# FINAL REPORT

Gas Engine-driven Heat Pump (GHP) Cold Climate Field Demonstration

ESTCP Project EW-201515

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**Gas Technology Institute**



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**14. ABSTRACT**
Gas engine-driven heat pumps (GHPs) are an emerging space conditioning technology for multi-zone commercial buildings. GHPs have potential to reduce peak electric demand, energy use, and lifecycle costs compared to conventional equipment. This demonstration was a side-by-side comparison of two variable refrigerant flow (VRF) technologies, a GHP and an electric cold climate heat pump (CCHP), paired with a dedicated outdoor air system (DOAS). Based on measured data, the GHP/DOAS system reduced peak electric demand by up to 59 kW (90%) compared to the baseline VAV system, and by up to 30 kW (82%) compared to the CCHP/DOAS. Despite lower than expected part-load performance, the GHP/DOAS life-cycle costs were 4% lower than CCHP/DOAS and 37% lower than the baseline. Both VRF systems improved comfort along with significant energy savings, lower peak electric demand, and potential savings in life-cycle costs compared to conventional VAV systems. The demonstration also identified operational issues for both heat pump technologies in this application.

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# ACRONYMS AND ABBREVIATIONS

| | |
|---|---|
| ANSI | American National Standards Institute |
| ASHP | Air source heat pump |
| ASHRAE | American Society of Heating, Refrigeration, and Air Conditioning Engineers |
| | |
| BAS | Building Automation System |
| Btu | British thermal unit |
| | |
| CFR | Code of Federal Regulations |
| CHP | combined recovered heat and electrical power |
| CCHP | electric cold climate heat pump; a type of ASHP designed for cold climate use |
| CDD | Cooling Degree Days |
| COP | Coefficient of Performance |
| | |
| DoD | United States Department of Defense |
| DoE | United States Department of Energy |
| DAS | data acquisition system |
| DOAS | dedicated outdoor air system |
| | |
| EO | Executive Order |
| EPA | U.S. Environmental Protection Agency |
| ESTCP | Environmental Security Technology Certification Program |
| EW | Energy and Water |
| | |
| FEMP | Federal Energy Management Program |
| | |
| GHP | Natural gas engine-driven heat pump |
| gpm | Gallons per minute |
| GWP | Global Warming Potential |
| | |
| HDD | Heating degree days based on a 65°F base temperature |
| HVAC | Heating, Ventilation and Air Conditioning |
| | |
| kW | Kilowatt |
| kWh | Kilowatt-Hour: 1,000 Watt-Hours of electricity |
| | |
| lb/hr | Pounds per hour |
| | |
| MW | Megawatt |
| M&V | Measurement and verification |
| MMBtu | Million British Thermal Units |
| MW | MegaWatt: 1 million Watts of power |
| MWh | MegaWatt-Hour: 1 million Watt-Hours of electricity |
| | |
| NIST | National Institute of Standards & Technology |

| | |
|---|---|
| OA | Outside air |
| O&M | Operation and maintenance |
| | |
| PI | Principal Investigator |
| PM | Project Manager |
| POC | Point of Contact |
| | |
| RH | Relative humidity |
| RCRA | Resource Conservation and Recovery Act |
| | |
| SCADA | Supervisory Control and Data Acquisition |
| SCFM | Standard cubic feet per minute |
| | |
| THC | Total hydrocarbon |
| TMY | Typical Meteorological Year |
| | |
| VOC | Volatile organic compound |
| VAV | Variable air volume packaged HVAC unit; provides variable air flow, gas heating, and electric air conditioning with electric reheat. |
| VRF | Variable refrigerant flow aka VRV variable refrigerant volume (Daikin's proprietary VRF design) |

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# ABSTRACT

## INTRODUCTION AND OBJECTIVES

Gas engine-driven heat pumps (GHPs) are an emerging technology offering a cost-effective option to reduce peak electric demand, electricity consumption, and lifecycle costs as compared to conventional space conditioning equipment. The objective of this project was to evaluate the annual performance of a GHP system in a side-by-side comparison with an electric cold climate heat pump (CCHP) relative to the baseline performance of existing conventional HVAC at Naval Station Great Lakes (NSGL) in North Chicago, Illinois. Both heat pumps selected for this demonstration had variable refrigerant flow (VRF) configurations to provide multi-zone heating and cooling. The demonstration evaluated the energy and economic benefits of each technology for cold climate DoD applications based on measured performance data.

## TECHNOLOGY DESCRIPTION

The GHP design is similar to an electric vapor compression heat pump, but in place of the electric motor, GHPs use an advanced natural gas engine with a demonstrated long life (30,000 hours). During cooling, GHPs consume natural gas in place of electricity, significantly reducing peak electric use. GHP rated efficiency is 50% higher than standard gas furnaces or boilers. Heat recovered from the engine is used to supplement the GHP output during heating mode to increase the overall system efficiency and maintain supply temperatures at low ambient conditions. In contrast, electric heat pumps often require inefficient resistance heating to supplement the heat pump output at cold temperatures.

## PERFORMANCE AND COST ASSESSMENT

Based on measured data, the GHP reduced peak electric demand by up to 59 kW (90%) compared to the VAV baseline and by up to 30 kW (82%) compared to the CCHP. The GHP also reduced annual energy costs by 71% relative to the baseline and by 41% relative to the CCHP/DOAS. Despite lower than expected part-load performance, GHP life-cycle costs were 4% lower than CCHP and 37% lower than the baseline.

## IMPLEMENTATION ISSUES

Both VRF systems improved comfort and provided significant energy savings, lower peak electric demand, and potential savings in life-cycle costs compared to conventional VAV systems. The demonstration also identified operational issues for both VRF technologies in this application. The electric CCHP was unable to meet the heating load for several days when ambient temperatures dropped below winter design conditions indicating a need for supplemental heating for this climate (ASHRAE Zone 5). The demonstration also highlighted the low range of part-load operation for VRF heat pumps, even when sized appropriately. Very low part-load operation adversely impacted both heat pumps; however, this specific GHP model had lower than expected performance at part-load operation. While not inherent in this class of technology, this may be due to product-specific controls or equipment sizing. These results suggest additional development is needed to optimize GHP performance and to reduce installed costs in order to improve regional economics and support broader market adoption.

## PUBLICATIONS

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# EXECUTIVE SUMMARY

## INTRODUCTION

Variable refrigerant flow (VRF) heat pump systems are increasingly used in small commercial buildings in the U.S. as a high efficiency heating and cooling option for multi-zone applications. However, the complexity of VRF configurations and the customized design for specific buildings make it difficult to monitor and predict energy savings relative to baseline HVAC systems. Due to limited field data available for VRF systems, especially in colder climates, energy savings are often based on energy modeling or data from controlled laboratory testing. This ESTCP demonstration provided a unique opportunity to directly compare measured performance data for two VRF heat pump technologies to the baseline variable-air-volume (VAV) system and determine the potential energy and economic benefits for DoD facilities.

This field study was a side-by-side demonstration of two VRF heat pump technologies that offer significant potential for energy and cost savings, and improved comfort with zoned temperature control. One VRF system was a natural gas engine-driven heat pump (GHP) - an emerging technology designed to reduce peak electric demand and generate savings in both annual energy costs and life-cycle costs compared to conventional equipment. The second VRF system was an electric cold climate heat pump (CCHP) - a relatively mature technology, designed for colder ambient conditions without supplemental heating.

## OBJECTIVES

The objective of this demonstration was to quantify the energy savings and economic benefits of two VRF heat pump technologies for cold climate DoD applications based on measured performance data. The VRF performance was also compared to the measured baseline performance of the VAV system for the same building. This evaluation compared peak electric demand, site and primary energy use and full-fuel-cycle GHG emissions for each type of equipment. The economic assessment determined annual energy and life-cycle costs based on measured energy use data, installed costs, and local utility rates. Qualitative benefits, such as reliability and comfort were also explored. The overarching goal was to identify the potential benefits of VRF systems for DoD facilities as well as other commercial markets.

## TECHNOLOGY DESCRIPTION

The overall GHP design is similar to an electric heat pump but with an advanced natural gas engine in place of an electric motor (Figure 1). The NextAire™ GHP uses high efficiency scroll compressors and a variable speed engine with a demonstrated long life (30,000 hours). GHPs combine high efficiency heating (rated at 1.2-1.4 coefficient of performance [COP]) and cooling (0.95-1.2 COP). During cooling, GHPs consume natural gas in place of electricity, significantly reducing peak electric demand in comparison to electric chillers or electric heat pumps. During heating, GHPs are rated 50% more efficient than standard gas furnaces or boilers commonly used at DoD facilities. Heat recovered from the engine cooling jacket and exhaust can supplement the GHP output during heating mode to increase the overall system efficiency. Heat recovery also allows GHPs to deliver a higher supply temperature at cold ambient conditions. In contrast, electric heat pumps often require inefficient resistance heating to supplement their heating capacity at low outdoor temperatures.

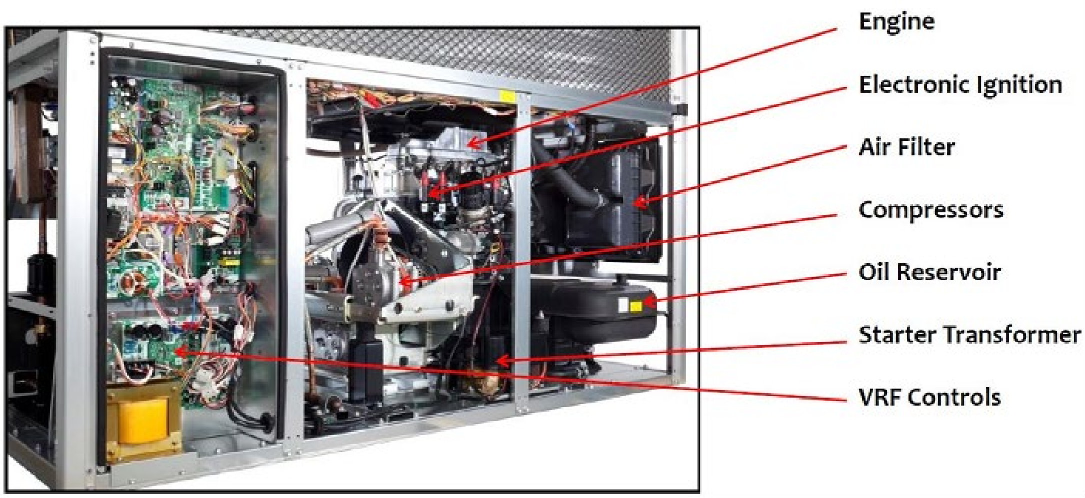


**Figure 1. The GHP Utilizes Two Scroll Compressors and an Advanced Natural Gas Engine**

Electric VRF heat pumps, such as the CCHP unit demonstrated in this project, are a mature technology with several U.S. manufacturers and a modest but growing market. VRF systems are increasingly used for multi-zone commercial buildings, driven by the potential for energy savings, economic benefits, and improved comfort with zoned temperature control. Electric VRF systems typically use variable-speed electric motors to drive variable-speed or multi-stage compressors and a single refrigerant circuit with individually controlled fan coils to provide zoned heating and cooling (Figure 2). Both the gas and electric heat pumps featured in this demonstration used the same type of indoor VRF fan coil units and controllers provided by the same manufacturer.

Studies report energy savings up to 30% compared to conventional HVAC systems [Thorton]; however, energy savings are typically based on manufacturer data and modeled simulations. Due to the custom nature of VRF installations, direct comparisons of energy use and economics can be difficult to quantify.

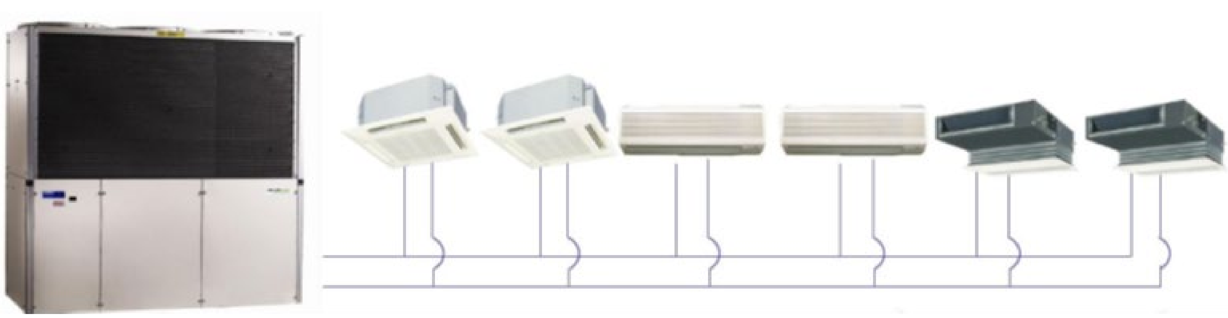

**Figure 2. GHP System Utilize the Same Indoor VRF Air Handlers as the Electric VRF System**

In addition, the performance of all air source heat pumps (both electric and gas engine-drive) varies significantly with ambient temperatures, so performance and energy savings for one climate will not be the same as a different climate. VRFs are primarily installed in moderate or hot climates that can benefit from their high cooling efficiency. In colder climates, VRFs are often installed in heated mechanical rooms or with backup electric resistance heaters, thus increasing installed costs and reducing energy savings [Swanson, Schutter]. Recently some manufacturers have introduced cold climate versions of electric VRF systems without supplemental heating; but limited field data is available to validate their performance. This demonstration provided measured field data needed to validate modeled energy savings for VRF systems, especially in colder climates.

## PERFORMANCE ASSESSMENT

The field site was a small multi-zone office building at Naval Station Great Lakes (NSGL) in North Chicago, Illinois. The building was divided into two thermal zones, one served by the GHP and the other served by the electric CCHP. Since VRF systems typically do not provide ventilation, these were paired with a single dedicated outdoor air system (DOAS) sized to meet ventilation requirements and deliver supply air at space neutral conditions. Prior to the VRF installation, GTI conducted 12 months of baseline monitoring of the existing VAV system to characterize its performance across the full range of operating conditions. VRF heat pump performance data was then collected during the following year.

Measured energy use data was weather-normalized. Energy use for each VRF heat pump system was also normalized to the total building load to allow for a direct comparison to the baseline VAV system. This controlled for changes in on-site routines or activities over the course of the baseline and VRF system monitoring. GHP performance metrics were directly compared to CCHP to evaluate the performance objectives. Energy consumption for both VRF systems was also compared to the baseline. Site and primary energy, full-fuel-cycle GHG emissions, annual energy costs and life-cycle costs were calculated based on normalized energy use.

### *Baseline Characterization*

During heating operation, the baseline VAV electric consumption was found to be higher than expected. Without a building automation system (BAS) for integrated controls, the central VAV gas heating and distributed VAV-boxes operated independently resulting in excessive electric resistance heating by the VAV boxes and a higher peak electric demand. Although the baseline system did not operate as designed, this may be typical VAV operation for smaller buildings or sites without a central BAS.

### *Measured Performance Objectives*

Peak Electric Demand Reduction

Both VRF systems significantly reduced peak electric demand by eliminating the electric resistance trim heating in the winter and the summer overcooling/electric reheat approach used in VAV systems (Table 1).

**Table 1. Peak Electric Demand Reduction**

| | Savings over Baseline VAV | Savings over CCHP/DOAS |
|---|---|---|
| GHP/DOAS – Heating Season | 90% (59 kW) | 82% (30 kW) |
| GHP/DOAS – Cooling Season | 79% (34 kW) | 36% (5 kW) |

For typical buildings, the highest electric demand occurs during the summer cooling operation due to conventional electric air conditioning; however, electric heating can create a secondary winter peak. This demonstration illustrates how the growing use of electric heat pumps can increase the winter peak electric demand, potentially exceeding the summer cooling peak especially in cold climates.

Savings in Life-cycle Costs

The NIST Building Life-cycle Cost program was used to determine the economic benefits for each technology. The GHP/DOAS met the performance objective for savings in both annual energy costs and life-cycle costs compared to the CCHP/DOAS. Based on measured data, the GHP/DOAS reduced energy costs by 41% compared to the CCHP/DOAS and by 71% compared to the baseline. Despite lower than expected part-load performance, GHP/DOAS life-cycle costs were 4% lower than CCHP/DOAS and 29% lower than the VAV baseline.

Primary Energy Savings

Primary energy use (i.e., full-fuel-cycle) accounts for all upstream energy used to generate power or to supply fuel to the building meter. In other words, primary energy includes all energy for fuel extraction (natural gas, oil, coal, renewables), conversion (e.g., power generation), and distribution (pipelines, power transmission lines). Primary energy is a more comprehensive approach to evaluate energy use and is more relevant to energy security for DoD facilities than the energy metered at the site. A growing number of codes and standards are shifting to the use of full-fuel-cycle or primary energy use as the metric to quantify the environmental impact of appliances, rather than site energy. For this analysis, primary energy was calculated based on measured energy use and primary energy factors for electricity and natural gas. Both VRF systems had significantly lower natural gas and electricity use compared to the baseline VAV, reducing primary energy use by about 57%. This validates published VRF modeling studies reporting savings from 20% to 60% relative to VAV systems [EES Consulting]. Primary energy use for the GHP/DOAS did not meet the performance objective of 10% savings relative to the CCHP/DOAS due the GHP/DOAS lower than expected part-load performance.

Reduction in Full-Fuel-Cycle GHG Emissions

Full-fuel-cycle GHG emissions were calculated based on estimated annual energy use and regional CO2e emission factors to estimate upstream emissions from power generation, transmission, or distribution of the energy to the building meter. Emission factors for electricity were based on published non-baseload power generation mixes for eGrid regions.

Natural gas emission factors were also based on the published regional average [SEEAT]. Both VRF systems significantly reduced full-fuel-cycle GHG emissions by 55%-63% compared to the baseline VAV system. GHP/DOAS full-fuel-cycle GHG emissions were 13% lower than CCHP/DOAS but did not meet the performance objective of 20% due to lower than expected part-load performance. In addition, GHP engine exhaust emissions were not included due to lack of available data but would increase the total full-fuel-cycle GHG.

Comfort and Reliability

Comfort and reliability were qualitative performance objectives to assess the impact of VRF systems on the multi-zone conditioned space and its ability to meeting space conditioning loads over a wide range of real-life conditions. VRF heat pump configurations provide zoned space conditioning and are expected to improve comfort as well as energy efficiency. Measured data showed both VRF heat pumps successfully maintained zone temperatures within +/-2°F of the setpoint during typical operation. Feedback from the field site also confirmed claims of improved comfort. With the baseline VAV system, the building tenants complained about low temperatures in the winter and used electric space heaters to address comfort issues. The field site chose to keep the demonstration VRF systems at the conclusion of the field study.

As an emerging technology, heat pumps must prove to be as reliable as conventional HVAC equipment to gain market acceptance. The GHP/DOAS did not meet this success criteria due to multiple installation/component issues and reduced performance which impacted the total delivered heating capacity. As shown in Figure 3, the CCHP also failed to meet the heating load at very cold ambient conditions, indicating a need for supplemental heat for this climate (ASHRAE Zone 5).

