurred during setback recovery (change in set point from unoccupied to occupied). This issue is discussed in greater detail in Section 8.3. If the four-hour averages associated with recovery are removed, then the bin-averaged COP becomes higher (grey points).

The code was also modified during the cooling data in order to increase the indoor air flow rate. ***Figure 36***-right shows the breakdown before and after. It can be seen that there was no significant improvement in COP. Issues associated with cooling COP are also discussed in Section 8.0.

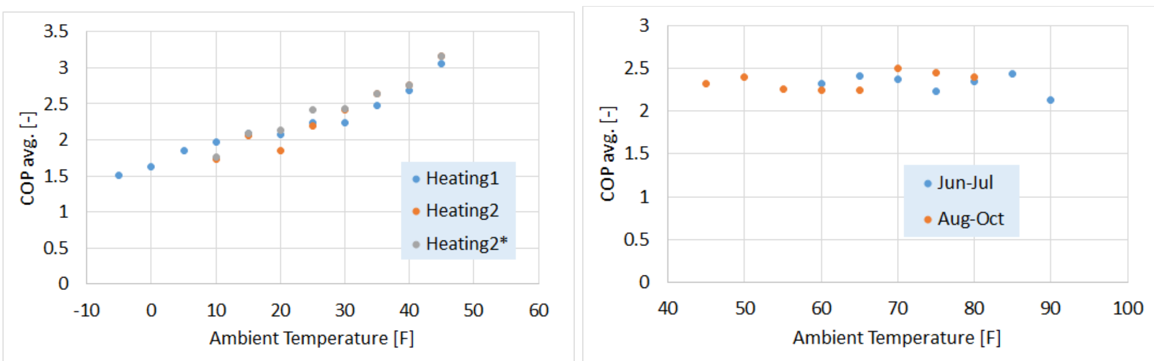


Figure 36. CCHP Temperature-bin-averaged Data, Split Heating and Cooling Seasons.

To calculate projected savings one needs:

1. Averaged COP = f(Tamb),
2. Averaged zone load = f(Tamb)
3. Annual number of hours at a temperature.

The COP curves are defined above. For the load line we used the 2018 data (which was higher than the 2019). For the number of hours at a temperature we used the TMY3 data for Augusta Maine. ***Figure 37***-left shows the combined load line from the 2018 data. It can be seen that the required heating load is twice the required cooling load for this site. ***Figure 37***-right compares the bin-hours for the 2018, 2019 and TMY3 data. It can be seen that TMY3 has more hours at mid-temperatures (45-65F), but also more hours below 15F. 2018 and 2019 were similar. Calculating the approximate heating-degree days from this data (=Sum[bin_cnt*(60F – Tbin)/24] over each temperature bin) we get for 2018 => 6150, 2019 => 6210, and TMY3 => 6250, which are all about the same.

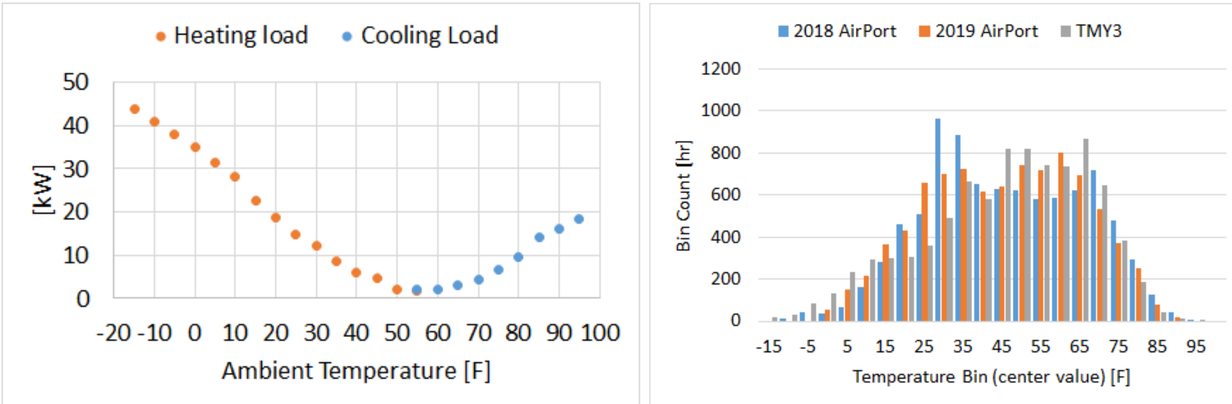


Figure 37. Left:Load-line Used for Analysis (2018 Data); Right: Annual Temperature Bin-counts.

The total power consumed over one year is:

$$kWh = \sum_i \frac{N_i Q_i}{COP_i}$$

Where:

- i covers each temperature bin -15 to 100F,
- N is the number of hours in each bin (TMY3),
- Q is the Camp Keys load (above plot), and
- COP is the Q/P which varies depending on baseline or CCHP.

Figure 38 shows the bin-summed-values for capacity (left) and power (right). It can be seen that heating dominates the total power consumption over the year. It can also be seen that the CCHP power was significantly lower during the heating season, but slightly higher during the cooling season.