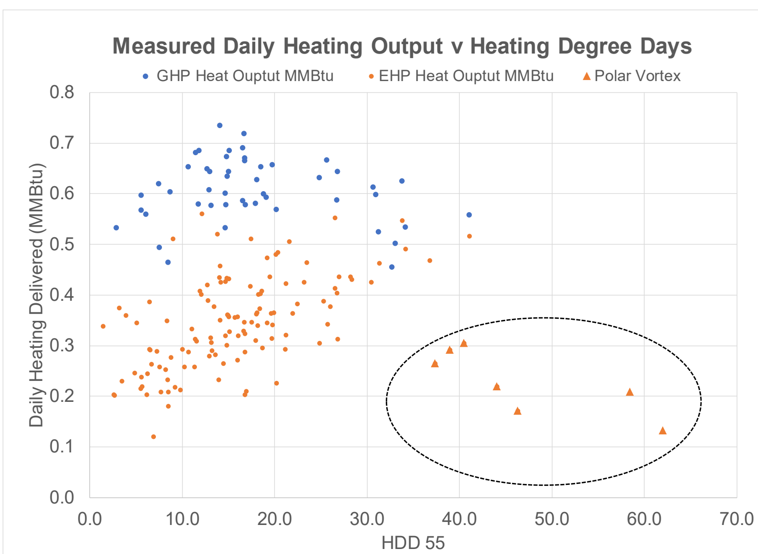


**Figure 3. CCHP Was Not Able to Meet the Heating Load at Extreme Cold Temperatures**

## REGIONAL PERFORMANCE ASSESSMENT WITH ENERGY MODELING

To apply these demonstration results to the broader DoD community, GTI conducted hourly simulations using EnergyPlus models to predict the energy savings and economic benefits of the VRF systems for various climates. EnergyPlus models of DOE reference buildings (small office) were modified to match the layout and floorspace of the field site. EnergyPlus equipment models were adapted to measured field data from the demonstration and optimized to be more representative of typical HVAC operation. The VAV baseline was modeled with integrated controls (such as would be seen with a BAS), thus reducing the higher than expected electric resistance heat discovered during baseline monitoring. Hourly energy simulations were conducted for five different climates to predict regional energy and economic benefits for the baseline and VRF systems. The following sections address each performance objective based on modeled energy performance.

Modeled results for different regions show GHP/DOAS had the lowest peak electric demand for heating or cooling (Figure 4). The magnitude of demand reduction was higher for colder climates (Chicago, Richmond) than warmer climates. Both VRF systems reduced peak electric demand compared to the baseline due to elimination of electric resistance heating. For colder climates, CCHP/DOAS had a higher winter peak electric demand than summer. Note that winter peak demand for CCHP/DOAS would increase if electric resistance heat was used to provide the necessary supplemental heating at low ambient temperatures.

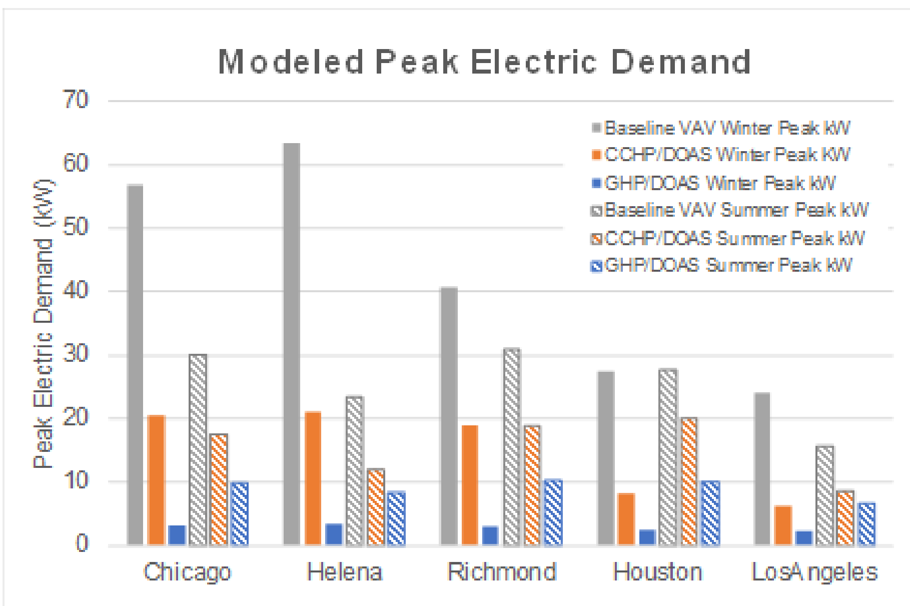


**Figure 4. Regional Modeled Comparisons for Peak Electric Demand**

Primary energy was calculated using modeled energy use and regional primary energy factors based on non-baseload electricity mix per eGrid 2016. Modeled results show significant savings in primary energy for both VRF systems compared to the baseline for all regions (Figure 5). Primary energy savings for the VRF/DOAS ranged from 47% to 62% compared to baseline.

GHP/DOAS primary energy savings were slightly higher than the CCHP/DOAS in colder climates, and slightly lower in warmer regions.

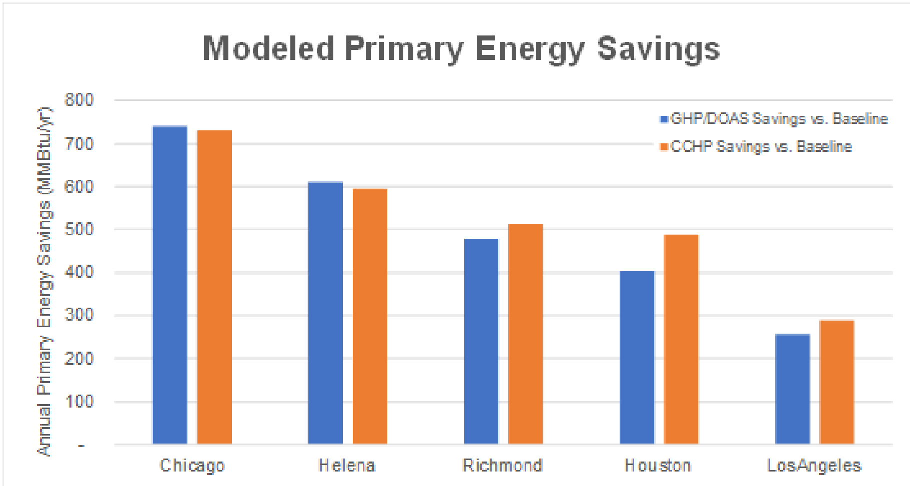


**Figure 5 Regional Modeled Comparisons for Primary Energy Savings**

Regional GHG emission factors were used to estimate full-fuel-cycle GHG emissions based on non-baseload electricity mix per eGrid 2016. Modeled results show significant savings in full-fuel-cycle GHG for both VRF systems compared to the baseline VAV for all regions (Figure 6). VRF/DOAS generated savings in GHG emissions from 51% to 62% compared to baseline. For the GHP/DOAS, GHG savings were slightly higher than the CCHP/DOAS in colder climates, and slightly lower in warmer regions.

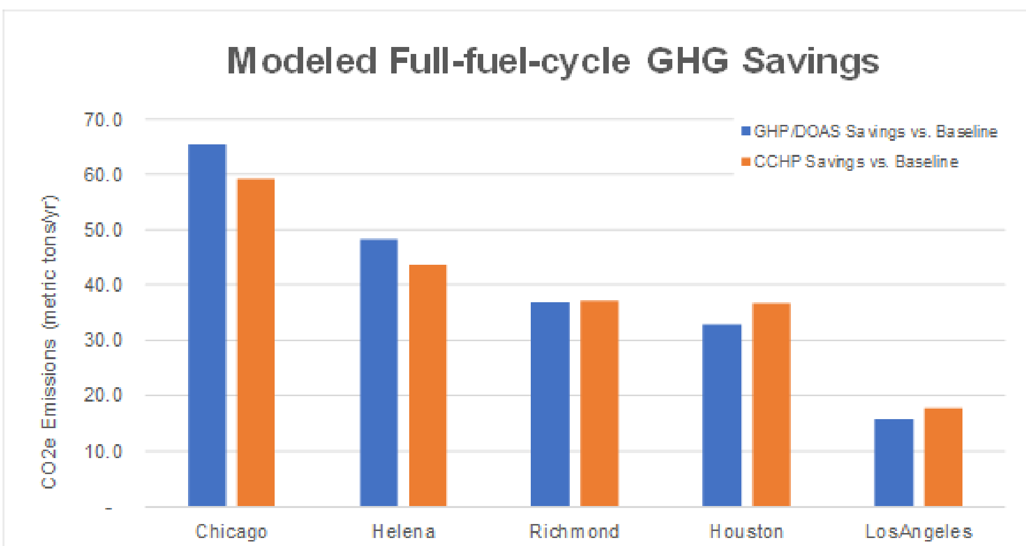


**Figure 6 Regional Modeled Comparisons for Full-fuel-cycle GHG Emissions**

## COST ASSESSMENT

The NIST Building Life-cycle Cost program was used for the life-cycle cost assessment for each technology. The cost assessment included actual equipment costs and installation costs based on rule-of-thumb or RS Means 2016 estimates. Aside from the engine maintenance for the GHP, it was assumed that maintenance costs were similar for the baseline VAV and both VRF systems. To address customers' concern about unexpected service costs for the GHP engine maintenance, one manufacturer offers a service agreement included with the equipment purchase. This scheduled maintenance plan provides engine maintenance (oil change, belts, etc.) at intervals of 1,000, 2,000, and 3,000 runtime hours. This option was selected for the demonstration and also used for the cost assessment.

Annual energy costs were based on the incremental composite rates provided by the field site. Utility demand charges and energy rate structures can vary widely by state and can impact energy cost savings. As expected, modeled energy savings were lower than measured savings due to a more optimized VAV operation baseline with less resistance heat operation. The efficiency of the VRF heat pumps (CCHP: 12.3 EER/2.3 COP; GHP: 1.3 COPg heating/cooling) exceeds the baseline VAV efficiency (9.5 EER/80%) but only covers a portion of the heating and cooling load, as the ventilation load is met by the standard efficiency DOAS (11.3 EER/80%).

Modeled results shown in Figure 7 indicate that VRF systems reduced annual costs in all regions. GHP/DOAS had the highest savings in annual energy costs for cold climates with lower savings for moderate and hot climates with high cooling loads (i.e., Houston and Los Angeles). Annual energy costs for CCHP/DOAS would increase if electric resistance heat was used to provide the necessary supplemental heating at low ambient temperatures.

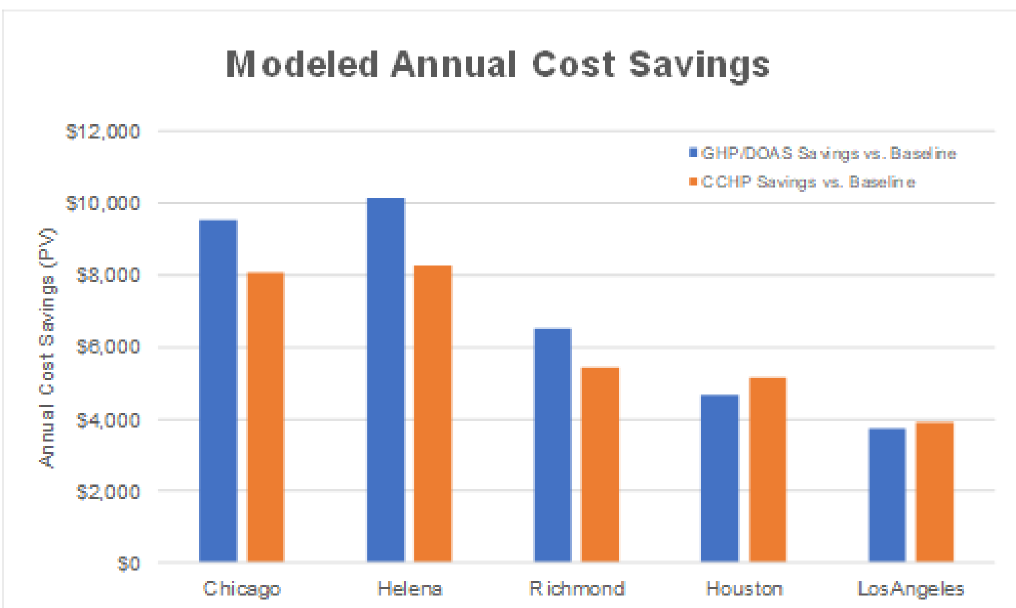


**Figure 7 Regional Modeled Comparisons for Annual Cost Saving (PV)**

Based on energy pricing for the field site, modeled results show both VRF systems reduced life cycle costs compared to the baseline VAV system in most climates. Savings in life-cycle costs were highest in cold climates (e.g. Chicago, Helena). The GHP/DOAS had savings in life-cycle costs for all locations except Los Angeles (Figure 8). Modeled results indicate the CCHP/DOAS has potential to reduce life-cycle costs in all locations. As previously discussed, life-cycle costs for CCHP/DOAS would significantly increase if supplemental heating was added for operating at low ambient temperatures.

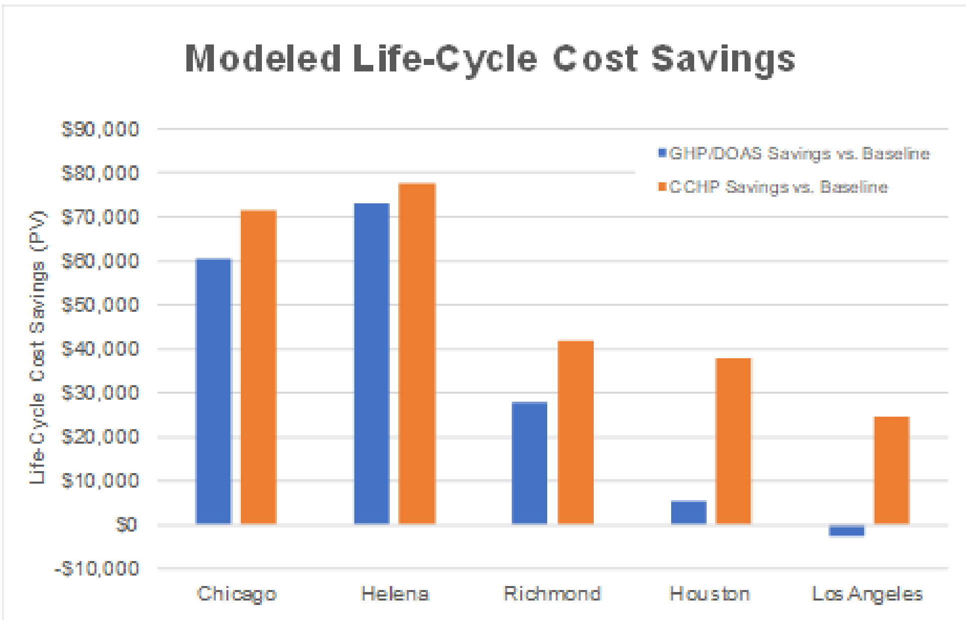


**Figure 8. Regional Modeled Comparisons for Savings in Life-Cycle Costs**

Modeled results in Table 2 compare GHP/DOAS economics to the baseline VAV. The GHP/DOAS was cost-effective in cold climates (e.g. Chicago, Helena) with simple paybacks of 5 years. Although the GHP/DOAS reduced annual costs in all climates, additional development to optimize performance and reduce first costs is needed to achieve sufficient payback in warmer climates.

**Table 2. Life-cycle Assessment of GHP/DOAS with respect to Baseline VAV based on Modeled Data**

| | Equipment Premium | First Cost Premium | Annual Savings (PV) | Life-Cycle Cost Savings (PV) | | Simple Payback (years) | SIR |
|---|---|---|---|---|---|---|---|
| Chicago | $32,000 | $47,000 | $8,970 | $60,640 | 18% | 5 | 2.3 |
| Helena | $32,000 | $47,000 | $10,015 | $73,181 | 22% | 5 | 2.6 |
| Richmond | $32,000 | $47,000 | $6,229 | $27,747 | 8% | 8 | 1.6 |
| Houston | $32,000 | $47,000 | $4,376 | $5,513 | 2% | 11 | 1.1 |
| Los Angeles | $32,000 | $47,000 | $3,710 | ($2,485) | -1% | Not Achieved | 1.0 |

## IMPLEMENTATION ISSUES

Several regulations apply to the use of VRF systems for DoD facilities. Best Practices for Variable Refrigerant Flow (VRF) System is summarized in the Unified Facilities Criteria UFC 3-410-01 Heating, Ventilating, and Air Conditioning Systems. Requirements for VRF systems are addressed in detail in Appendix D of the full report.

This demonstration presented several important findings and valuable lessons learned. Baseline monitoring highlighted operational issues with the existing baseline VAV system. As confirmed by the manufacturer, a BAS is required to integrate controls for the central gas heating and distributed VAV-boxes. Without a BAS at the field site, the central VAV gas heating and the indoor VAV-boxes operated independently resulting in excess electric resistance heating especially during temperature recovery from night/weekend setback. As a result, the baseline operation had higher than expected peak electric demand and higher energy costs. Although the baseline system did not operate as designed, this may be a typical situation for smaller buildings or sites without a central BAS. Since VAV systems are widely used for multi-zone applications, such as office buildings, retro-commissioning and/or retrofitting integrated controls may improve efficiency and reduce energy costs for existing equipment.

Based on this demonstration, VRF systems improved comfort while providing significant energy savings, lower peak electric demand, and lower annual costs compared to conventional VAV systems. Both CCHP and GHP systems have the potential to reduce life-cycle costs depending on regional heating/cooling loads and energy pricing. The demonstration also identified operational issues and limitations for both VRF technologies in this application. The field site experienced a number of outages of both VRF equipment and conventional HVAC equipment (e.g. the DOAS igniter). Several outages were due to equipment installation issues, highlighting the importance of well-trained service providers, a common concern for emerging technologies.

For this demonstration in ASHRAE Climate Zone 5, the electric CCHP was unable to meet the heating load for several days. This specific VRF heat pump model was designed for cold climate applications without supplemental heating. For this demonstration, the CCHP was oversized based on the de-rated heating capacity at low temperatures to match the estimated peak heating load. During the monitoring period, ambient temperatures dropped below winter design conditions on several occasions. During extreme cold, the CCHP continued to function but was unable to maintain zone temperatures and operated at very low efficiencies. As weather conditions become less predictable with more extreme temperatures, the CCHP will require a means of supplemental heating for reliable cold climate operation. The use of supplemental heat will impact both energy use and life-cycle costs.

As a key finding, this demonstration highlights how VRF heat pumps regularly operate at very low part-loads even when sized appropriately. This is amplified when paired with a DOAS which can reduce the facility heating or cooling loads. Currently, GHP manufacturer rating is based on full-load capacity and efficiency at select rating conditions; however, depending on the climate, the heat pump might never operate at those specific conditions. By design, VRF systems usually operate at lower part-loads by modulating to meet the multi-zone heating and cooling loads. Existing GHP performance ratings are not a good indicator of seasonal performance.

Current efforts to update GHP standards, developed prior to the introduction of VRF products, will include performance metrics that reflect actual installed conditions, including part-load operation. This evolution of GHP performance standards will help support the development of more optimized designs.

Part-load operation adversely impacted the performance of both heat pumps; however, the specific GHP model used in the demonstration had much lower than expected performance at low part-load operation. This extent of decreased part-load performance is not inherent in this class of technology and could be due to product-specific controls or engine sizing. GHPs use variable-speed engines to enable them to closely follow the load and maintain efficiency. This is an important finding and warrants further investigation to optimize part-load performance.

In summary, both VRF heat pumps demonstrated improved comfort along with significant potential for energy savings and economic benefits compared to conventional HVAC. While this demonstration compared an emerging technology (GHP) with a more mature technology (CCHP) that has multiple manufacturers and decades of design optimization, both VRF heat pumps had operational limitations in this cold climate application. Results further suggest the need for additional research and development to optimize GHP performance and reduce installed costs in order to improve regional economics and support broader market adoption.

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# 1.0 INTRODUCTION

Facility energy costs for the United States Department of Defense (DoD) are significant, exceeding $4 billion (FY11) and representing 21% of total energy costs. This energy use also contributes to a disproportionate share of greenhouse gas (GHG) emissions. To address these issues, the Environmental Security Technology Certification Program (ESTCP) conducts a demonstration and validation program for environmental technologies at DoD facilities to document improved efficiency, reduced liability, improved environmental outcomes, and cost savings. The ESTCP is designed to achieve the overall goals of reducing energy usage and intensity and improving energy security.

This demonstration presents a side-by-side comparison of a GHP variable refrigerant flow (VRF) system and an electric cold climate heat pump (CCHP) VRF system installed at a small office building at Naval Station Great Lakes (NSGL) in Illinois. Gas engine-driven heat pumps (GHPs) are an emerging technology for space conditioning that offers a cost-effective option to reduce peak electric demand and life-cycle costs savings compared to conventional equipment or electric air-source heat pumps. On-going GHP developments, such as black start capability, have potential to improve reliability for critical operations, and reduce the dependency of DoD facilities on the commercial electric infrastructure.