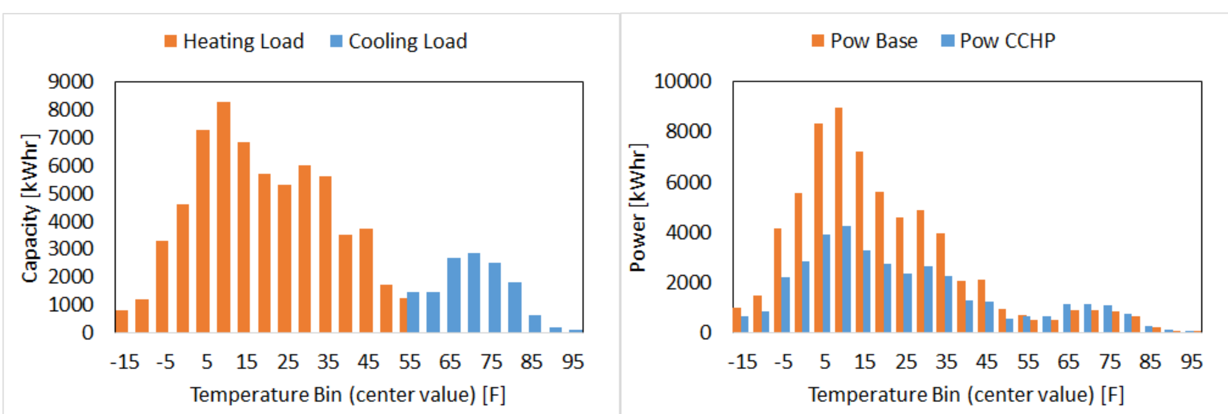


Figure 38. Left: Temperature-bin-summed Load; Right: Temperature-bin-summed Power.

The final step is to add up all the bins:

The heating season by itself achieved a <u>savings</u> of 30066/61465 = 49%.

	Base Unit	CCHP	Savings
Annual kWhr	61465	31399	30066
Cost Elect.	$ 0.12	$ 0.12	
Annual Cost	$ 7,376	$ 3,768	$ 3,608

The cooling season by itself had an <u>increase</u> of 1131/4696 = 24%.

However, the energy use during cooling season is small compared to heating, and it has little effect on the overall cost/energy savings.

	Base Unit	CCHP	Savings
Annual kWhr	4696	5827	-1131
Cost Elect.	$ 0.12	$ 0.12	
Annual Cost	$ 564	$ 699	$ (136)

The combined results (heating + cooling) demonstrated 37226/66161 = 44% <u>savings</u>, far exceeding the 20% target, at least for this climate zone.

	Base Unit	CCHP	Savings
Annual kWhr	66161	37226	28935
Cost Elect.	$ 0.12	$ 0.12	
Annual Cost	$ 7,939	$ 4,467	$ 3,472

As discussed below in Section 8.0, the lower performance during cooling season was due to control and operation (BCMS) settings, not unit performance. If it had it equaled the baseline unit in cooling mode, then the combined results would have been: 30066/66161 = 45.4% savings.

	Base Unit	CCHP	Savings
Annual kWhr	66161	36095	30066
Cost Elect.	$ 0.12	$ 0.12	
Annual Cost	$ 7,939	$ 4,331	$ 3,608

The same analysis can be done using TMY data from other locations. The following three additional sites were chosen to calculate expected savings:

a. Fort Knox, Kentucky, in ASHRAE zone 4A.

b. Colorado Springs, Colorado (Fort Carson), in ASHRAE zone 5B.

c. Houlton Maine, ASHRAE zone 7A.

The binned temperature data is shown in ***Figure 39***.

The same capacity load line and COP as a function of outdoor temperature was used as above. It can be seen that >20% savings is achieved for all cold climates.

Table 7. Energy and Cost Savings for 4 Different Climate Zones

Fort Knox, Kentucky
ASHRAE Zone 4A

	Base Unit	CCHP	Savings	%
Annual kWh	41039	28481	12558	31%
Electric $/kWh	0.12	0.12		
Annual Cost	$ 4,925	$ 3,418	$ 1,507	

Colorado Springs, CO
ASHRAE Zone 5B

	Base Unit	CCHP	Savings	%
Annual kWh	46766	28842	17924	38%
Electric $/kWh	0.12	0.12		
Annual Cost	$ 5,612	$ 3,461	$ 2,151	

Augusta Maine,
ASHRAE Zone 6A

	Base Unit	CCHP	Savings	%
Annual kWh	66161	37226	28935	44%
Electric $/kWh	0.12	0.12		
Annual Cost	$ 7,939	$ 4,467	$ 3,472	

Houlton, Maine
ASHRAE Zone 7A

	Base Unit	CCHP	Savings	%
Annual kWh	87451	47908	39543	45%
Electric $/kWh	0.12	0.12		
Annual Cost	$ 10,494	$ 5,749	$ 4,745	

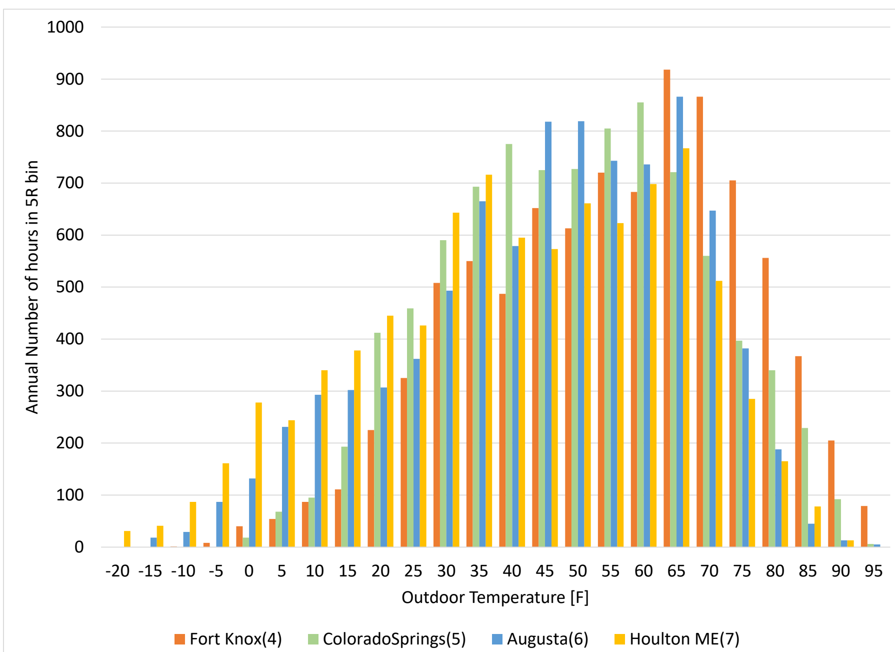


Figure 39. TMY3 Binned Data for Four Selected Sites in Climate Zones 4-7.

2. Air Supply Temperature:

Supply air temperature (SAT) was directly measured during the Phase I and Phase II testing. When the unit is cycling, the SAT varies with time. For the baseline unit the SAT is low when the electric heat is off, but increases significantly with electric heat. For the CCHP the SAT was >95F for all steady operation since it has greater heat pump capacity and has variable indoor fan logic. Generally, it was > 100F. ***Figure 31*** and ***Figure 32*** above show SAT data.

3. Direct Greenhouse Gas Emissions:

The direct greenhouse gas emissions are calculated directly from the annual energy consumed by the heat pump. Since the unit met the energy efficiency target (sub-Section 1 above), and consumes only electric power, it by definition meets the greenhouse gas target of >20% reduction. The U.S. Energy Information Administration site states:[3]

> "In 2018, total U.S. electricity generation by the electric power industry of 4.17 trillion kilowatthours (kWh) from all energy sources resulted in the emission of 1.87 billion metric tons—2.06 billion short tons—of carbon dioxide (CO_2). This equaled about 0.99 pounds of CO_2 emissions per kWh."

For the 10TR unit deployed at the Maine field trial site, 28935 kWh of electricity could be saved each year compared to using a traditional heat-pump/cooling unit. This would translate to 14 tons of CO_2 at this site.

4. Scalability across DoD:

The technology used in the CCHP is scalable beyond 40TR, cost effective with no change to the baseline unit's footprint and installation complexity from existing roof top units.

5. Ease of Use:

There were no product-related issues associated with installation and use of the CCHP. There was a water-leak issue that was noticed during the first few weeks after installation which required removal and rework of the roof-top curb. This issue was due to the fact that the unit-cassis had been used as a laboratory unit for several development cycles, and the base pan was not properly resealed. This is not a product issue. After Phase II at the Camp Keys site was completed, the unit was returned to UTRC and placed in our psychometric chambers. The unit was found to perform as-per design, proper oil level was found in the compressors, and all components behaved as before the field trial. As stated above, the CCHP is designed to fit in a standard roof-top cassis, and to the customer there are no additional costs or learnings compared to current rooftop units.

6. Thermal Comfort:

Thermal comfort was assessed by AJ. Ballard at the Maine Camp Keys field trial site via verbal surveys of the building occupants. All site surveys produced positive feedback relative to thermal comfort. In addition, it can be demonstrated from the data that the CCHP unit was successful in maintaining the zone temperature at the set point. ***Figure 33*** above shows data from the highest heat load day which required supplemental electric heat. ***Figure 40*** below shows additional examples from both heating and cooling. These results were typical.