**Figure 9. NextAire™ Multi-Zone**

This demonstration provides a unique opportunity to directly compare measured performance data for a VRF system to the baseline variable-air-volume (VAV) system for the same building. Heat pump variable refrigerant flow (VRF) systems are increasingly used in U.S. small commercial buildings in response to the demand for cost-effective energy-efficient heating and cooling for multi-zone applications. The complexity and customized design of VRF systems for specific buildings make it difficult to predict energy savings relative to baseline HVAC systems.

Due to limited field data available for VRF systems, especially in colder climates, energy savings are often based on energy modeling or laboratory data obtained under controlled conditions.

## 1.1 BACKGROUND

The NextAire™ GHP was designed and marketed as a gas cooling option and successfully demonstrated in the hot/dry climates. Multiple GHP installations demonstrated the benefits of gas cooling, including significant reductions in peak electric demand, reduced operating costs, and savings in water use. The latest version of the NextAire™ Multi-Zone GHP, the Model E, was commercialized in 2013 and can be adapted for any U.S. climate. For heating-dominated and lower life-cycle costs compared to conventional HVAC equipment or electric heat pumps.

## 1.2 OBJECTIVE OF THE DEMONSTRATION

The objective of this demonstration was to determine the relative energy and economic benefits of two VRF technologies for cold climate DoD applications. This demonstration provides measured performance data for a side-by-side comparison of a GHP and an electric CCHP at a small office building at NSGL. This project also provides a direct comparison of measured VRF performance data for both heat pump systems to the baseline variable-air-volume (VAV) performance for the same building.

Performance data was collected for a full calendar year to determine annual energy use and performance. The economic analysis incorporated energy use data and local utility rates to estimate building specific payback and life-cycle costs for each heat pump technology at the demonstration site. The demonstration evaluated the hypothesis that GHP will reduce peak electric demand, primary energy, GHG emissions and life-cycle costs as compared to an electric CCHP for space conditioning DoD facilities in a cold climate.

The overarching goal of the demonstration is to support the broader adoption of VRF systems in both government and commercial markets. Energy modeling based on measured performance data was used to predict the relative energy and economic benefits of GHPs, electric CCHPs, and packaged rooftop units across the range of DoD locations and climates. The results of the demonstration and lessons learned are summarized in a Best Practice Guideline to disseminate these findings to the wider DoD user community. The guideline also discusses any potential barriers or implementation issues along with recommendations for addressing these barriers.

## 1.3 REGULATORY DRIVERS

The implementation of GHP technology for space conditioning of DoD facilities addresses several regulations, Executive Orders, DoD directives, including but not limited to those listed below. GHP technology can improve energy efficiency and reduce building energy intensity as compared to conventional HVAC. In addition, GHPs are air-cooled resulting in significant water savings compared to water-cooled electric chillers. This demonstration quantified the energy savings of a GHP unit with respect to a conventional packaged rooftop and an electric cold climate heat pump. GHPs significantly reduce peak electric demand and total electricity use reducing the dependency of DoD facilities on the grid. The demonstration also estimated potential reduction in primary energy and full-fuel-cycle greenhouse gas emissions.

- Executive Order 13834, *Efficient Federal Operations*: GHP reduce building energy use annually and implement cost-saving energy efficiency measures.
- Executive Order 13693, March 19, 2015, *Planning for Federal Sustainability in the Next Decade*: GHP installations are a viable option for demand side management and demonstrated potential to reduce life-cycle costs, lower building energy intensity and reduce full-fuel-cycle GHG emissions.
- *Energy Policy Act of 2005*: This demonstration meets the goals for conducting "…programs of energy efficiency research, development, demonstration, and commercial application, including: (2) cost-effective technologies, for new construction and retrofit, to improve the energy efficiency and environmental performance of buildings, using a whole-buildings approach…" (Title IX, Subtitle A, Sec. 911)
- *Energy Independence and Security Act of 2007*: The GHP demonstration addresses the following sections of this act. GHPs qualify for additional LEED points for green building ratings and increase energy efficiency compared to conventional HVAC.
  - Sec. 421 Commercial high-performance green building
  - Sec. 433. Federal building energy efficiency performance standards
  - Sec. 436. High-performance green Federal buildings
  - Sec. 437. Federal green building performance
  - Sec. 441. Public building life-cycle costs
- Federal Leadership in High Performance and Sustainable Buildings MOU 2006, *Guiding Principles for Federal Leadership in High Performance and Sustainable Buildings*: GHPs are a viable option to reduce energy costs and optimize energy performance through energy efficiency improvements and water conservation.

DoD Policy:

- *Strategic Sustainability Performance Plan*, Energy Security MOU with DOE: GHP technology reduces electric demand through the use of natural gas cooling in place of electric technology. Previous GHP installations have demonstrated reductions in long-term energy costs. Current natural gas prices are historically low and are expected to remain stable due to abundant U.S. supplies. This reduces the impact of energy price fluctuations. Future development of black start GHP technology can increase the resiliency of DoD facilities by providing heating and cooling during power outages.
- Department of Defense Strategic Sustainability Performance Plan FY 2014: GHPs can reduce the energy intensity of DoD facilities by decreasing the use of fossil fuels through high efficiency heating and cooling.

Guides:

- National Institute of Building Sciences, *Whole Building Design Guide*
- Federal High Performance and Sustainable Buildings

Specifications:

- Standard 15-2019 (packaged w/ Standard 34-2019): Safety Standard for Refrigeration Systems and Designation and Safety Classification of Refrigerants (ANSI Approved)
- ANSI/ASHRAE Standard 62.1-2016: Ventilation for Acceptable Indoor Air Quality
- ANSI Z21.40.4/CGA 2.94 (2014): Performance Testing and Rating of Gas-fired Air Conditioning and Heat Pump Appliances
- ANSI/AHRI Standard 1230 (2010): Standard for Performance Rating of Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment.
- Standard 90.1-2016: Energy Standard for Buildings Except Low-Rise Residential Buildings
- ASHRAE 2016 Handbook HVAC Systems and Equipment
- U.S. Green Building Council *Leadership in Energy & Environmental Design (LEED)*: GHPs may qualify for up to 32 points

During this project, after the VRF installation at the demonstration site, revisions were issued to the Unified Facilities Criteria UFC 3-410-01 *Heating, Ventilating, and Air Conditioning System* presenting additional design considerations regarding the use of VRF in DoD facilities. These considerations are addressed individually in Appendix D.

## 2.0 TECHNOLOGY DESCRIPTION

### 2.1 TECHNOLOGY OVERVIEW

#### 2.1.1 Natural gas engine-driven heat pumps

GHP design is similar to an electric heat pump but utilizes an advanced natural gas engine in place of an electric motor. The NextAire™ Multi-Zone GHPs combine high efficiency scroll compressors and an Aisin/Toyota engine with a demonstrated long life (30,000 hours). The units have a maintenance interval of 6,000 to 10,000 hours. Variable-speed engine controls allow the GHP to more closely follow the load and maintain efficiency. The Multi-Zone GHP is equipped with an electronic speed controller that regulates the revolution rate of the engine by taking both indoor and outdoor temperatures into account. This results in minimal energy loss and a constant indoor temperature when **compared** to traditional electric heat pumps.

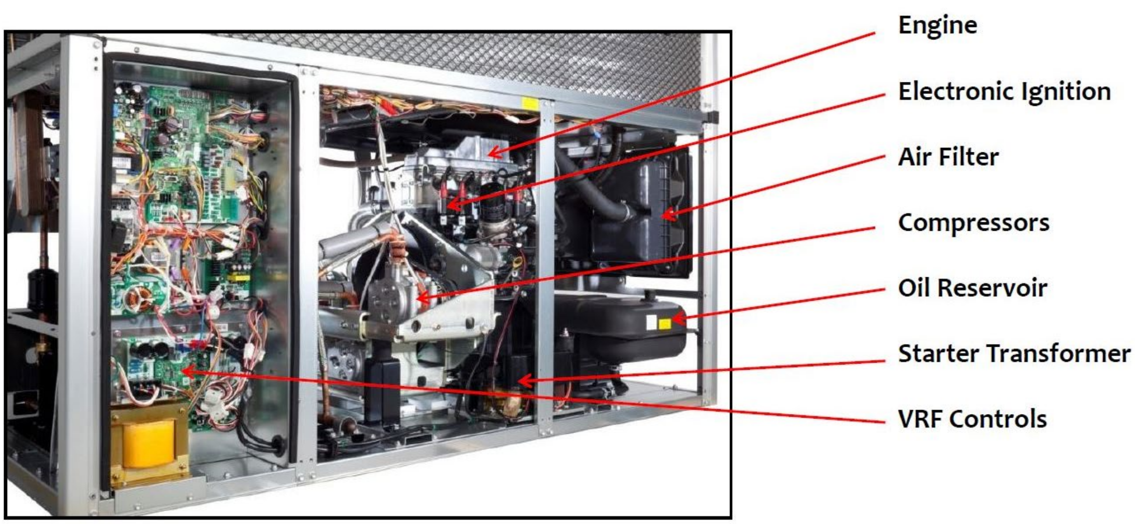



**Figure 10. NextAire™ GHP Utilizes Two Scroll Compressors with an Aisin/Toyota Engine**

During heating and cooling operation of the NextAire VRV GHP, engine coolant is circulated throughout the system by a coolant pump. Warm coolant is pumped through an exhaust air heat exchanger where its temperature is raised a few degrees by heat recovered from the engine exhaust. The coolant then flows to the water-cooled exhaust manifold located on the internal combustion engine, where its temperature is increased more. Finally, the coolant enters the internal combustion engine block where it removes even more heat directly from the engine. In heating mode, the recovered waste heat is utilized for heating the refrigerant to increase its evaporating and suction temperatures. The system uses a plate frame-type heat exchanger to transfer heat from the coolant to the evaporating side of the refrigerant system. Based on input from sensors (ambient conditions, return air temperatures, amount of refrigerant superheat, etc.), the flow of the recovered waste heat is controlled by electronic valves to maximize the heat recovery process and avoid a defrosting cycle. In cooling and moderate heating mode conditions, engine waste heat can also be directed to an optional hot water heat exchanger system commercially available through a third-party vendor.

While natural gas is a clean burning fuel, the primary environmental concern for natural gas engines is NOx emissions. Currently, there are no emission standard for gas engine-driven heat pumps. The NextAire 15-ton GHP engine, rated at 23.75 HP, is exempt from the EPA emission standard for stationary engines over 25 HP (19kW) (EPA 40CFR 1068.215) and can be installed at any location in the U.S. Likewise, SCAQMD does not regulate small stationary engines under 50 HP. Future developments of larger engines (i.e. GHPs over 15 tons) may require gas after-treatment for EPA non-attainment areas.

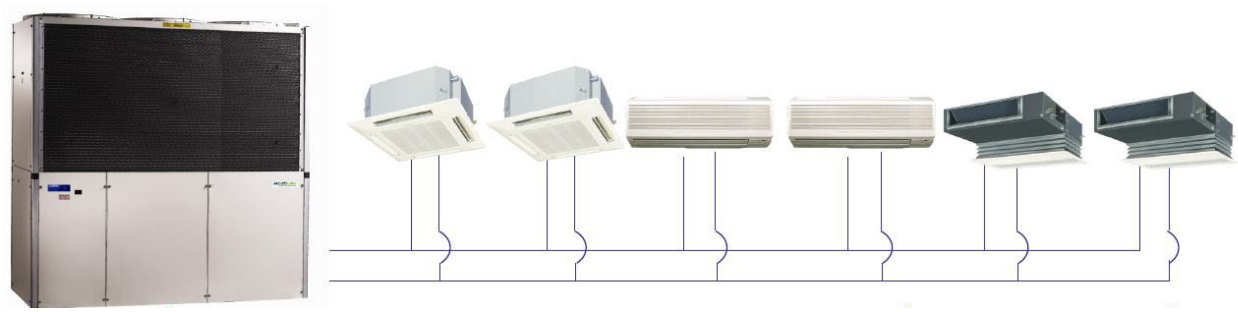


**Figure 11. GHP Systems Utilize the Same VRF Air Handlers as Electric Heat Pump Systems**

### 2.1.2 Variable Refrigerant Flow (VRF) Systems

Both gas and electric heat pumps featured in this demonstration are variable refrigerant flow (VRF) systems. VRF systems typically use variable speed or multi-stage compressors and a single refrigerant circuit with individually controlled fan coils to provide zoned heating and cooling. VRF systems typically do not provide ventilation and are often paired with DOAS which is sized to meet the ventilation requirements and deliver supply air at space neutral conditions.

Electric VRF heat pumps are a mature technology with several U.S. manufacturers, but have yet to achieve the 30%-50% market share VRFs have in Asia and Europe. VRF systems are increasingly used for multi-zone commercial buildings, driven by the potential for energy savings, economic benefits, and improved comfort with zoned temperature control. Studies report energy savings up to 30% compared to conventional HVAC systems [Thorton]; however, energy savings are typically based on manufacturer data and modeled simulations. Due to the custom nature of VRF installations, direct comparisons of energy use and economics can be difficult to quantify. Additional field studies are needed to validate modeled energy savings for VRF systems, especially in colder climates.

In addition, the performance of all air source heat pumps (both electric and gas engine-drive) varies significantly with ambient temperatures, so performance and energy savings for one climate will not be the same as a different climate. VRFs are mainly installed in warm climates that benefit from their high cooling efficiency. In colder climates, VRFs are often installed in heated mechanical rooms or with backup electric resistance heaters increasing installed costs and reducing energy savings. [Swanson, Schutter]. Recently some manufacturers have introduced cold climate versions of electric VRF systems without supplemental heating; but limited field data is available to validate their performance.

## 2.2 TECHNOLOGY DEVELOPMENT

While GHPs have a significant share of the Japanese and European space conditioning markets, these products have only recently been introduced to the U.S. GHPs are an emerging technology successfully demonstrated in warmer climates, but still under-utilized and unfamiliar to building owners, contractors, and engineering firms. The NextAire™ Multi-Zone VRF GHP (Model D), manufactured by Aisin World Corporation and distributed by ICE, was introduced in 2009. The NextAire Multi-Zone GHPs included 8-ton and 15-ton units to provide zoned heating and cooling for up to 17 zones and 33 zones, respectively. Units could be combined for larger installations up to 300+ tons. NextAire™ GHPs were tested extensively in the hot/dry climate of the Southwestern United States by federal agencies and gas utilities.

The latest version of the NextAire™ Multi-Zone VRF GHP, the Model E, was certified for the U.S. market in July 2013 and included a cold region option for installation colder climates. The 8-ton Model E was used in this demonstration. Currently over 800 NextAire™ GHPs have been installed in the U.S, predominantly in warmer climates.

The ICE product line also includes an 11-ton Packaged GHP, certified in 2011, for rooftop installation with similar high efficiency ratings (18 SEER). ICE also recently developed a 5-ton residential GHP. In 2018, ICE became Blue Mountain Energy (BME) and is partnering with Mestex while continuing to distribute the Aisin NextAire™ Multi-Zone GHP as a product under the Sierra Fresh Aire Systems brand.

A second manufacturer, Yanmar, introduced its GHP product line to the U.S. in 2016. Yanmar offers a range of VRF systems, from 8- to 14-tons. These products, as well as ICE's GHPs, are 2-pipe systems that can operate either in cooling or heating mode. Yanmar also offers a 14-ton GHP with a 3-pipe heat recovery VRF system that provides simultaneous heating and cooling.

## 2.3 ADVANTAGES AND LIMITATIONS OF THE TECHNOLOGY

GHPs offer some advantages compared to electric air-source heat pumps and conventional HVAC equipment. Descriptions of these advantages as well as limitations of both heat pump technologies are described below.

<u>High Efficiency</u>: Manufacturer specifications for GHPs indicate high efficiency heating (1.2-1.4 COP) and cooling (0.95-1.2 COP) at rating conditions, exceeding the heating efficiency (80%-95%) of conventional furnaces or boilers commonly used at DoD facilities. Based on this field study, the overall average efficiency of the demonstration equipment was adversely impacted by part-load operation.

Manufacturer specifications for CCHP at rating conditions indicate high efficiency cooling (12.3 EER/ 24.1 IEER), almost twice the federal minimum standard. The specified heating efficiency ranges from 4.1 COP at moderate ambient temperatures (47°F) but decreases in both capacity and efficiency at lower ambient temperatures (2.3 COP @ 17°F). This demonstration confirmed reduced heating capacity and efficiency at colder temperatures. Measured heating and cooling efficiencies were slightly lower than ratings.

Low Temperature Performance: Air-source heat pumps extract heat from the ambient air during heating operation to achieve efficiency over 100% (COP>1). At cold ambient conditions, all air-source heat pumps (both gas-fired and electric) have reduced efficiency. Therefore, actual installed seasonal efficiency can vary greatly depending on the local climate. Figure 4 shows the relationship between the daily average heating efficiency and outside air temperatures from the demonstration.

GHP recovers heat from the engine to supplement heating output and maintain capacity and supply temperatures at colder ambient conditions than electric air source heat pumps. GHP heat recovery supplements the heat output and increases heating efficiency. GHP heat recovery also provides a higher supply temperature than an electric heat pump at the same outdoor conditions. During this demonstration, the GHP operated at its rated efficiency for only a limited range of operating conditions. Due to reduced part-load performance, this demonstration was not able to confirm the improved heating performance at low ambient temperatures. The GHP can be provided with an optional cold region kit to enable cold starts at low temperatures. Although the kit was not used in the demonstration, the GHP did not experience any start up failures due to cold temperatures.

In cold climates, electric heat pumps heating capacity and performance significantly decreases with lower ambient temperatures, often requiring supplemental heating. The demonstration electric CCHP was designed for cold climates without backup heating. The minimum operating temperature for the electric CCHP was -4°F. To compensate for its derated heating capacity at low ambient temperatures, the electric CCHP was oversized for the demonstration site; however, it still failed to meet the heating load for several days during the demonstration.

Part-load: Conventional equipment such as electric air conditioners or gas furnaces are typically oversized and often operate at part-load. For most HVAC equipment, part-load operation can have an adverse effect on efficiency. Likewise, heat pump efficiency is also be impacted by low part-load operation. Heat pump VRF systems must be sized appropriately so heat pumps can operate at higher loads to maintain performance.

For this analysis, part-load was defined as the measured heating and cooling output (using 5-minute intervals) relative to the total indoor capacity at specified rating conditions. For this field site, the GHP operated between 20% and 35% full capacity. The electric heat pump was oversized to meet the heating load and operated at part-loads between 5% and 30%. On the coldest days, heat pump runtime increased up to 24 hours with only slight increases in part-load due to reduced capacity at colder conditions.

In this demonstration, this GHP model had lower than expected performance due to lower part-load efficiency. Although GHPs use variable speed engines, field data shows lower efficiency at part-load operation below 60% rated output. This issue is likely due to product-specific controls and engine sizing. This extent of decreased performance at part-load is not seen in other engine-driven heat pump designs and is not inherent in this class of technology.

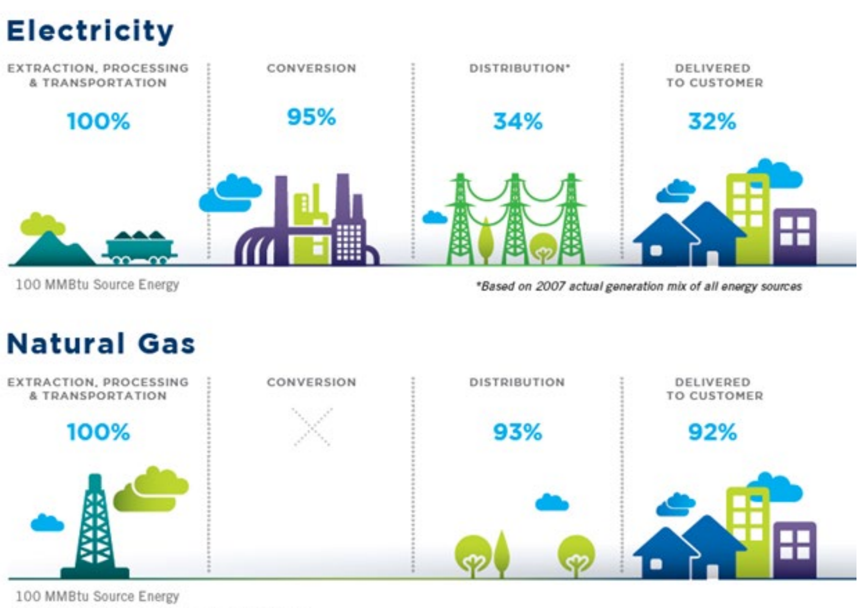
While CCHP manufacturer data report an increase in CCHP efficiency up to 50% part-load relative to full load ratings; however, measured field data showed a decrease in CCHP efficiency at part-load operation below 40%.