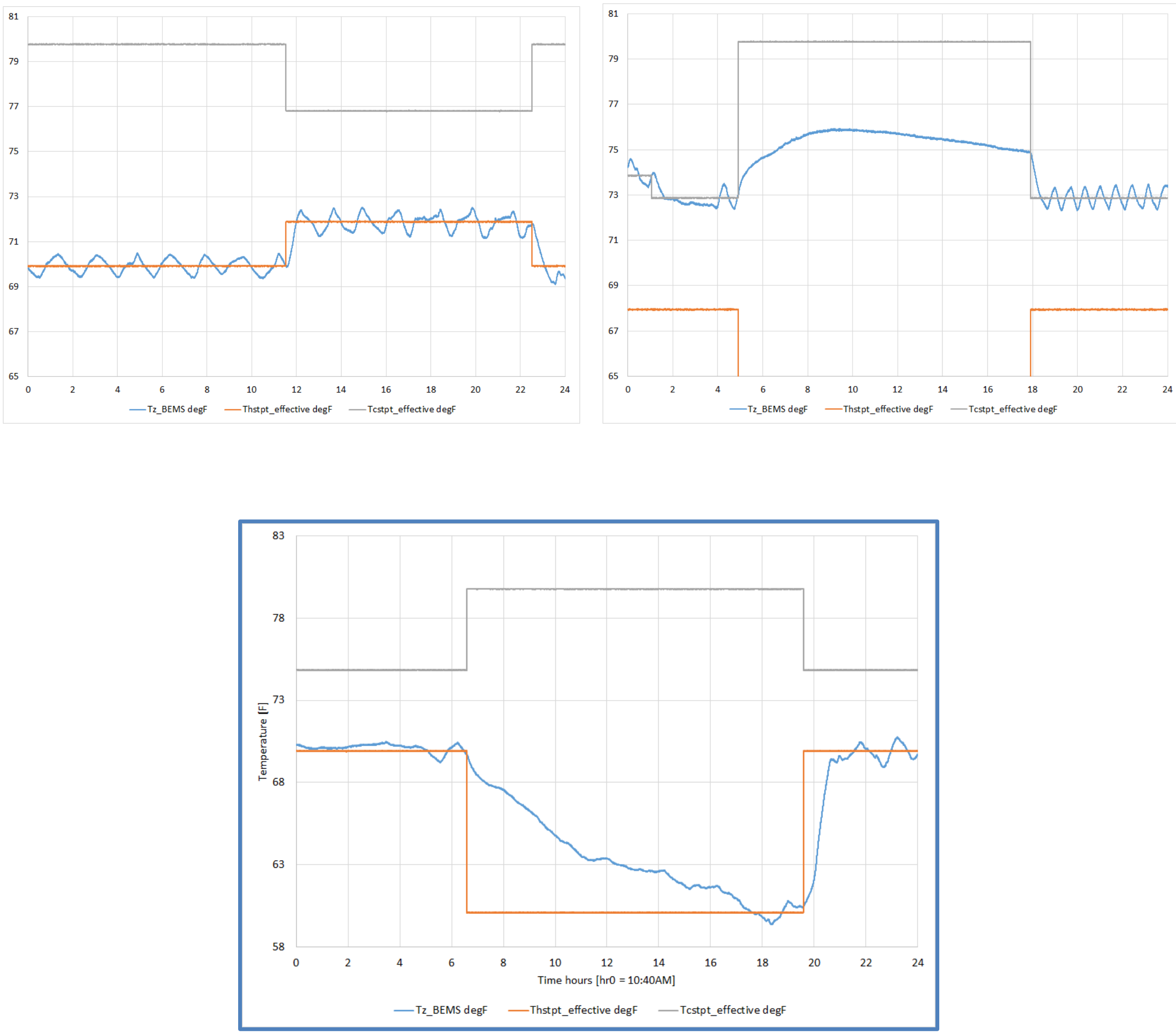


Figure 40. Three Examples of Typical Zone Control.

Top-Left: heating Feb. 24-25th, 2019 (no electric heat required except during defrost); Top-Right: cooling July 17th-18th; Bottom: Heating Nov. 12-13 (large change in zone set-point from occupied to unoccupied).

7.0 COST ASSESSMENT

There are three primary costs associated with the proposed technology

- Purchase,
- Installation, and
- Operational cost.

The technology developed by UTRC for the CCHP has been designed to **meet the current pricing** of state-of-the-art (SOA) mid-tier heat pumps; therefore, there is little to no cost penalty to purchasing of this technology. In addition, since standard RTU packaging has been maintained, the installation is the same as for a standard cooling/heating RTU. The annual operational cost benefit exceeds all other cost penalties.

Cost Element	Nominal 10TR unit cost	Estimated Cost difference From SOA heat pump.
Hardware capital costs	Roof top heat pump unit	<$1200 additional cost to customer from added compression technologies.
Installation costs	Labor and material required to install	No change (possible significant reduction if electric heater can be removed, see below)
Facility operational costs	Reduction in energy required vs. baseline data	Savings between $1-5K annually, depending on climate zone.
Maintenance	• Frequency of required maintenance • Labor and material per maintenance action	No change.
Hardware lifetime	Estimate based on components degradation during demonstration	No change (no degradation found from post-field-trial tests in lab)
Operator training	Estimate of training costs	No training required.

The field trial unit deployed had a 33kW electric heater. It was used to meet capacity for the peak load days, and to limit the effect of cold blow during the defrost cycle. If the peak load were slightly lower, then it might be possible to eliminate the electric heater all together along with the associated cost. Also, since this unit defrosted much quicker than the baseline unit, the momentary cold supply air would be unlikely to reach the zones. However, since we did not demonstrate this during field trial, we cannot claim credit at this time for cost avoidance that may be achieved through a) lower kW demand charge, b) lower hardware cost due to reduced or eliminated electrical heater requirements, and c) avoidance of oversized building wires, breakers, and associated installation costs.

8.0 IMPLEMENTATION ISSUES

As mentioned above in Section 6.0, under "Ease of Use," there was one installation issue that resulted from the fact the prototype chassis had been used for multiple development cycles, but this was not a product related issue. In addition, there were several minor code changes made during the field trials which included changes in sensor limit trip-points and fan speed control, but these changes were successful implemented and related trips were avoided after they were made. None of these issues will translate to product. In fact, one of the main reasons for doing field trials is to develop the controls to TRL 8. The other is to look for reliability issues (such as oil return) that may not have shown up in the laboratory testing. (Oil return is often a concern especially when multiple compressors share a common refrigerant circuit. During part load, the tube velocities are low and the oil may not get dragged back through the vapor lines. In addition, a system needs to be in place to ensure equal oil is maintained between the compressors and does not get "sucked out" during transients. The baseline unit had separate refrigerant circuits for the two compressors which did not mix.)

As mentioned above, no issues of this type were found. The CCHP is designed to fit in standard roof-top chassis, and there are no installation differences to the end user, and no differences in applicable regulations. Most components in the CCHP are COTS. The one component that is not (UTC proprietary) can and has been made by standard manufactures.

> *Note: The control details and new defrost logic are considered proprietary and are not included in this public document. A separate document has been prepared for Carrier detailing all control modifications.*

<u>Field Trial related</u> issues were more numerous. These relate to being able to do a fair comparison of the baseline unit to the CCHP, but do not directly relate to the product. They include:

1. Outdoor air temperature from BMCS was not accurate, especially during cooling seasons. Neighboring airport data was downloaded and used instead.
2. The zone load as a function of temperature was not consistent between Phase I (2018) and Phase II (2019), even though the outdoor temperature profiles were very similar. This likely resulted from differing zone control settings in neighboring zones in the Camp Keys building.
3. In our zone, the unoccupied set-point was not always consistent (early fall season). This resulted in excess electricity during set-back recovery for the fall heating season, and therefore poorer integrated COP during these times. Improved use of set-back and code modifications to avoid electric heat under these circumstances should be implemented.
4. Cooling COP was worse than expected. This was due to a combination of a) running in cooling mode at low temperatures but not implemented the economizer, b) cool return air temperatures leading to cool supply air temperatures and therefore lower compressor suction pressures, and 3) insufficient air flow given the above.

Each of these issues are discussed below.

8.1 OUTDOOR AIR TEMPERATURE ACCURACY

The outdoor air temperature is used in the analysis to:

1. Calculate the heating/cooling load associated with outdoor ventilation air,

2. Determine the heating/cooling loads as a function of ambient, and
3. Rate units' efficiency as a function of ambient temperature.

We had several sources of outdoor air measurements:

1. The existing BMCS system provided two different outdoor air sensors that it had previous to this field trial,
2. For Phase II we had a sensor in the outdoor air ventilation hood. This sensor would read outdoor air temperature properly whenever we were ON, and air was being drawn into the unit.
3. The site was located directly adjacent to an airport (***Figure 41***-left). Airport data was available and downloaded to be merged with the other data sources.

Up to 20R differences were found between the BMCS data and the airport data during the cooling season (***Figure 41***-right), but <4R during the heating season. Airport data was downloaded and merged with all test data to be used in the final analysis. Other examples of outdoor air differences can be found in the following sub-sections. The BMCS OAT sensor was located on a wall that faces north-north-east. It is likely that when the sun is both higher and further north in the sky that radiation heating from the surrounding pavement is affecting the reading.

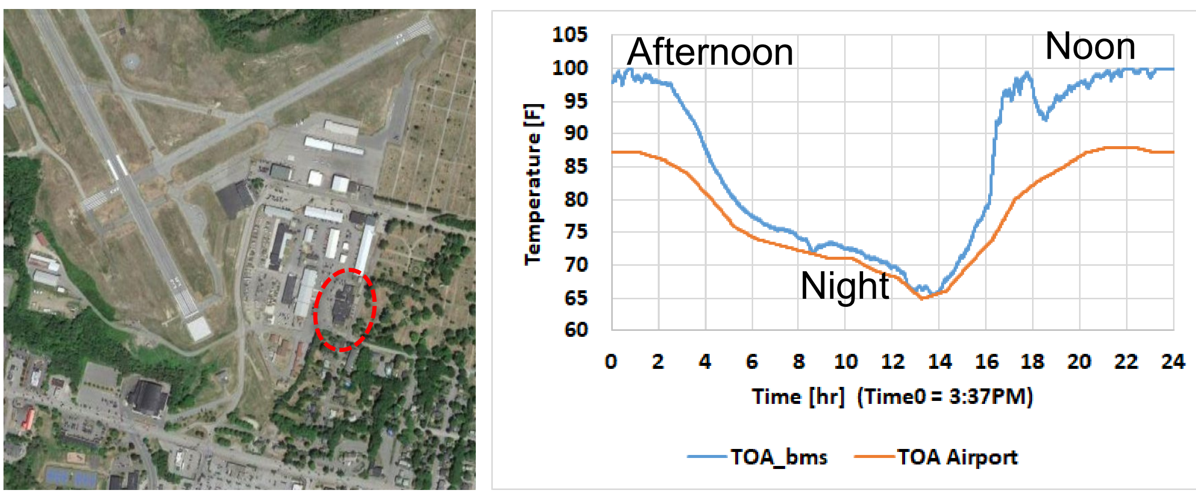


Figure 41. Left: Aerial View Showing Building 39 and Augusta Airport; Right: Example from July 4-5th, 2019, Comparing BMCS Data to Airport.

8.2 LOAD CONSISTENCY FROM 2018 TO 2019

As mentioned above the peak loads from 2018 to 2019 were consistent, but the bin-averaged loads were not. It appears that this may have resulted from neighboring zone effects. Two examples are shown in ***Figure 42***. The top shows data from October 19-20, 2019. The left plot shows key temperatures, the unit did not run during this period. The right shows the zone temperature alone. Even though the outdoor air temperature dropped to ~30F overnight, the zone only cooled <2R. ***Figure 42***-bottom shows data from March 27-28, 2019. During the occupied period of the 27th the zone temperature was rising even though the CCHP did not run and the ambient was ~40F.