Peak Electric Demand: During cooling, GHPs consume natural gas in place of electricity, significantly reducing peak electric demand in comparison to electric chillers or electric heat pumps. Reduction in peak electric demand reduces the strain on local electric grids, decreases electric service requirements and reduces backup power needed for critical operations. This demonstration confirmed the GHP reduction in peak electric demand for both heating and cooling.

Electric heat pumps can potentially reduce summer peak electric demand if more cooling efficiency is higher than baseline electric air conditioners; however, during heating operation, electric heat pumps significantly increase peak electric demand if replacing gas-fired heating. For this demonstration, the heating peak was significantly higher than the cooling peak demand. Increases in electric heating, especially in cold climates, will result in a winter peak electric demand potentially larger than the summer peak.

Primary Energy and Full-Fuel-Cycle GHG Emissions: GHPs consume natural gas in place of electricity to provide space conditioning. Direct use of natural gas uses less primary energy (full fuel cycle) than comparable electric equipment.  Primary energy takes into account all the energy used to generate power or to supply fuel for heating or cooling. This includes the full fuel cycle, i.e. all energy required to extract fuel (natural gas, oil, coal, renewables), conversion (e.g. power generation), and distribution (pipelines, power transmission lines).

The full fuel cycle efficiency for natural gas is 91.74%, i.e. for each unit of gas, approximately 8.26% is lost due to processing or distribution. For electricity, the full fuel cycle efficiency is only 30.3% because most of the original energy extracted (coal/gas/oil) is lost due in the conversion to electricity at the power plants in addition to losses in the power transmission lines. As a result, even though site efficiency of electric heat pumps appears to be greater than GHPs, when considering the full fuel cycle, GHPs typically have a higher primary energy efficiency.



Source: American Gas Association

**Figure 12. Comparison of Primary Energy for Natural Gas and Electricity**

Economics: First costs are a key barrier for GHPs as compared to electric heat pumps and conventional HVAC equipment. Previous pilot studies have demonstrated reduced operating costs as compared to conventional HVAC equipment Although GHPs reduce energy costs due to energy savings as compared to conventional HVAC equipment. In addition, GHPs also significantly reduce peak electric demand along with the associated demand charges. Thirdly, since GHPs consume natural gas in place of electricity the difference in fuel prices can provide additional cost savings as natural gas prices are at historically low level and are expected to remain stable.

Based on this demonstration, both the GHP and electric heat pump reduced energy costs and lowered life-cycle costs compared to the baseline VAV system. In addition to energy savings, electricity pricing and demand charges will impact the heat pump economic benefits.

Maintenance: The GHP are based on natural gas reciprocating engines. Although the engine has a long maintenance interval (3 years or 10,000 hours), it requires additional periodic maintenance as compared to electric heat pumps and conventional HVAC. To address this barrier, ICE includes the cost of periodic engine maintenance in the purchase price of the system.

Altitude: NextAire™ GHP uses a lean burn engine and its efficiency can be impacted by higher altitudes. Per the manufacturer, the suggested elevation limit for the GHP is 1,000 m (about 3,300 ft); however, ETL/CSA certification testing was successfully conducted at 4,000 ft. In addition, several units are installed in Elko, Reno, and Carson City, NV, ranging from 4400 feet to 5066 feet elevations, and have operated without issues. It is unknown how higher altitudes would impact engine and heat pump performance

## 3.0 PERFORMANCE OBJECTIVES

The performance objectives for this demonstration focused on potential energy savings, cost savings, and full-fuel-cycle environmental benefits of heat pump VRF technology as compared to conventional HVAC. Both heat pump VRF systems were compared to each other and then to the baseline VAV system to determine which system best meets the performance objectives. Energy savings was based on measured energy use. Economic benefits were based on incremental installed costs and savings in operating and life-cycle costs. Environmental benefits were based on estimates of primary energy use and full-fuel-cycle GHG emissions (i.e. upstream from the meter). Qualitative benefits, such as reliability and comfort, were also evaluated. A summary of performance objectives for the gas engine-driven heat pump (GHP) and the electric cold climate heat pump (CCHP) field demonstration are listed in Table 1, including quantitative and qualitative performance objectives along with the corresponding success criteria.

**Table 3. Performance Objectives**

| Performance Objective | Metric | Data Requirements | Success Criteria | Results |
|---|---|---|---|---|
| Peak Electrical Load Reduction | Peak electric load (kW) | Interval/TOU Electric meter readings | 50% Reduction in peak electric load compared to CCHP | **Met Criteria**<br>82% (30 kW) reduced peak demand (heating); 36% (5kW) reduced peak demand (cooling) |
| Reduction in Annual Energy Costs | % Cost Savings | Energy use (electricity and natural gas), energy prices | 25% Reduction in annual energy costs compared to CCHP | **Met Criteria**<br>GHP had 41% lower energy costs than CCHP |
| System Economics | Life-cycle cost savings ($) | Energy use (electricity and natural gas), energy prices, equipment costs, equipment life | Lower life-cycle costs compared to CCHP | **Met Criteria**<br>GHP life-cycle costs were 4% lower than CCHP |
| Reduction in Primary Energy Use | Total primary energy use (MMBtu, kWh) | Estimated primary energy use based on measured energy use and regional primary energy factors | 10% Reduction in primary energy compared to CCHP | **Did Not Meet Criteria**<br>GHP primary energy use was similar to CCHP (+1%) but did not meet the criteria |
| Full Fuel Cycle GHG Emissions | Full fuel cycle GHG emissions (metric tons CO2e) | Estimated GHG emissions based on measured energy use and regional emission factors | 20% Reduction in GHG emissions compared to CCHP | **Did Not Meet Criteria**<br>GHP had 13% lower GHG emissions compared to the CCHP but did not meet the criteria |
| Comfort and Reliability | Meet specified heating and cooling capacity (0-100%)<br>No comfort or performance issues | Feedback from facility on reliability and comfort<br>Runtime for supplemental heating and cooling | Zero complaints regarding comfort or performance<br>0% supplemental heating or cooling required | **Did Not Meet Criteria**<br>due to GHP installation / component issues and reduced performance; CCHP also failed to meet heating loads at cold temperatures. |

*Performance Objective #1*

- Name and Definition: **Peak Electrical Load Reduction**

  The peak electric load is the highest hourly electricity consumption which typically occurs during high ambient temperatures due to peak electric air conditioning loads.

- Purpose: Peak electric load dictates the size of backup generators and electric service even when occurring for short periods. Electric utilities typically charge higher rates for electricity during peak usage for commercial customers through time-of-use rates or demand charges. Lower peak electric loads can generate significant savings in operating costs by reducing the associated electric demand charges. Lower peak electric loads also reduce the requirements for electric service for DoD facilities, which can be a key factor for building additions or locations with limited electric supply. Reductions in peak electric loads can improve energy resiliency by reducing the backup power needed for critical operations.
- Metric: Maximum hourly peak electric load (kW).
- Data: The electric load of the baseline and demonstration equipment (DOAS and both gas and electric heat pumps) was measured by electric watt meters for both heating and cooling operation.
- Analytical Methodology: The peak hourly electric load during heating and cooling operation was calculated for each heat pump system (including the DOAS) and compared to the baseline VAV system.
- Success Criteria: Achieve a 50% reduction in peak electric load for the GHP as compared to a CCHP at peak cooling loads.
- Results: GHP met this success criteria
  - GHP/DOAS reduced peak demand compared to the electric CCHP/DOAS by 30kW (82%) during heating.
  - Compared to baseline, GHP/DOAS reduced peak demand by 59kW (90%) during heating.
  - Compared to baseline, CCHP/DOAS reduced peak demand by 29kW (44%) during heating.

*Performance Objective #2*

- Name and Definition: **Reduction in Annual Energy Costs**

  For this demonstration, reduction in annual energy costs is based on annual energy costs including demand charges, natural gas and electricity use for heating and cooling.

- Purpose: GHPs provide space conditioning fueled primarily by natural gas with significantly lower electric use compared to conventional HVAC. With historically low gas prices, this can reduce annual energy costs for DoD bases. In addition, GHPs reduce peak electric loads which can generate significant savings in electric demand charges.
- Metric: Annual energy costs were based on measured peak electric use (kW), electricity consumption (kWh) and natural gas use (c.f.) for the demonstration and baseline equipment.

- Data: Gas meters and electric meters were installed on both heat pumps and baseline equipment to directly measure real-time gas and electric consumption. Regional commercial utility rates were used to determine annual energy savings.
- Analytical Methodology: A sensitivity analysis determined the potential annual energy savings for the range of published commercial utility rates.
- Success Criteria: Achieve a 25% reduction in GHP annual energy costs compared to CCHP.
- Results: GHP met this success criteria

GHP/DOAS reduced annual energy costs compared to the electric CCHP/DOAS by $3,551 (41%)

Compared to baseline, GHP/DOAS reduced annual energy costs by $12,671 (71%).

Compared to baseline, CCHP/DOAS reduced annual energy costs by $9,120 (51%).

*Performance Objective #3*

- Name and Definition: **System Economics**

  For this demonstration, system economics was determined by comparing current life-cycle costs for both heat pumps.
- Purpose: GHPs offer savings in energy costs but have higher first costs than CCHPs and conventional packaged heating and cooling systems. This analysis evaluated cost-effectiveness of GHPs for DoD facilities based on current costs and utility rates.
- Metric: Life-cycle costs savings indicate whether GHP energy savings offset its higher installed costs as compared to electric CCHPs.
- Data: For the life-cycle cost analysis, data on the incremental cost for equipment, installation, maintenance, and hardware lifetime is based on demonstration costs, published studies, and manufacturer information. Operational costs were based on measured energy use and commercial utility rates (Performance Objective #2) and estimated maintenance costs.
- Analytical Methodology: The Building Life-Cycle Cost Program model and the NIST *Life-Cycle Costing Manual for the Federal Energy Management Program* guideline was used to evaluate the system economics performance objectives.
- Success Criteria: Achieve lower life-cycle costs for the GHP as compared to a CCHP.
- Results: GHP met this success criteria

GHP/DOAS had 4% lower life-cycle costs than CCHP/DOAS

Compared to baseline, GHP/DOAS life-cycle costs were 37% lower

Compared to baseline, CCHP/DOAS life-cycle costs were 25% lower

*Performance Objective #4*

- Name and Definition: **Reduction in Primary Energy Use**

  Primary energy accounts for all upstream energy, i.e. energy used to generate power or to supply fuel for heating or cooling. This includes the full fuel cycle, i.e. all energy required to extract fuel (natural gas, oil, coal, renewables), conversion (e.g. power generation), and distribution (pipelines, power transmission lines).

- Purpose: Primary energy is a more comprehensive approach to evaluate energy use and is more relevant to energy security for DoD facilities than the energy metered at the site. DOE and a growing number of codes and standards are shifting to the use of full fuel cycle or primary energy use as the metric to quantify the environmental impact of appliances, rather than site energy.

- Metric: Primary energy use for the demonstration and baseline equipment is reported in terms of MMBtu and kWh.

- Data: Estimated primary energy use was calculated based on measured energy use, as described in Performance Objective #2, then multiplied by regional primary energy factors for natural gas and electricity. Regional factors were based on published values (e.g. EPA, DOE) for the full fuel cycle for natural gas and non-baseload electric generation for eGrid region RFCW. [SEEAT]

- Analytical Methodology: The difference in annual primary energy use for the GHP was compared to the CCHP and to baseline equipment to determine annual savings.

- Success Criteria: Achieve 10% reduction in primary energy for the GHP compared to CCHP.

- Results: Although the GHP did not met this success criteria due to lower than expected part-load performance, its primary energy use was similar to the CCHP based on measured data. Both VRF systems significantly reduced primary energy use compared to the baseline VAV system

GHP/DOAS primary energy use was only 1% higher than CCHP/DOAS

Compared to baseline, GHP/DOAS reduced primary energy use by 57%

Compared to baseline, CCHP/DOAS reduced primary energy use by 57%

*Performance Objective #5*

- Name and Definition**: Reduction in Direct Full Fuel Cycle GHG emissions**

  Full fuel cycle greenhouse gas (GHG) emissions include fossil fuel emissions produced by natural gas combustion at the site, in addition to upstream GHG emissions generated for each fuel. Full fuel cycle emissions include estimates of GHG emissions resulting from fuel extraction (natural gas, oil, coal, renewables), conversion (e.g. power generation), and distribution (pipelines, power transmission lines).

- Purpose: Full fuel cycle GHG missions represent a more global and comprehensive approach to evaluate the environmental impact of different technologies and fuel choices for DoD facilities.

- Metric: Full fuel cycle GHG emissions for the demonstration and baseline equipment is reported in terms of metric tons CO2e.
- Data: Full fuel cycle GHG emissions were estimated based on measured energy use, as described in Performance Objective #2, multiplied by regional GHG full fuel cycle emission factors for natural gas and electricity. Regional factors were based on published values (e.g. EPA, DOE) for the full fuel cycle for natural gas and non-baseload electric generation for eGrid region RFCW. [SEEAT] This total does not include GHG engine emissions due to lack of data.
- Analytical Methodology: Annual full fuel cycle GHG emissions for the GHP were compared to the CCHP and existing baseline equipment to determine savings.
- Success Criteria: Achieve 20% reduction in GHG emissions for the GHP compared to CCHP.
- Results: GHP did not met this success criteria but had similar full-fuel-cycle GHG emissions compared to the CCHP/DOAS even with lower than expected performance. In addition, GHP engine exhaust emissions were not included due to lack of available data which would slightly increase full-fuel-cycle GHG emissions.

GHP/DOAS had 13% savings GHG emission use than CCHP/DOAS

Compared to baseline, GHP/DOAS reduced GHG emissions use by 63%

Compared to baseline, CCHP/DOAS reduced GHG emissions use by 55%

*Performance Objective #6*

- Name and Definition: **Comfort and Reliability**

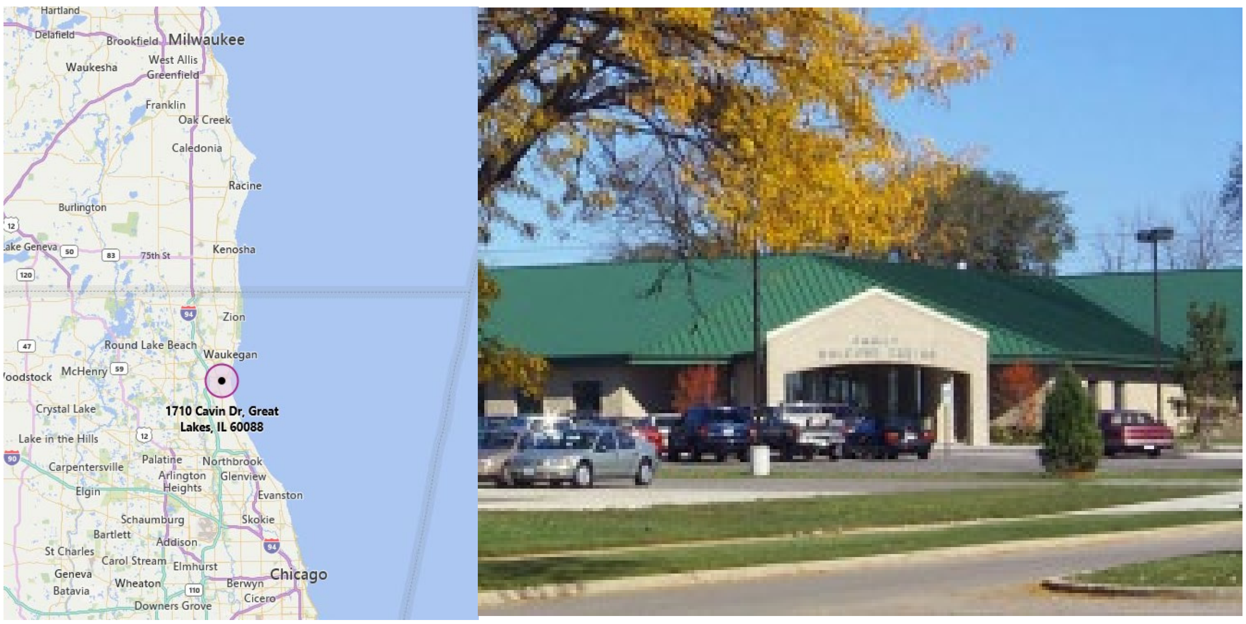
  For this demonstration, comfort and reliability is a qualitative performance objective to determine the impact of the heat pump zoned VRF configuration on occupants' comfort and the reliable performance of the technology under a range of real-life conditions.
- Purpose: Heat pump VRF configurations provide zoned space conditioning, which is expected to improve comfort as well as energy efficiency. Although sometimes subjective, comfort is a key objective of any space conditioning system and has been show to impact employee performance and productivity. As a new emerging technology, heat pumps are not familiar to many facility personnel or energy managers, and so must prove to be as reliable as conventional HVAC equipment to gain acceptance in the market.
- Metric: Success is measured by the number and content of comments from the building occupants the regarding comfort or reliability.
- Data: This objective was assessed by a summary of anecdotal complaints or comments in response to a survey of the facility manager and available occupants.
- Analytical Methodology: Anecdotal perspectives methodology was used by surveying facility personnel and occupants regarding noticeable changes in comfort or issues with reliability.

- Success Criteria: 0% supplemental heating or cooling required, and zero complaints regarding comfort or performance.
- Results: GHP did not met this success criteria due to installation/component issues and reduced performance which impacted the total delivered heating capacity. The CCHP also failed to meet the heating load at very cold ambient conditions.

# 4.0 FACILITY/SITE DESCRIPTION

## 4.1 FACILITY/SITE LOCATION AND OPERATIONS

The site selected for this demonstration was Family Housing Welcome Center (B8100) at NSGL, as shown in Figure 13. NSGL is located in northeast Illinois about 32 miles north of GTI headquarters. NSGL is the Navy's largest training facility and includes 1,153 buildings over 1,628 acres with over 25,000 military and civilian staff that work, train and live on the base.




Source: www.google.com

**Figure 13. NSGL B8100 Family Housing Welcome Center at Naval Station Great Lakes**

B8100 is a single-story multi-zone office building constructed in 2001. B8100 has two wings which extend North and East from the main entry shown in Figure 14. Each wing had an existing zoned VAV system. The North wing, circled in red, was selected for the demonstration due to its symmetric zones with higher heating loads than the East wing.

As shown in Figure 15, the existing HVAC system for the North wing was a ground-mounted conventional VAV system (AHU-1) with a total cooling capacity of 30-tons. Twenty-four months of gas and electric utility data for the total building indicated the VAV system had roughly twice the heating and cooling capacity needed for the space. When demonstration equipment was installed, the VAV system was decommissioned but remained in place so any modifications would be reversible.