The next day you can see three times there was heat added to the zone. The first was from the CCHP at 16hr, the second was from the old baseboard unit that was maintained in the zone and run a few times a week to keep it operational, and the final was the CCHP running with the set-point changing from 70 to 72F. At 21.5hr the zone begins to heat again for no apparent reason.

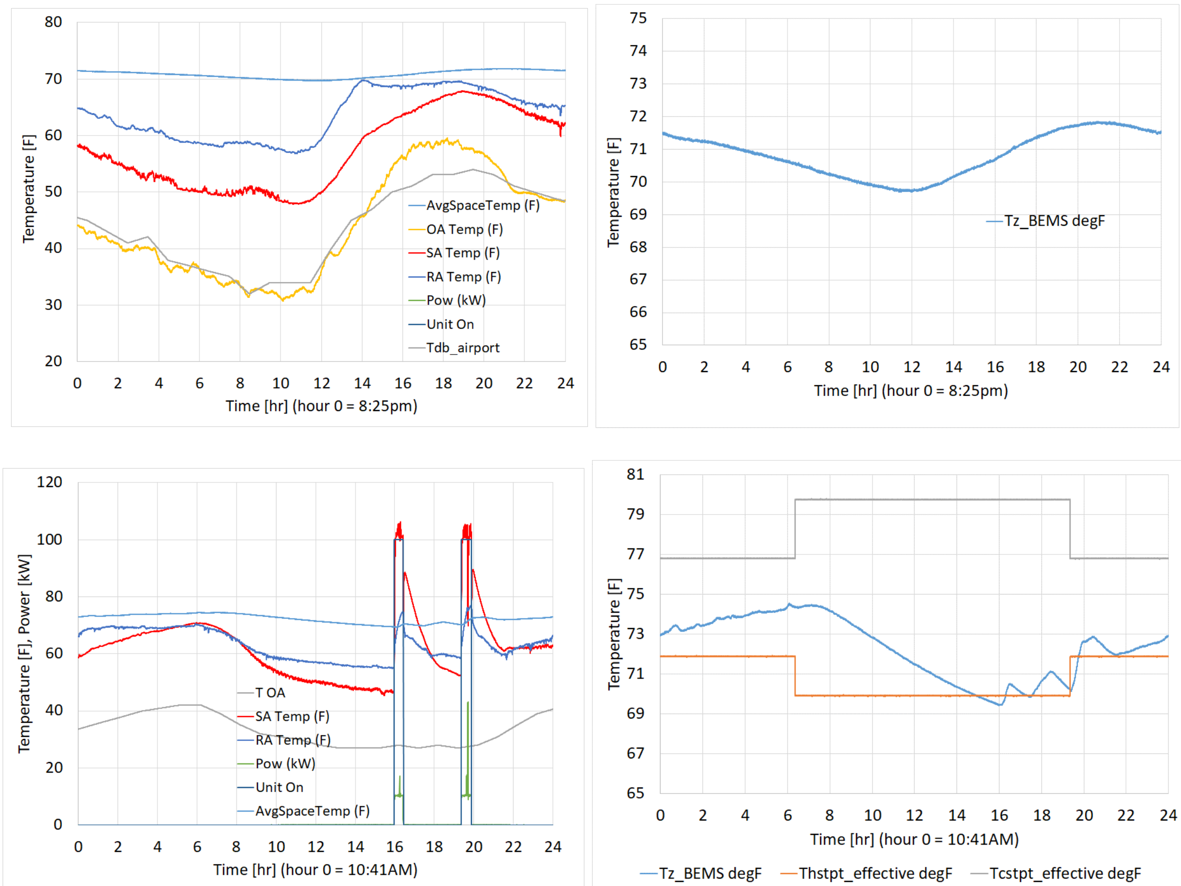


Figure 42. Top-left: Temperatures from Oct 19-20, 2019; Right: Expanded View of Zone Temp.

Bottom-left: temperatures from Mar 27-28, 2019; Right: Zone and set-points.

8.3 USE OF ZONE SET BACK MAY ACTUALLY COST ENERGY

The main benefit of the CCHP is that it limits the need for supplemental electric heat; however, during the fall heating season it was found that extensive electrical heat was used due to large changes in the zone set-point. ***Figure 43***-left shows the zone temperature and set points for Nov 12 to 13. During the night (unoccupied) the heating set-point dropped from 70 to 60F. Even though the outdoor air was only ~10F, the zone did not reach 60F until morning, less than 2hrs before "occupied." ***Figure 44*** shows key temperatures and unit power draw expanded around the time the unit comes back on. Peaks in power draw (green) above 40kW occur when the electric heater is on. The two shorter periods are during defrost cycles. The long one starting at 20hr results from set-back recovery. The controller sees that the current zone temperature is far from the desired set-point and it stages up to full heating capacity including the electric heaters.

With just the heat-pump the supply air temperature is held around 100F (2.), but when the electric heat is added the supply temperature jumps to >120F, even with full air flow. During this time stratification in the zone becomes significant in that the return air temperature climbs to 90F (3.), with the corresponding supply air temperature reaching 140F (4.). The unit trips on high SAT, then resets and restarts with just the compressor stages. Since it is now close to set-point, the electric heater stays off and the zone temperature is maintained properly without electric heat. For most of the winter season (***Figure 43***-right) the set-point was 72F occupied and 70F unoccupied, and electric heat was not required on recovery. After this difference was noticed in the fall data, the set points were made to be consistent with the winter operation.