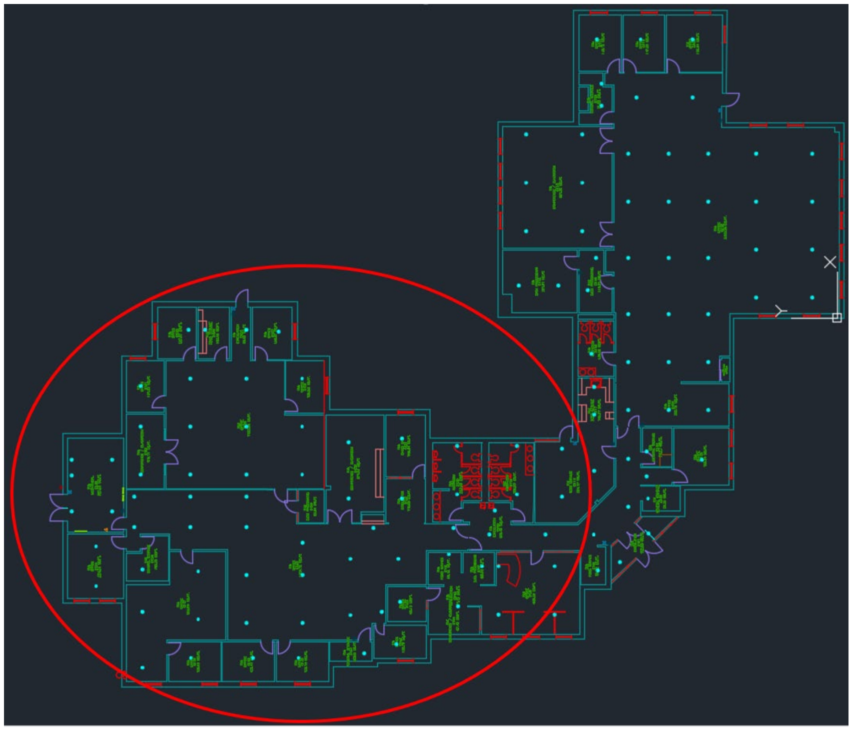


**Figure 14. Demonstration Units Provide Space Conditioning to the North Wing of the Family Housing Welcome Center, Circled in Red.**


**Figure 15. Demonstration Equipment Replaced the Existing Ground-mounted VAV System**

## 4.2 FACILITY/SITE CONDITIONS

As one of the first cold climate installations of the Model E GHP, this demonstration provided a unique opportunity to directly compare measured performance for the electric CCHP and the GHP in a cold climate. Although both heat pumps are commercially available, there are a limited number of cold climate installations in the U.S. NSGL is located in a cold climate (ASHRAE Climate Zone 5) and experiences high heating loads. Northern Illinois has cold and moderately long winters with 6,536 annual heating degree days and 752 cooling degree days, and an average 131 days per year below freezing, making it an ideal climate to demonstrate the heating efficiency and capacity of heat pump technology. The Daikin CCHP is designed for cold climates without supplemental heating and has a minimum rated temperature of -4°F. The ASHRAE 99% design temperature for this area is -5°F; however, during the demonstration period, minimum temperatures exceeded -5°F on several occasions including historic cold temperatures of -23°F during the January 2019 Polar Vortex.

The selected demonstrated site is a multi-zone office building, a common application for VRF technologies. In addition to office buildings, heat pump VRF technologies are appropriate for other multi-zone building types in military installations including schools, retail, hospitals, and hotels. For an accurate comparison of the VRF heat pump systems, the North wing of the building was divided into two equivalent sections with a similar number of zones and thermal loads. One section was served by the electric CCHP with ten indoor VRF fan coils and the remaining section was served by the GHP with ten indoor VRF fan coils. Heat pump equipment was sized based on building design loads. To estimate annual energy use for comparison to the baseline system, measured energy use was normalized with respect to the total heating and cooling delivered to correct for any differences in building conditions or operations.

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# 5.0 TEST DESIGN

This demonstration provides comparable performance metrics for the GHP and electric CCHP to address the performance objectives described in Section 4. The field study included a full year baseline monitoring the existing VAV system, followed by a full year demonstration monitoring to quantify heating and cooling performance across a full range of operating conditions. Measured data was used to determine potential energy savings, cost savings, and environmental benefits of both heat pump technologies compared to conventional systems for cold climate DoD facilities.

## 5.1 CONCEPTUAL TEST DESIGN

This demonstration provided a side-by-side comparison of two VRF systems, a GHP and an electric CCHP, in a cold climate application. The field site was a small multi-zone office building divided into two thermal zones, one served by the GHP and the other served by the electric CCHP. Performance data was collected over a full calendar year and used to determine annual energy performance compared to the baseline HVAC equipment. Prior to the heat pump installation, GTI conducted a full year of baseline monitoring of the existing variable air volume unit (VAV) across the full range of heating and cooling loads. An economic analysis estimated the annual energy and life-cycle costs based on measured energy use data, installed costs, and local utility rates. Qualitative benefits, such as reliability and comfort were also addressed.

- Hypothesis:
  - GHP will achieve a 50% reduction in peak electric load as compared to CCHP
  - GHP will achieve a 25% reduction in annual energy costs compared to CCHP
  - GHP will achieve lower life-cycle costs compared to CCHP
  - GHP will achieve a 10% reduction in primary energy compared to CCHP
  - GHP and CCHP will provide specified heating and cooling over a range of ambient conditions
  - GHPs and CCHP will improve or maintain reliability and comfort levels

The independent variable is the space conditioning technology used to provide heating or cooling to DoD facilities. Three technologies were monitored: 1) conventional VAV with gas-fired heating and electric air conditioning; 2) natural gas engine-driven heat pump (GHP) VRF system; and 3) electric cold climate heat pump (CCHP) VRF system.

- <u>Dependent variables</u> included natural gas and electricity consumption, heating and cooling delivered, equipment runtime, temperature and humidity in the conditioned zones.
- <u>Controlled variables</u> included the building heating and cooling loads and the ambient temperature profiles. This side-by-side demonstration ensures both heat pumps operate under similar ambient conditions for a direct comparison. The VRF configuration was designed so both heat pumps have equivalent space conditioning loads. To correct for any differences, the energy use for each heat pump was normalized with respect to the total heating and cooling delivered.

- Test Design: Existing HVAC equipment was monitored for a full year to quantify the baseline for comparison. After installation of the heat pump VRF systems, gas and electric consumption was monitored over full year to determine the relative energy savings for each technology. Energy use and manufacturer data were used to estimate annual energy, operating, and life-cycle costs for both heat pumps. Comfort and reliability were determined by the ability of each system to meet the heating and cooling loads and feedback from the building occupants.
- Test Phases:
  - Verify test site
  - Baseline sensor installation and commissioning
  - Baseline data collection and analysis
  - GHP and CCHP VRF configuration design for demonstration
  - GHP and CCHP installation and commissioning
  - Sensor installation and commissioning
  - Data collection for a full calendar year
  - Data analysis to determine performance and economic metrics
  - Technology transfer of findings

## 5.2 BASELINE CHARACTERIZATION

Energy use of the existing HVAC equipment was monitored during the 2016/2017 heating and cooling season to determine the baseline for energy savings.

Baseline monitoring included direct measurement of gas and electric consumption, air supply and return temperature and humidity levels of the existing VAV system. Gas meters equipped with pulse counters were installed in the gas supply to accurately monitor baseline gas consumption. A compact watt-hour meter monitored electricity consumption of the supply fan and a second watt-hour meter monitored electricity consumption of the total system, including the compressor. A current switch recorded the runtime of the supply fan and economizer operation. Room temperatures and relative humidity were monitored in each conditioned zone. Outdoor air, return air, and supply air temperatures were also measured recorded. Monitored data points are shown in the diagram in Figure 16.

Baseline data was collected with a data acquisition system (DAS) with remote communications capability providing real time access as needed for diagnostics or troubleshooting. Data was recorded at 5-minute intervals.

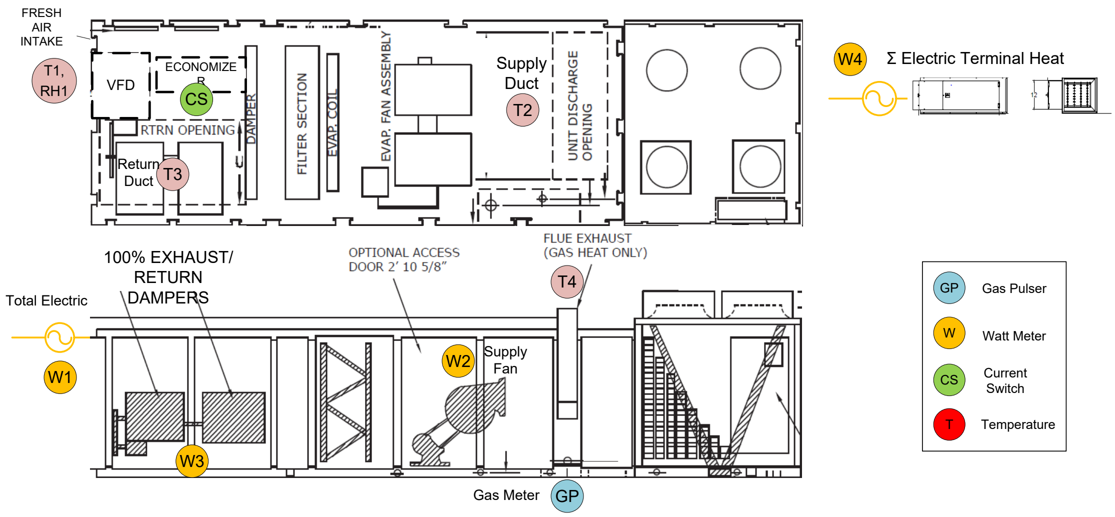



**Figure 16. Baseline VAV Monitored Data Points**

For the baseline characterization, daily energy use was normalized with respect to published TMY3 30-year Normals cooling and heating degree days [National Climactic Data Center] to predict baseline annual energy use. The base temperature of 55F was used for heating degree days and 60F was used for cooling degree days as the best correlation to the measured data. Measured energy use for the baseline VAV system was highly correlated to heating and cooling degree days. As shown in Figure 17, cooling energy use was linear with respect to CDD60 for the VAV outdoor unit. There was no correlation for electric resistance heat provided by the VAV boxes (gray). For heating operation, shown in Figure 18, both natural gas use and total electric consumption (including VAV-box electric reheat) correlated with HDD55.

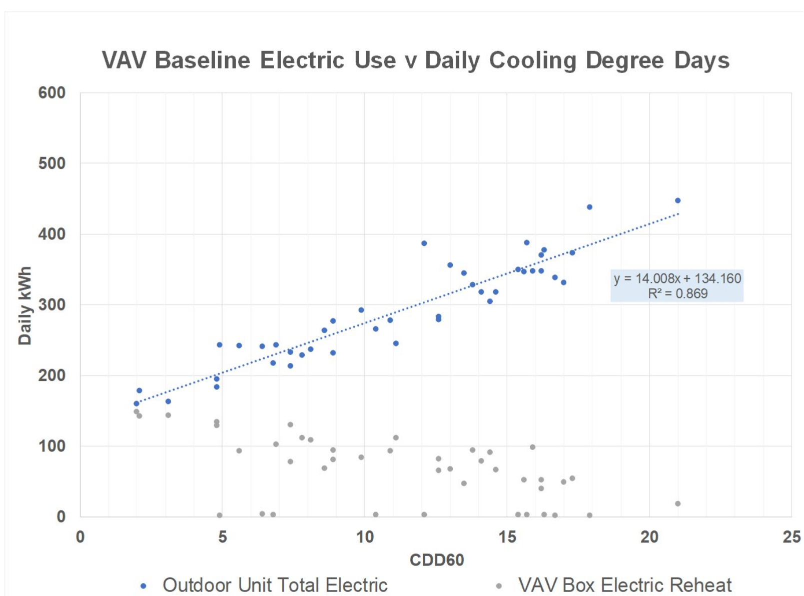



**Figure 17. Baseline Measured Energy Use for VAV Cooling Operation Correlated with CDD60**

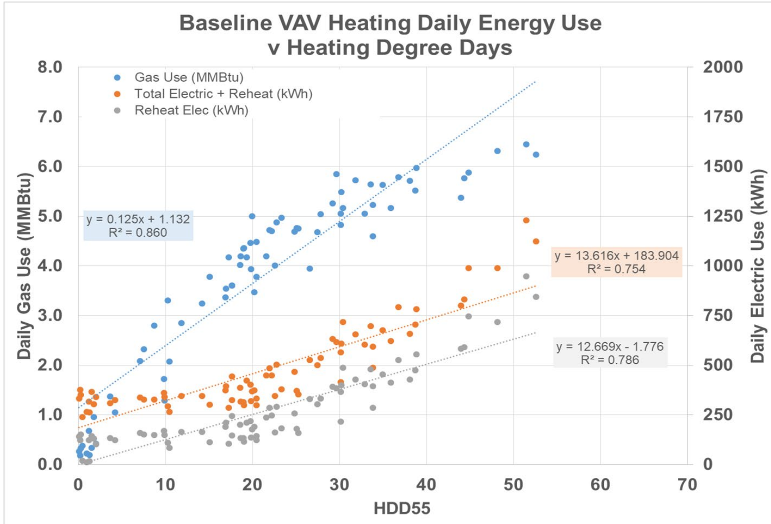



**Figure 18. Baseline Measured Energy Use for VAV Heating Operation Correlated with HDD55.**

*Reheat energy use during heating (shown in gray) was higher than expected*

During heating operation, energy use for the VAV-boxes was higher than expected. VAV-boxes are designed to provide electric resistance heat for trim heating and to adjust temperatures between zones. Baseline data in Figure 19 shows VAV-boxes operating at peak heating capacity (yellow) to achieve setpoints while the outdoor unit's modulating gas burner (orange) operated at low fire. As confirmed by the manufacturer, a building automation system (BAS) is required to integrate the controls for the gas burner and VAV-boxes. Without a BAS, the outdoor unit and VAV-boxes operated independently resulting in excess electric resistance heating with a higher peak electric demand.

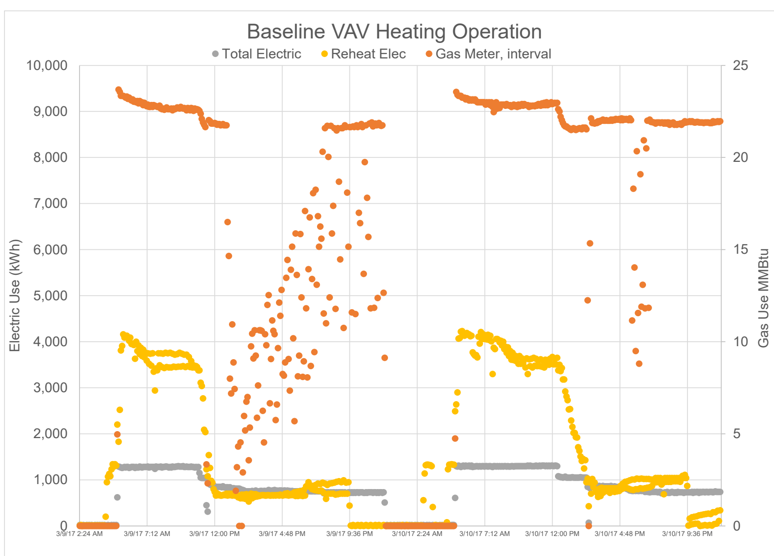



**Figure 19. Lack of integrated Controls for the VAV Natural Gas Burner and the Electric Resistance Heaters in the VAV-boxes Resulted in Higher than Expected Energy Use.**

Although the baseline system did not operate as designed, this may be typical VAV operation for smaller buildings or sites without a central BAS. Due to the lack of integrated controls, baseline electric consumption and peak electric demand may be higher than typical installations. Table 4 presents the baseline annual energy use normalized to TMY3. A large portion of the total electricity consumptions is due to VAV-box resistance heating.

**Table 4 Baseline Estimated Annual Energy Use for Existing VAV System**

| Baseline | Gas Use (Therm) | Electric Use (kWh) | Heating Peak Electric Demand (kW) | Cooling Peak Electric Demand (kW) | Annual Primary Energy (MMBtu) | GHG Emissions (MTCO2e) |
|---|---|---|---|---|---|---|
| Total VAV System w Reheat (9.5 EER, 80%Te) | 5,474 | 125,713 | 65.4 | 43.2 | 1,995 | 158.2 |
| VAV-box Heating | | 55,259 | | | | |
| VAV-box Cooling | | 4,969 | | | | |

## 5.3 DESIGN AND LAYOUT OF TECHNOLOGY COMPONENTS

### 5.3.1 System Design

The demonstration HVAC system replaced the baseline VAV heating and cooling system (AHU-1) serving the North wing of the building (Figure 14). The demonstration equipment consisted of two commercially available heat pumps VRF systems paired with a DOAS. Table 5 lists the key HVAC system components. Both heat pumps utilize Daikin VRF air handlers for zoned applications. For a direct comparison of performance, the demonstration site was divided into two separate, comparable HVAC zones. Both systems have similar loads, number of fan coils, and identical ventilation. To calculate energy savings, measured energy use was normalized with respect to heating and cooling delivered to correct for any differences in building conditions or operations.

The VRF systems were paired with a conventional DOAS, with gas-fired heating and electric DX cooling, to provide ventilation. For cold climates, gas heating is needed to meet the needed to achieve the large temperature rise for conditioning 100%OA. The DOAS delivered conditioned air at 64°F directly to each zone via ceiling diffusers. The DOAS was able to use the existing ductwork which significantly reduced installation costs for this site. The DOAS air flow was balanced using the dampers in the existing VAV-boxes and the electric reheat coils were powered off.

Specifications for the VRF fan coil units are listed in Table 6 and design drawings are shown in Figure 20 and Figure 21. Each heat pump VRF systems were matched with ten cassette-style VRF fan coil units that were installed above the ceiling panels throughout the building. To accurately size the heat pumps to the building load, a design engineering firm developed load calculations based on code-required minimums taking into account a range of DOAS setpoints, e.g., 60°F supply to address outdoor latent loads during the cooling season and a neutral 70°F during the heating season. A 10% safety factor was added to both heating and cooling capacities.

An 8-ton GHP was specified to meet the cooling load for offices on the west side of the building. The GHP outdoor unit was installed with ten indoor VRF fan coil units with a total 7.2 tons cooling capacity. The electric CCHP was specified for offices on the east side of the building. A 12-ton electric CCHP outdoor unit was paired with ten indoor VRF fan coil units with a total of 6.8 tons cooling capacity. Because of the reduced heating capacity of the CCHP at lower ambient temperatures, the CCHP outdoor unit was oversized to meet the heating load at the coldest design conditions. The specified minimum temperature rating of the CCHP was -4°F.

**Table 5 Demonstration Equipment Specifications**

| Component | Cooling | Heating |
|---|---|---|
| NextAire Multi-Zone Gas Heat Pump<br>Model: AXGP096D1NHS, R410A<br>208/230V 1Ph, 6 MCA, 20 MOCP<br>Dimensions: 52"L x38"W x 82"H<br>1365 lbs | 96 MBH / 8RT (28 kW)<br>Efficiency: 1.3 COPg<br>Gas use: 73 MBH<br>Power consumption: 1.12kW | 103 MBH (30 kW)<br>Efficiency: 1.3 COPg<br>Gas use: 79.5 MBH<br>Power consumption: 0.9kW |
| Daikin Electric Heat Pump<br>Model: RXYQ144TYDN, R410A<br>460V-3-60, 25.9 MCA, 35 MOCP<br>Dimensions: 49"L x 30"W x 67"H, 12" stand added for snow clearance<br>710 lbs | 138 MBH / 12RT (40 kW)<br>Efficiency: 12.3 EER;<br>24.1 IEER<br>*2018 Federal Min: 12.2 IEER* | 154 MBH (45 kW)<br>Rated at -4°F<br>Efficiency:<br>4.1 COP @ 47°F<br>2.3 COP @ 17°F |
| Ventilation: Daikin DOAS<br>Model: DPS006A, 800 cfm<br>460V-3-60, 10.8 MCA, 15A MOCP<br>Modulating natural gas heat; electric R410A air conditioner | 73.5 MBH / 6RT<br>Efficiency: 11.3 EER | 96 MBH<br>Gas use: 120 MBH<br>Efficiency: 80%Et |

**Table 6. Cassette VRF Fan Coil Units**

| HEAT PUMP RUNS WITH ELECTRIC CONDENSING UNIT | | | |
|---|---|---|---|
| TAG | SERVICE | NOMINAL CAPACITY (TONS) | AIR FLOW (CFM) |
| EHP 101 | 0132 CONFERENCE / CLASSROOM | 0.75 | 244 |
| EHP 102 | 0131 RM OFFICE | 0.50 | 212 |
| EHP 103 | 0130 RM OFFICE | 0.50 | 212 |
| EHP 104 | 0129 RM FOOD SERVICE | 0.50 | 212 |
| EHP 105 | 0125 RM OFFICE | 1.50 | 488 |
| EHP 106 | 0127 RM OFFICE | 0.50 | 212 |
| EHP 107 | 0126 RM OFFICE | 0.50 | 212 |
| EHP 108 | 0123 CONFERENCE / CLASSROOM | 1.00 | 283 |
| EHP 109 | 0121 RM CHILD CARE | 1.00 | 424 |
| EHP 110 | 0102 RM CIRCULATION | 1.00 | 424 |

**Table 7. Cassette VRF Fan Coil Units**

| HEAT PUMP RUNS WITH GAS CONDENSING UNIT | | | |
|---|---|---|---|
| **TAG** | **SERVICE** | **NOMINAL CAPACITY (TONS)** | **AIR FLOW (CFM)** |
| GHP 101 | 0134 RM OFFICE | 0.50 | 212 |
| GHP 102 | 0136 RM GENERAL STORAGE | 0.50 | 212 |
| GHP 103 | 0137 RM OFFICE | 1.00 | 283 |
| GHP 104 | 0138 RM OFFICE | 0.50 | 212 |
| GHP 105 | 0139 RM OFFICE | 0.50 | 212 |
| GHP 106 | 0140 RM OFFICE | 0.50 | 212 |
| GHP 107 | 0135 RM OFFICE | 2.00 | 566 |
| GHP 108 | 0142 RM OFFICE | 0.50 | 212 |
| GHP 109 | 0143 RM OFFICE | 0.50 | 212 |
| GHP 110 | 0148 RM OFFICE | 1.50 | 488 |

### 5.3.2 System Controls

A central touch screen controller (iTouch) was installed in the building mechanical room to operate both heat pumps and all fan coils and provide high level monitoring of the DOAS. The iTouch is a standalone system and not connected to any building automation system. The HVAC operation is as follows:

- DOAS operated continuously during occupied hours delivering 64°F supply air directly to each zone.