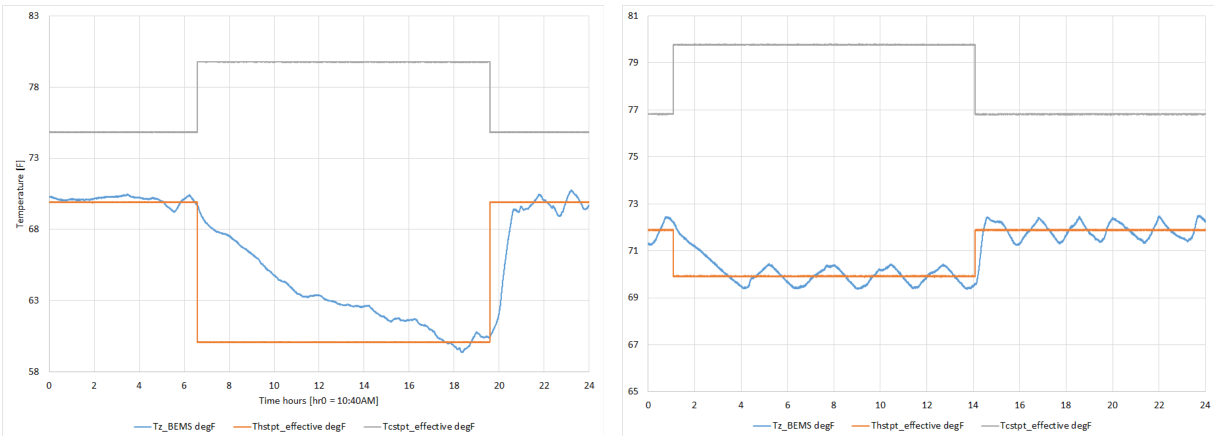


Figure 43. Zone Temperature and Zone Set-points.

Left: Nov 12-13, 2019; Right: Feb 12-13, 2019

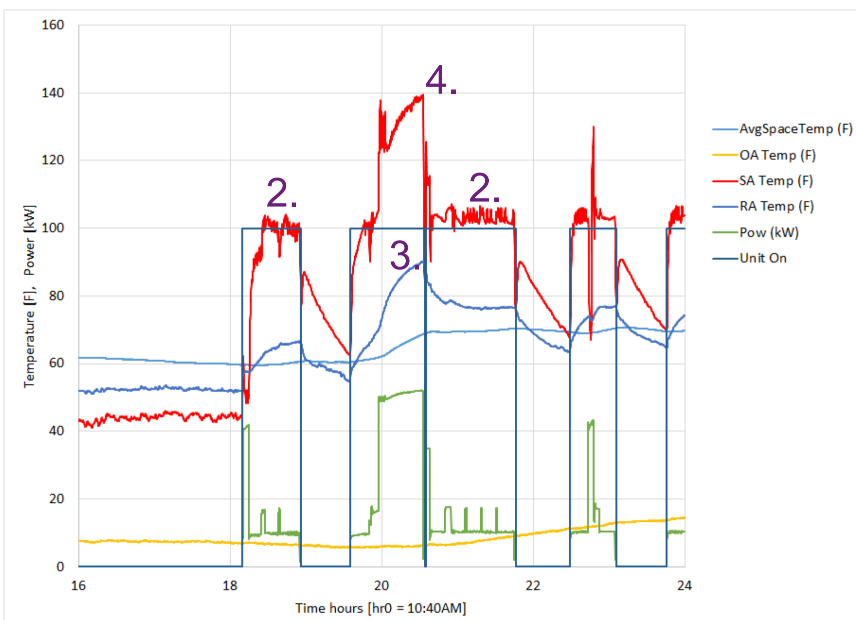


Figure 44. Time-expanded View Around Set-back Recovery, Nov 13, 2019, Showing Key Temperatures and Unit Power Draw.

In general energy can be saved by using occupancy setback if the unit runs at similar or higher efficiency during the recovery time. Most of the benefit is due to a lower averaged zone-to-ambient temperature difference and therefore integrated load. There also may be a benefit if the unit runs at higher efficiency due to less on/off cycling (avoiding cycle degradation). For cooling, occupancy setback it likely to save energy. This is because:

A) The averaged load is reduced due to smaller difference between the zone and outdoor air temperature.

B) The unit runs more continuously during recovery than it would have over night and therefore has slightly higher average efficiency, and

C) The recovery occurs in the early morning hours when the outdoor air is usually at its minimum value and therefore the unit COP for cooling is higher.

For heating, it may not save energy because:

A) Although averaged load is reduced due to a smaller difference between the zone and outdoor temperature,

B) the unit may bring on electric heat and therefore run at a much lower COP, and

C) the recovery occurs in the early morning hours when the outdoor air is at its minimum and therefore the unit heat pump COP is lower.

For this example a 10R drop in set point from 70F to 60F resulted in a drift down over 13 hours with an average zone temperature of ~63F. The outdoor ambient was ~10F.

- The percent load reduction was therefore (70F – 63F) / (70F – 10F) = 12%.

- The unit heat pump only COP is ~2 at 10F, but with both electric heat and heat pump active is about (2 + 1.2)/(1 + 1.2) = 1.45, or a 27% reduction in COP which more than cancels the savings.

Of course, the control logic could be modified to recognize set-back recovery and not use the electric heat during this time. In fact, for this application electric heat was really only necessary during defrost operation. Without electric heat the recovery time would be longer, but that could be accounted for in when the set point is changed.