- Each VRF indoor fan coil was controlled by a thermostat/controller installed in that zone to monitor room temperatures. The design setpoints for each zone were originally: summer occupied/unoccupied: 70°F/80°F and winter occupied/unoccupied 68°F/65°F. During the demonstration, the project team discovered several building tenants used electric space heaters to address comfort issues due to the baseline VAV system. To eliminate any space heater operation during the demonstration, zone temperature setpoints were increased to 72°F during occupied hours for heating or cooling. Switchover from heating to cooling was set at 75°F and switchover from cooling to heating was set at 69°F. Measured data showed both VRF heat pumps successfully maintained the interior zone temperatures within ±2°F of the setpoint during typical operation. In the future, additional energy savings might be achieved with lower zone setpoints (e.g. 70°F heating).

- Heat pumps operate fans continuously during occupied hours to monitor space temperature. Fan speed is set automatically by the heat pump internal controller.

- The DOAS was installed with an Emergency Air Distribution Shutoff per DoD regulations to shut down the air intake if necessary.

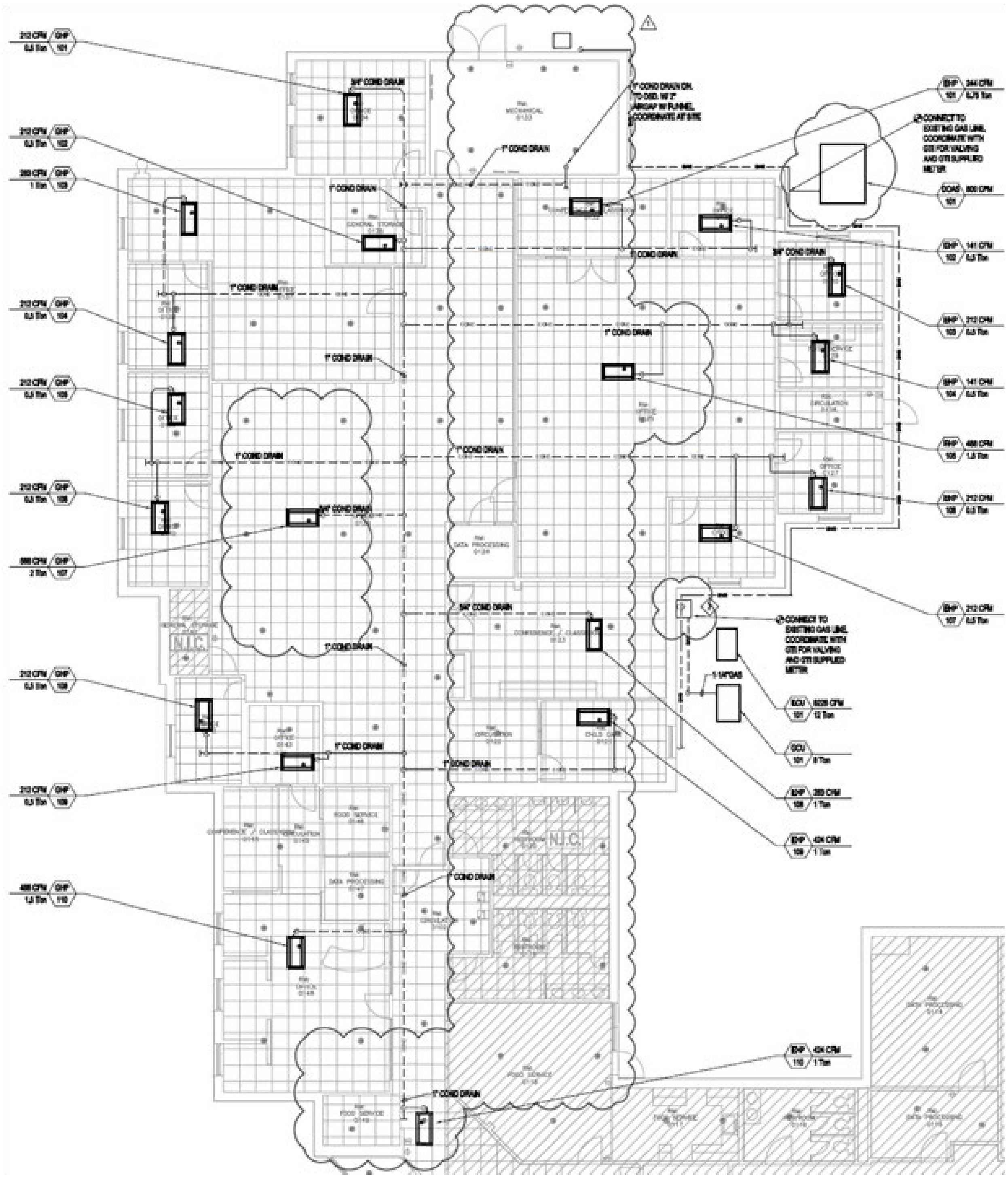


**Figure 20. Layout of Heat Pump VRF Piping and Fan Coils**

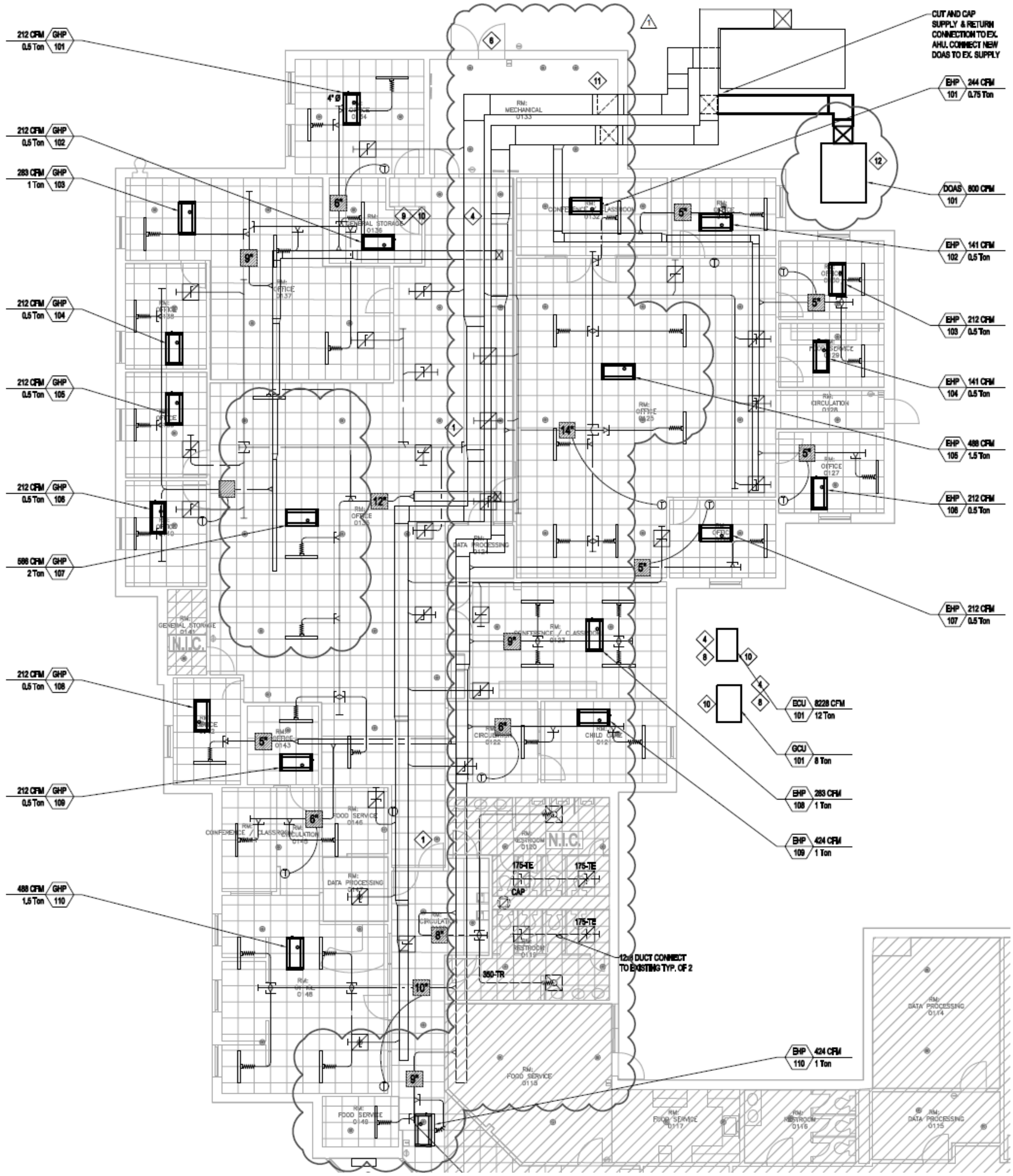


**Figure 21. Design for DOAS to Supply Conditioned OA Using Existing Ducting**

## 5.4 OPERATIONAL TESTING

### 5.4.1 Start-up and Commissioning

The project team and representatives from both manufacturers commissioned the installation and start-up of both heat pump VRF systems. The installation contractor followed manufacturer instructions for installing VRF system tubing, vacuum and pressure testing tubing, then added refrigerant. During commissioning, the installation contractor noted that temperature sensors in each zone controllers were reading higher than actual temperatures (i.e. displaying 72°F when actual room temperatures were 67°F). All zone controllers were adjusted to correct for this issue.

### 5.4.2 Monitoring

The field study included a full year baseline monitoring followed by a full year demonstration monitoring to quantify heating and cooling performance across a full range of operating conditions. The equipment was fully instrumented to determine system performance based on measured heating and cooling output relative to the energy use. Data was recorded at 5-minute intervals. The data logger and cellular modem allowed data to be accessed remotely providing real time access to data as needed.

### 5.4.3 Data Analysis

Energy use for each heat pump VRF system was adjusted with respect to the total heating or cooling delivered to account for any differences in loads on each section of the building. Higher heating value of natural gas was as 1050 Btu/cubic feet used to convert volumetric meter measurements to natural gas energy consumption. The same conversion factor was used for baseline and technology demonstrations. Measured energy use was correlated with daily heating and cooling degree days.

To predict annual energy and cost savings, energy use data was then normalized to published TMY3 cooling and heating degree days (30-year normal; National Climactic Data Center) Energy savings for both heat pumps systems (e.g. heat pump VRF system and DOAS) was determined with respect to the baseline VAV measured data. GTI also compared reductions in peak electric demand for the north wing of the building with facility energy bills to validate electric demand reductions.

An economic assessment was conducted to determine relative energy cost savings, primary energy, full fuel cycle GHG emissions, life-cycle costs and simple payback. Energy prices were based on incremental composite rates provided by the field site ($0.0559/kWh; $10.3037/kW; $0.49/therm). Utility demand charges and rate structures can vary widely from state to state and will impact energy cost savings. Demand charges for the field site were calculated from the highest hourly peak kW during the past 12 months, whether summer or winter.

### 5.4.4 Modeling and Simulation

To apply the demonstration results to other DoD facilities in multiple climates, existing EnergyPlus™ VRF models were adapted for the GHP and CCHP based on manufacturer data and measured field data. These models were used to estimate potential energy savings and economic benefits for five other climates. Model simulations were run for a small office building, based on the field site, for five different climates to predict the energy use of the demonstration equipment with respect to conventional HVAC.

### 5.4.5 Decommissioning

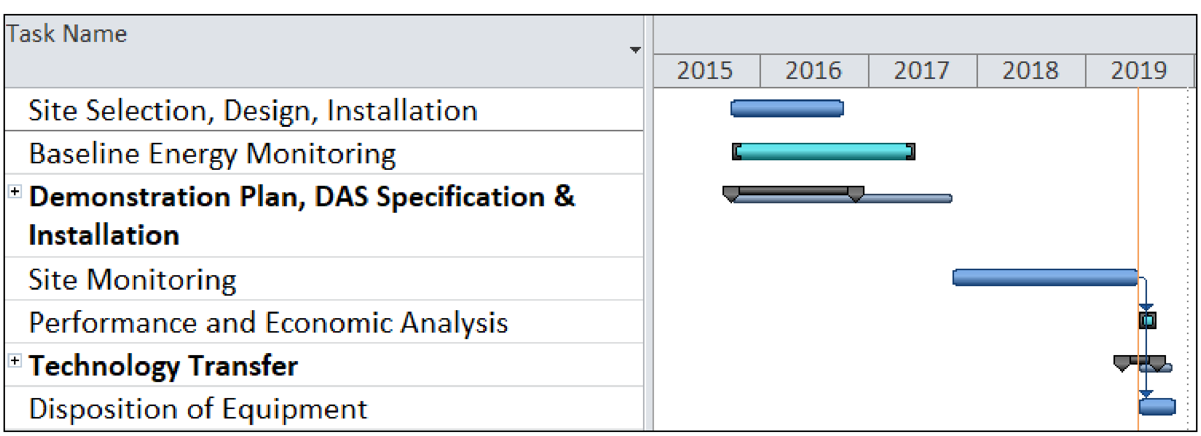
Following performance monitoring completed in May 2019, ownership of the demonstration equipment was transferred to NSGL. Despite some component issues during the demonstration, the site chose to keep the VRF systems and was pleased with the improved comfort. All meters were removed from the refrigerant lines and replaced with continuous tubing to eliminate leaks in the future.

The GHP manufacturer provided onsite service training for NAVFAC facility staff so they could conduct routine maintenance and support for the VRF systems. Moving forward, NAVFAC can do engine maintenance in house or use a qualified contractor. Training included hands on basic engine maintenance and service for the GHP; the second day covered the VRF system and controls. NAVFAC facility staff included both engine and HVAC experts. Following training, GTI surveyed facility staff regarding the level of difficulty in servicing the VRF systems, based on their existing experience with air conditioning refrigerants and engines. All staff felt confident they could transfer their expertise to providing technical support for the VRF systems.

### 5.4.6 Timeline

Baseline monitoring was conducted from October 2016 to October 2017. Installation of the demonstration equipment was completed in October 2017 and demonstration monitoring was conducted until May 2019. The demonstration monitoring period was extended for an additional heating season due to installation and component issues occurred during the first heating season.

**Table 8. Timeline of Operational Testing**



## 5.5 SAMPLING PROTOCOL

### 5.5.1 Demonstration M&V Approach

The demonstration equipment (DOAS, GHP and CCHP) was instrumented with gas and electricity meters to measure actual energy use. Compact watt meters monitored total electric consumption at the outdoor condensing units; additional watt meters measured the total energy use for each set of indoor fan coils. Outdoor conditions were monitored using temperature and humidity sensors installed near the outdoor units. DOAS supply air temperatures were also monitored. Hobo loggers measuring temperature and humidity were placed in several zones (offices) to evaluate changes in comfort (Figure 22). These loggers record data every 30 minutes and collected data continuously for over one year.

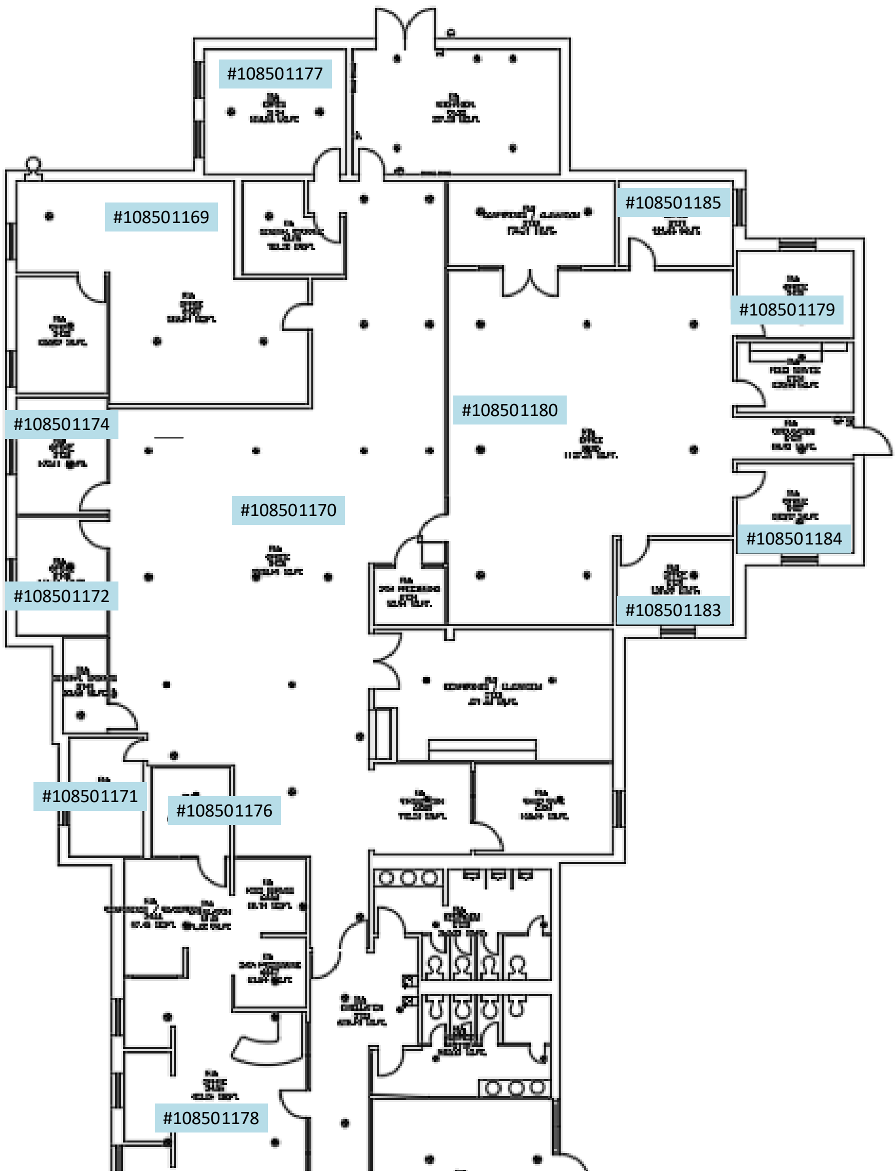



**Figure 22. Placement of Hobo Temperature and RH Loggers**

The average efficiency of both VRF systems was calculated based on total energy (heating and cooling) delivered to total energy consumed for a given period. The design of the cassette-style VRF fan coil units make it challenging to accurately measure the supply and return air temperatures in order to calculate the total energy delivered. For this demonstration, GTI developed an alternative approach to determine the total heating and cooling delivered by the VRF system by measuring changes in enthalpy at the refrigerant lines, shown by the sketch in Figure 23.

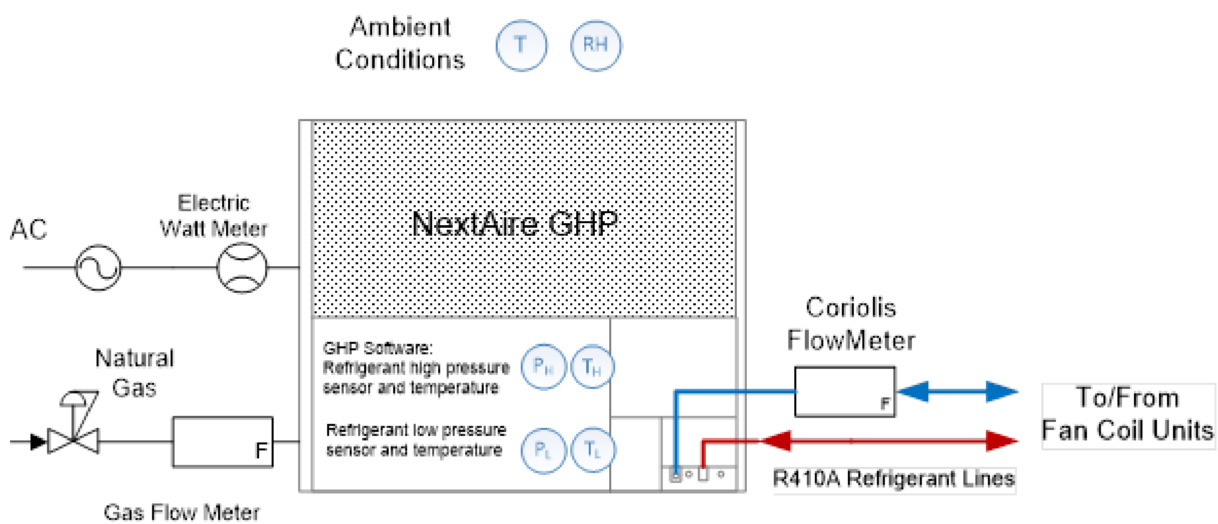



**Figure 23. Heat Pump VRF Systems Were Fully Instrumented to Measure Energy Consumption and Heating/Cooling Delivered**

A Coriolis flowmeter was installed in the liquid refrigerant line near the outdoor heat pump unit to measure the mass flow rate of the refrigerant delivered to the fan coils inside the building (Figure 24). Thermocouples and pressure sensors were installed in the liquid and vapor lines adjacent to the outdoor unit. Enthalpy of the liquid and vapor refrigerant was calculated based on R410A properties using the NIST Reference Fluid Thermodynamic and Transport Properties Database (REFPROP). Heating or cooling delivered was determined by the enthalpy change between the vapor and liquid refrigerant lines. In previous studies, this method of measurement was successfully validated through comparison of measured field data to laboratory data at similar conditions [confidential study]. This approach improved accuracy and reduced M&V costs compared to conventional methods based on measurements of air flow and supply/return temperatures at the fan coil units.

All instrumentation purchased for this demonstration was calibrated by the manufacturer. The only equipment commissioned onsite was the Micro Motion Coriolis meter. Although the meter was factory calibrated, the meter settings were adjusted to filter out noise due to low flow rates or bi-directional flow during stopping and starting to accurately measure the flow in the liquid refrigerant line. Flow startup services were provided by the manufacturer and all adjustments were made by a trained technician.

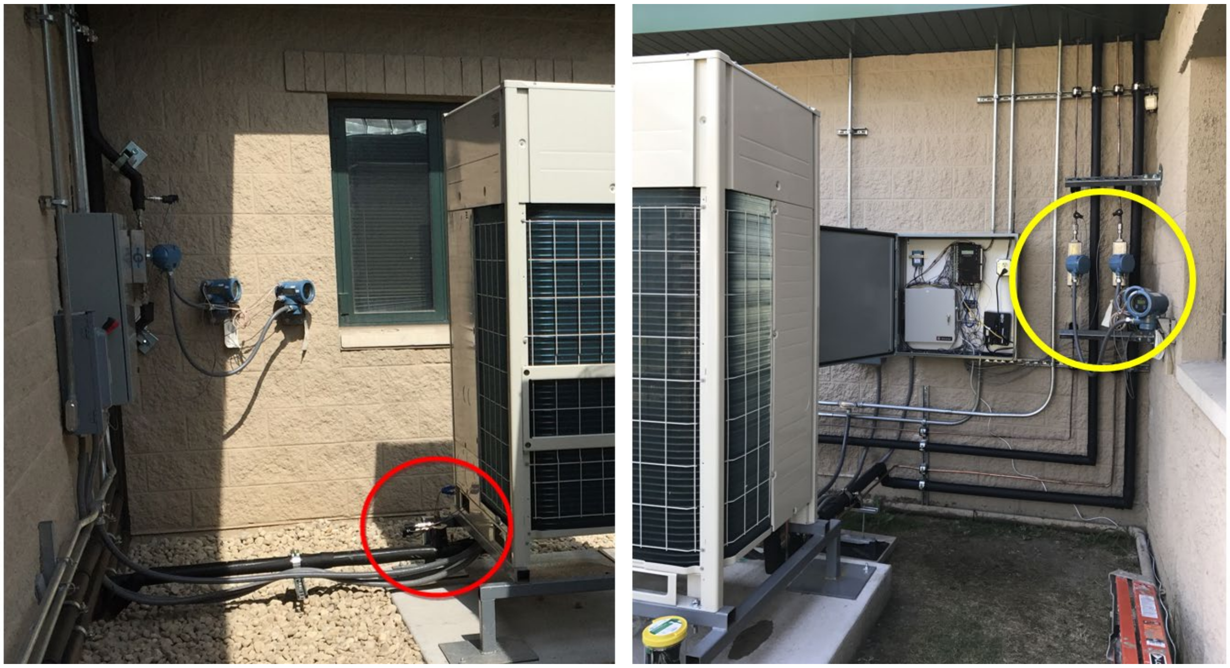


**Figure 24. (a) Pressure and Temperature Sensors Inserted in Liquid and Vapor Refrigerant Lines (circled in red) Were Used to Calculate Heating/Cooling Delivered. (b) Coriolis Flow Meters (circled in yellow) Measured the Liquid Refrigerant Mass Flow.**

## 5.6 SAMPLING RESULTS

The following measured data for the demonstration equipment (DOAS, GHP, and CCHP) cannot be compared directly but illustrates performance trends. For the side-by-side heat pump demonstration, the DOAS in this configuration meets the ventilation load for the total building section, while each VRF system serves about half the load. For the energy savings analysis, as described in Section 6.1, energy use for each heat pump VRF systems was normalized with respect to the total load of the building section to allow for direct comparisons between different equipment.

### 5.6.1 Demonstration Cooling Performance

Monitored cooling performance was based on a limited dataset due to unrelated component issues and operational outages. Figure 25 shows the measured daily energy use during cooling operation for both heat pump VRF systems. The left graph shows daily gas use of the GHP with respect to cooling degree days (base 60°F), and right graph shows daily electric consumption of the GHP (blue) and the electric CCHP (orange). The GHP uses a natural gas engine instead of an electric motor to drive the compressors, resulting in gas use during cooling with significantly reduction in electricity use and peak electric demand.

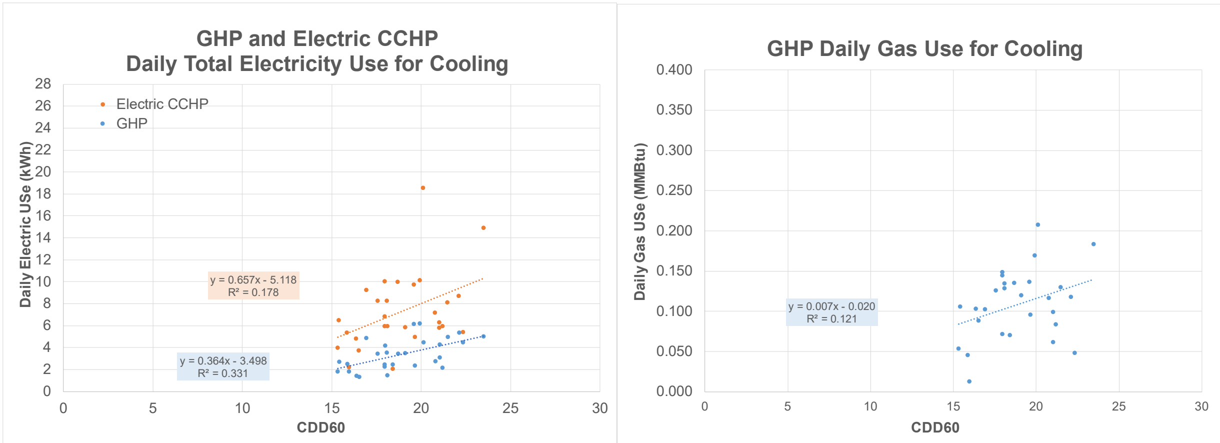



**Figure 25. Measured Energy Use for Heat Pump VRF Cooling Operation with Respect to Cooling Degree Days.**

*(a) Left: GHP and CCHP electricity use (b) Right: GHP gas use*

Figure 26 shows the DOAS measured cooling energy use correlated with CDD60. For the demonstration site, DOAS electric consumption is orders of magnitude higher than the VRF systems highlighting the larger ventilation cooling load for this climate. DOAS are typically designed to deliver space neutral supply air; however, for this site the DOAS setpoint was 64°F year-round which also reduces the building cooling load. As a result, the VRF systems have shorter runtimes and operate a lower part-loads during the cooling season.

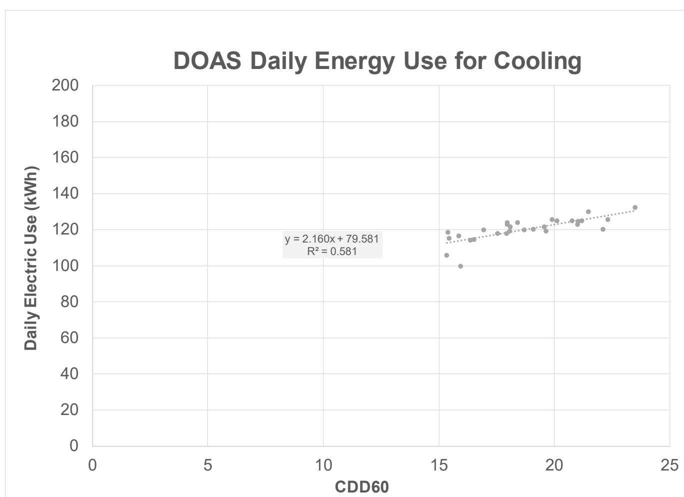



**Figure 26. DOAS Measured Electricity Consumption with Respect to CDD60**

### 5.6.2 Demonstration Heating Performance

Figure 28 shows the measured daily energy use of the CCHP correlated to HDD55. CCHP heating operation generated a high peak electric demand for this site, much higher than the summer peak demand. For this demonstration, the CCHP was unable to meet the heating load for seven days at daily average temperatures below 16°F, indicating the need for supplemental heating.

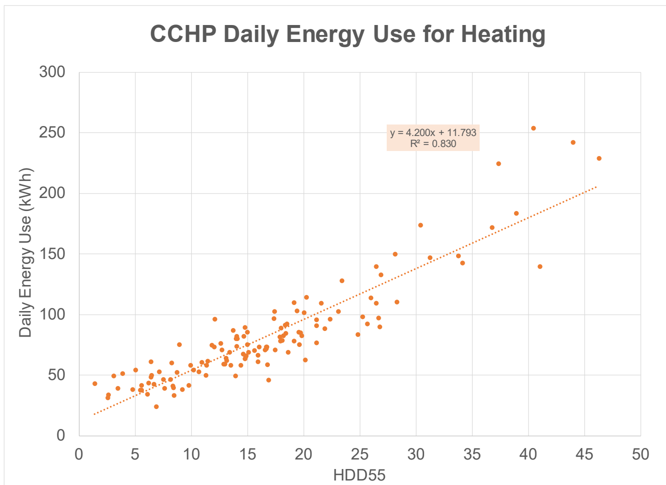



**Figure 27. CCHP VRF Measured Electricity Use for Heating Correlated to HDD55**

Figure 28 shows the measured daily energy use of the GHP and DOAS. As shown in the left graph, GHP natural gas consumption correlated to HDD55, while the electricity consumption for the outdoor unit and indoor fan coils remained relatively constant. The right graph shows the natural gas use of the DOAS meeting the OA ventilation load was highly correlated with HDD55. The DOAS electric use for fans and controls was relatively constant.

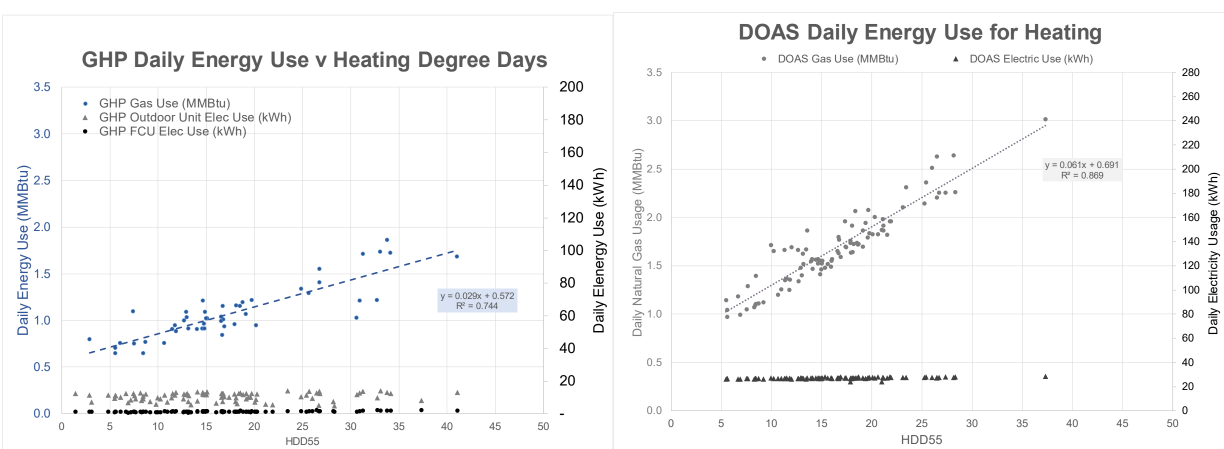



**Figure 28. (a) Left Graph Shows GHP Measured Gas Use for Heating Correlated to HDD55. (b) Right Graph Shows DOAS Measured Gas Use Correlated to HDD55.**

### 5.6.3 Part-load Performance

Most conventional HVAC equipment are over-sized to ensure adequate capacity and part-load operation can have an adverse effect on efficiency. Likewise, heat pump efficiency is also impacted by part-load operation. VRF heat pumps by design modulate to meet the heating and cooling loads of each zone. VRF systems often operate at lower part-loads depending on the sizing of the indoor VRF fan coils units and the outdoor heat pump. Indoor VRF fan coils are sized to meet the calculated heating or cooling loads for each zone. The outdoor unit can be paired with a total indoor fan coil capacity up to 130% the outdoor rated capacity, as it is unlikely all indoor fan coils will operate at peak loads at the same time. In addition, sizing of the outdoor unit relative to the building load will also depend on the available sizes in the product line. For example, the GHP product line includes only two sizes, an 8-ton and a 15-ton unit, although multiple units can be combined for larger systems. When specifying the outdoor unit, heating or cooling capacity must be rounded up to the next available size.

For this field site, the 12-ton outdoor CCHP unit was oversized capacity to compensate for reduced heating capacity at cold ambient temperatures. The indoor VRF fan coil capacity (6.8 tons) was about 60% capacity of the outdoor unit, resulting in even lower part-load operation for the outdoor unit during cooling. The 8-ton outdoor GHP unit for this field site was paired with 90% indoor VRF fan coil capacity (7.2 tons).

The following graphs present the measured daily efficiencies for both heat pumps compared with part-load operation and ambient temperatures. For this assessment, part-load was calculated for every 5-minute data interval based on measured heating or cooling output relative to total indoor capacity at rating conditions (47°F heating /95°F cooling). At this field site, the GHP operated between 20% and 35% full capacity. Since the electric CCHP was oversized to meet the heating load, the system operated at slightly lower part-loads between 5% and 30%, for both heating and cooling. On the coldest days, both heat pump heating runtime increased up to 24 hours with only slight increases in part-load due to reduced capacity at colder conditions.

#### *5.6.3.1 CCHP Part-load Performance*

Figure 29 shows how CCHP cooling efficiencies vary with respect to part-load and cooling degree days. Daily average cooling efficiency for the CCHP ranged from 8 to 15 EER in agreement with the specified 12.3 EER rating for 95°F (35°C). CCHP cooling efficiency was impacted more by part-load (left graph) than ambient temperatures (right graph). For this range of operating conditions, from 30% to 10% part-load, CCHP cooling efficiency decreased with lower part-load. Manufacturer data indicates CCHP cooling performance increases above rated efficiencies for part-load operation down to 40 or 50%; however, for this configuration, the CCHP operated at much lower part-loads due to over-sizing with lower than rated efficiency below 20% part-load.

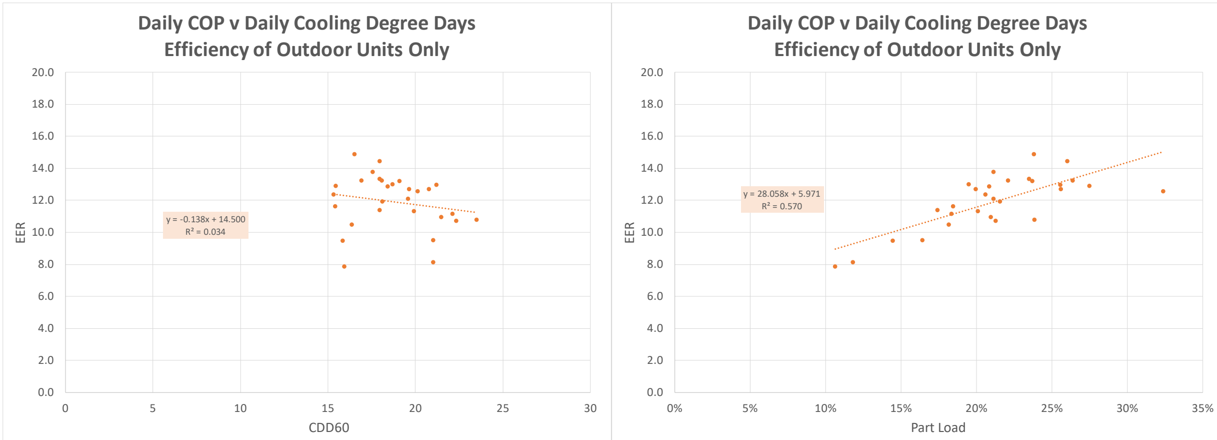



**Figure 29. (a) Left Graph Shows Minor Impact of Ambient Temperatures on CCHP Cooling Efficiency for this Range of Operating Conditions. (b) Right Graph Shows CCHP Decreasing Cooling Efficiency with Decreasing Part-load Operation.**

Daily average heating efficiency ranged from 0.5 to 4.0 COP, compared to manufacturer specifications of 4.1 COP at 47°F, 2.3 COP at 17°F. For heating, as shown in Figure 30, ambient temperatures had a larger impact on heating COP than part-load operation. Based on measured heating output, part-load operation ranged from 6% to 29% total indoor capacity.

On the other hand, Figure 30 illustrates how ambient temperatures had a larger impact on CCHP heating efficiency than part-load operation. The left graph shows a strong correlation between CCHP measured heating efficiency and ambient temperatures. The measured heating efficiency was slightly lower than specified ratings, likely due to part-load operation or actual installation conditions, e.g. defrost cycles. CCHP heating efficiently, based on average daily energy use, decreased from 3 COP at 47°F daily average OA temperatures, to 1 COP at 17°F. At colder temperatures (circled), the CCHP continued to operate but was unable to meet the heating load of the facility and could not maintain the temperature set points of the interior zones.