8.4 LOW COOLING AIR SUPPLY TEMPERATURE LEADING TO LOW COP

HVAC equipment is rated by the industry under "standard" conditions. The return air temperature (RAT) and humidity is assumed to be 80F with 67F wet bulb (or 51%RH), and typically the air flow rate is set so that the SAT is ~56F. Unfortunately, these conditions are rarely seen at actual field sites. For our field-trial site at Camp Keys, the cooling mode occupied set-point was typically set to 73F. The supply and return registers were both located on the ceiling, so stratification should have produced a return temperature that was slightly higher than the zone, but typically we saw a return temperature that was several degrees cooler. With cooler RAT, for the same capacity and air flow rate, the SAT is also cooler. This leads to lower compressor suction pressure and lower COP.

With outdoor ventilation air mixed with the return air, the air reaching the evaporator may be warmer or cooler depending on the ambient temperature. ***Figure 45*** shows cooling mode data from Aug 26-27th, 2019. There are two values for outdoor air temperature, yellow=BMCS, and grey=Airport. (As mentioned above, they agree at night and during cooler temperatures, but deviate during the heat of the day.) The SAT is very cold (almost 40F) when the unit starts in the morning and the ambient is only 50F. As the day warms up the SAT also warms, due to warmer ventilation air mixing with the return air. The return air temperature is always several degrees cooler than the SAT while the unit is ON.

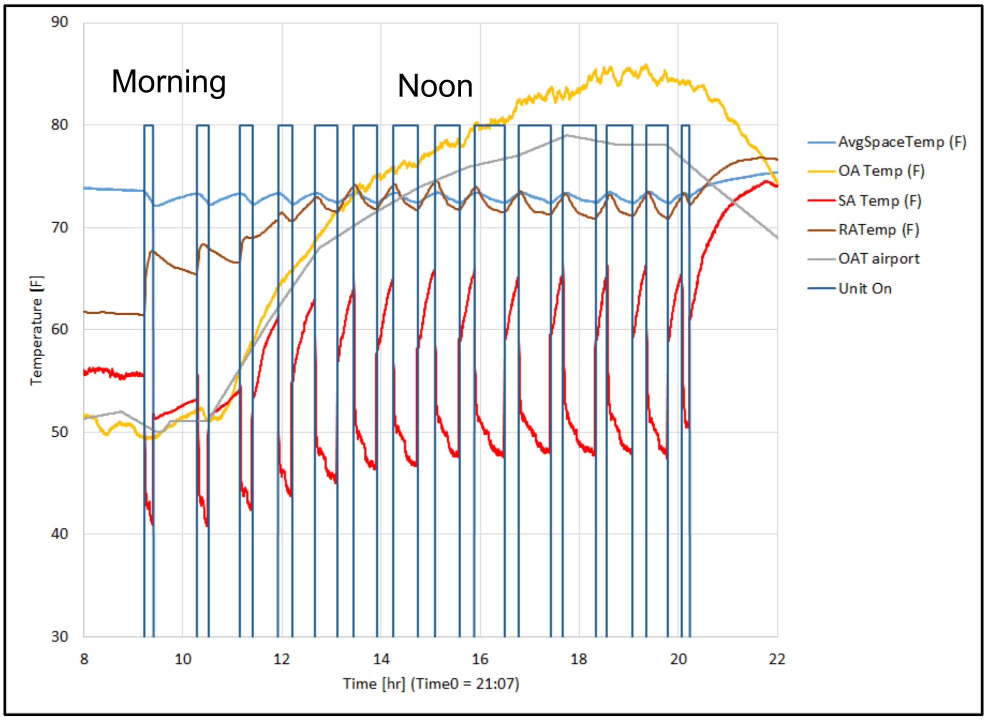


Figure 45. Cooling Example, Low Supply Air Temperatures.

Cooling COP would have been higher if:

A. We had implemented the economizer for operation at ambient temperatures < 65F, or

B. We had further increased the indoor air flow rate to increase the SAT.

However, in order to achieve rating values the RAT would need to be 80/67F, but this condition could not happened at the Maine site.

9.0 REFERENCES

1 Mahmoud, Ahmad. 2016. “High-Efficiency Commercial Cold Climate Heat Pump.” presented at the 2016 Building Technologies Office Peer Review. https://energy.gov/sites/prod/files/2016/04/f30/312104_Mahmoud_040716-945.pdf

[2] Verma, Parmesh. 2017. “Cold Climate Commercial (Rooftop) Heat Pump.” presented at the Energy Exchange, Tampa, Florida.

http://www.2017energyexchange.com/wpcontent/uploads/T2S7_Verma.pdf

[3] Government data used for CO2 generation for kWh of electricity used.

https://www.eia.gov/tools/faqs/faq.php?id=74&t=11

APPENDIX A POINTS OF CONTACT

Point of Contact	Organization	Phone & E-mail	Role in Project
Dr. Frederick Cogswell	UTRC	860-610-1688 cogswefj@utrc.utc.com	Principal Investigator
Mr. Alan J. Ballard	Maine Army National Guard	207-430-5679 alan.j.ballard.hfg@mail.mil	Energy Director
Dr. Ahmad Mahmoud	Carrier	860-610-7149 Ahmad.M.Mahmoud@carrier.com	Initial PI, Now key Carrier contact