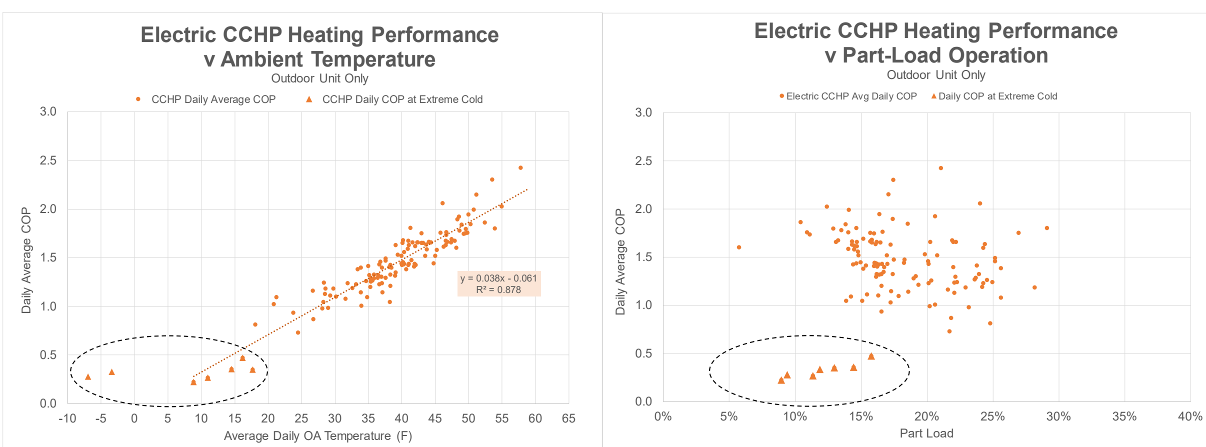



**Figure 30. (a) CCHP Heating Efficiency Decreasing with Ambient Temperatures (left graph). (b) Part-load Had Less Impact on CCHP Heating Performance (right graph).**

*CCHP was unable to meet the heating load at extreme cold temperatures (circled).*

### *5.6.3.2 GHP Part-load Performance*

Based on this demonstration, part-load operation had an adverse impact on overall efficiency for the NextAire™ GHP. Although GHPs use variable speed engines, field data showed reduced efficiency at part-load operation below 60% rated output. This extent of decreased performance at part-load is not seen in other engine-driven heat pump designs and is not inherent in this class of technology. This issue is likely due to product-specific controls and/or engine sizing.

Figure 32 shows GHP 5-minute interval data for gas consumption, cooling output, and COP for a representative day (July 13, 2018). At brief intervals near peak cooling output, the GHP cooling efficiency, COPg exceeded 1.0, approaching the manufacturer full-load performance rating of 1.3 COPg. Since the majority of runtime was at much lower cooling output, the daily average cooling efficiency was only 0.46 COPg. This data shows how the reduced part-load efficiency of the NextAire™ GHP resulted in higher energy use. For this example, the GHP average part-load operation (relative to rated cooling capacity) was about 9% compared to the range of 3% to 12% during July and August. GHPs use a variable speed engine which seems unable to modulate at these lower loads. This issue is not typical for GHP technologies in general and is likely due to product-specific control strategy.

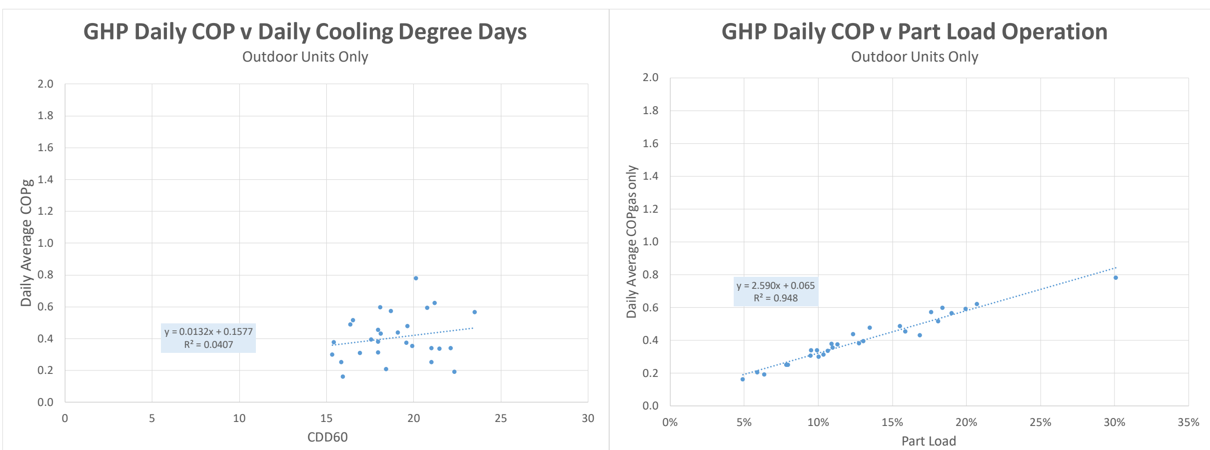



**Figure 31. (a) Minimal Impact of Ambient Temperatures on GHP Cooling Efficiency for this Range of Operating Conditions (Left Graph). (b) GHP Data Showed Decreasing Cooling Efficiency with Decreasing Part-load Operation (Right Graph).**

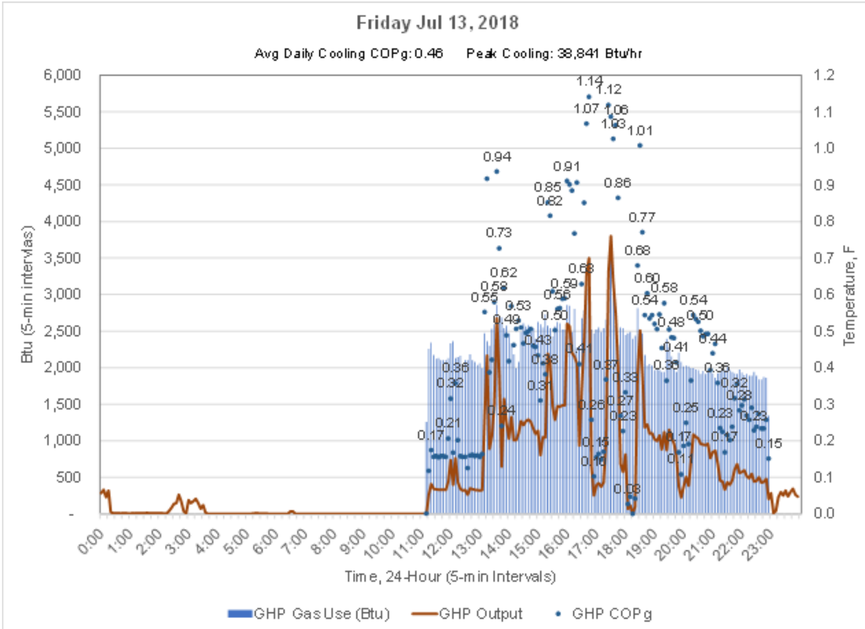



**Figure 32. GHP Data from July 13th: Cooling Output, Gas Consumption, and COP**

For heating, shown in Figure 33, GHP efficiency was impacted both by ambient temperatures and part-load operation. These graphs show a correlation between GHP measured heating efficiency and ambient temperatures, and to a lesser degree, with part-load. Based on measured heating output, part-load operation ranged from 18% to 35% total indoor capacity. The measured daily heating efficiency was less than specified rating of 1.3 COPg. The GHP was not operational during the coldest ambient temperatures due to unrelated component issues, so could not be evaluated at those extreme cold conditions.

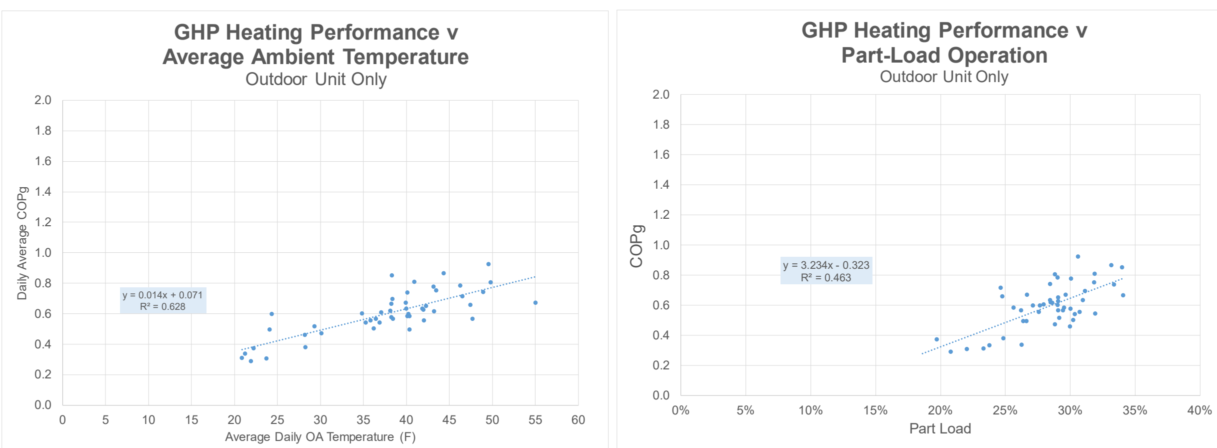



**Figure 33. Impact of Ambient Temperatures (Left Graph) and Part-load (Right Graph) on GHP Heating Efficiency for this Range of Operating Conditions.**

# 6.0 PERFORMANCE ASSESSMENT

## 6.1 METHODOLOGY

### 6.1.1 Data Analysis

To estimate annual energy and cost savings, measured energy use data was normalized to published TMY3 cooling and heating degree days based on the National Centers for Environmental Information NOAA 30-year Normals (1981-2010) [NCEI]. Energy use for the heat pump VRF systems was also normalized with respect to the total load of the building section for direct comparison of total energy use for each VRF system with the DOAS to the baseline VAV system.

GHP performance metrics were compared to CCHP to evaluate the demonstration performance objective. Energy consumption for both VRF/DOAS systems was also compared to the baseline VAV system. Primary energy and full-fuel-cycle GHG emissions were calculated based on estimated annual energy use. A summary of the normalized annual energy use estimated for each technology is presented in Table 9.

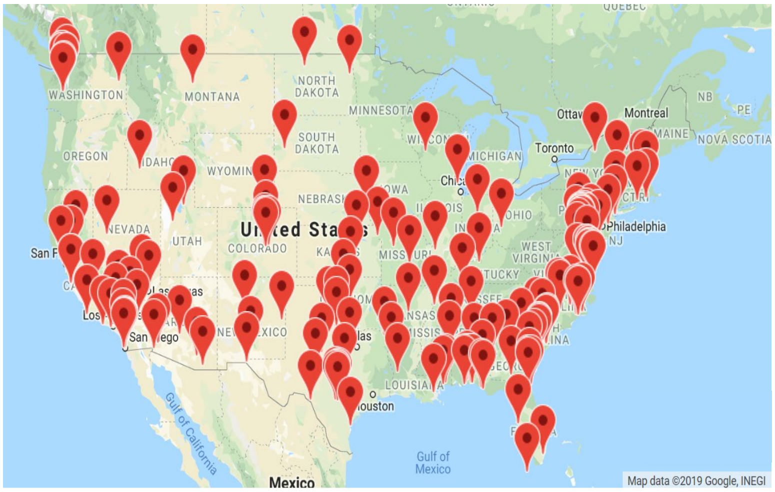
**Table 9. Normalized Annual Energy Use**

| | Gas Use (therm) | Electric Use (kWh) | Peak Electric Heating | Demand (kW) Cooling |
|---|---|---|---|---|
| **Baseline VAV with Electric Reheat** | **5,474** | **125,713** | **65** | **43** |
| Electric CCHP VRF | | 41,349 | | |
| DOAS | 2,651 | 9,326 | | |
| **Total CCHP/DOAS System** | **2,651** | **50,675** | **37** | **14** |
| GHP VRF | 3,409 | 8,791 | | |
| DOAS | 2,651 | 9,326 | | |
| **Total GHP/DOAS System** | **6,060** | **18,117** | **6.7** | **9.3** |

### 6.1.2 Energy Modeling

DoD military bases in the contiguous U.S. are located in multiple climates zones (Figure 34) with about 50 sites in colder climates and the remaining majority in moderate or hot climates. To apply these demonstration results to the broader DoD community, GTI conducted hourly simulations using EnergyPlus models to predict the energy savings and economic benefits of VRF/DOAS for various climates.

The EnergyPlus model for DOE reference building (small office) was modified to match the layout and floorspace of the field site. Existing EnergyPlus models for VAV systems were calibrated based on measured field data collected during the demonstration. Existing EnergyPlus heat pump VRF models were adapted for CCHP and GHP models based on measured cold climate field data and manufacturer data. More detailed descriptions of the EnergyPlus models and assumptions is included in Appendix C.




| Climate | DoD Military Bases |
|---|---|
| Very Cold | 8 |
| Cold | 37 |
| Warm | 63 |
| Hot | 58 |
| Marine | 11 |

https://www.military.com/base-guide/browse-by-location

**Figure 34. DoD Bases in Multiple U.S. Climate Zones**

EnergyPlus hourly modeled simulations were used to predict the energy and economic benefits for multiple climates. Table 10 presents the five climates used to compare the relative energy and cost savings of heat pumps and conventional HVAC for a small office building with similar construction to the demonstration site.

In addition to assessing the performance objectives for multiple climates, modeling provided a comparison of optimized baseline and demonstration equipment performance that is more representative of typical HVAC operation, correcting for some anomalies discovered at the field site. Since the existing VAV equipment did not have integrated controls for the gas burner and electric resistance heating elements, the baseline measured electricity use was higher than expected inflating the projected energy savings for the demonstration equipment. The EnergyPlus VAV model was based on a more typical operating strategy for commercial buildings. In addition, the amount of outside air ventilation was reduced to minimum design specifications for both the baseline and demonstration equipment. GHP models were developed for an alternate engine-drive design and control strategy to assess GHP performance in multiple regions without the part-load performance issues discovered during the demonstration. The EnergyPlus GHP model was validated based on manufacturer data and measured field data from other GTI demonstrations [unpublished]. For each region, the electric CCHP was sized to completely meet the peak heating load based on normalized weather data; however, as show in this demonstration, the CCHP capacity was not able to meet the heating load at more extreme temperatures that occurred during the two-year monitoring period.

**Table 10. Energy Modeling for U.S. Climates**

| City | ASHRAE Climate Zone | |
|---|---|---|
| Chicago, IL | Zone 5A | Cool-Humid |
| Helena, MT | Zone 6B | Cold-Dry |
| Seattle, WA | Zone 4C | Mixed-Marine |
| Houston, TX | Zone 2A | Hot-Humid |
| Los Angeles, CA | Zone 3B | Warm-Dry |

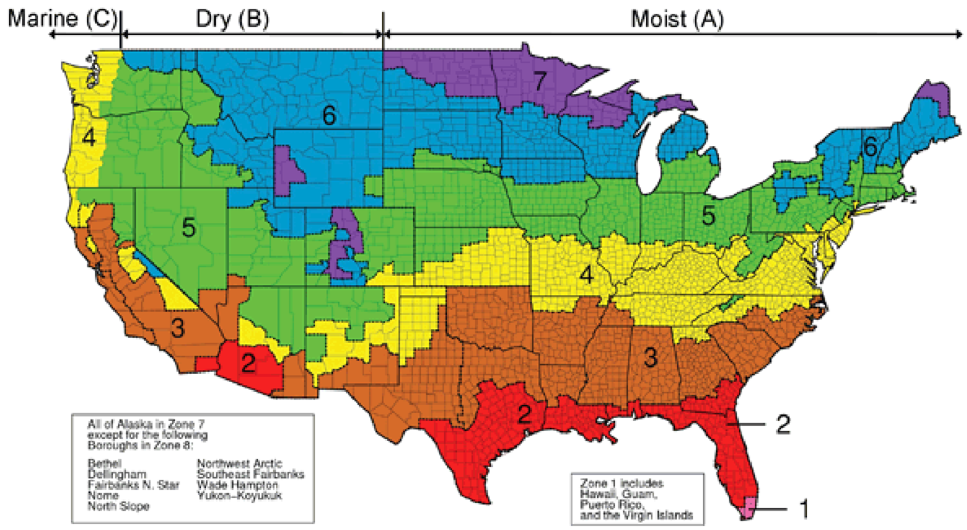



**Figure 35. ASHRAE Climate Zones**

## 6.2 ASSESSMENT OF PERFORMANCE OBJECTIVES

The following sections describe the data analysis for each Performance Objectives including both measured data and modeled energy performance for multiple climates.

### 6.2.1 Peak Electrical Load Reduction

The GHP/DOAS met the performance objective of exceeding 50% reduction in peak electric demand compared to the CCHP/DOAS. Both VRF/DOAS significantly reduced peak electric demand compared to the baseline by eliminating the electric resistance trim heating in the winter and the summer overcooling/electric reheat approach used in the VAV systems. (Table 11)

In heating operation, the GHP/DOAS significantly reduced peak electric demand relative to both the baseline VAV and the CCHP/DOAS. The baseline peak electric demand (65 kW) was higher than expected due to the lack of integrated controls, as previously discussed; however, the CCHP/DOAS heating operation also had a high winter peak electric demand (37 kW), much higher than the summer peak. For typical buildings, the highest electric demand occurs during the summer cooling operation due to conventional electric air conditioning; but, as shown by this demonstration, electric heating can create a secondary winter peak. With the growing use of electric heat pumps, the winter peak electric demand could exceed the summer cooling peak especially in cold climates.

If the CCHP/DOAS included supplement electric resistance heating at low ambient temperatures (e.g. daily HDD>30), the estimated winter peak electric demand would exceed 60 kW. During the monitoring period, this location experienced daily HDD>30 for eight days during 2017/2018 and for nine days the following winter, 2018/2019.

The GHP/DOAS peak electric demand is higher in cooling than heating operation due to the DOAS conventional electric air conditioner operation. This configuration still reduces peak electric demand by 36% with respect to the CCHP/DOAS and 79% with respect to the baseline.

**Table 11. Comparison of Peak Electric Demand**

| Normalized Measured Peak Electric Demand | Heating Peak Electric Demand (kW) | Cooling Peak Electric Demand (kW) |
|---|---|---|
| Baseline VAV with Electric Reheat | 65.4 | 43.2 |
| Total CCHP/DOAS System | 36.8 | 14.5 |
| Total GHP/DOAS System | 6.7 | 9.3 |
| CCHP Savings vs. Baseline | 28.6 (44% reduction) | 28.7 (66% reduction) |
| GHP Savings vs. Baseline | 58.7 (90% reduction) | 33.9 (79% reduction) |
| GHP Savings vs. CCHP | 30.1 (82% reduction) | 5.2 (36% reduction) |
| **Modeled Measured Peak Electric Demand** | **Heating Peak Electric Demand (kW)** | **Cooling Peak Electric Demand (kW)** |
| Baseline VAV with Electric Reheat | 56.9 | 30.1 |
| Total CCHP/DOAS System | 20.4 | 17.5 |
| Total GHP/DOAS System | 3.3 | 9.9 |
| CCHP Savings vs. Baseline | 36.5 (64% reduction) | 12.7 (42% reduction) |
| GHP Savings vs. Baseline | 53.6 (94% reduction) | 20.2 (67% reduction) |
| GHP Savings vs. CCHP | 17.1 (84% reduction) | 7.6 (43% reduction) |

Assumptions:

- Measured energy use normalized with respect to regional NOAA Annual Climate 30-yr Normals (1981 to 2010) CDD/HDD KUGN Waukegan National Airport [NCEI]. Heat pump energy use was normalized to the building load for direct comparison to the baseline.
- Modeled assumptions are presented in Appendix C.

Modeled results for different regions show GHP/DOAS had the lowest peak electric demand for heating or cooling (Figure 36). The magnitude of demand reduction provided by the GHP/DOAS is higher for colder climates (Chicago, Richmond) than warmer climates.

Both VRF/DOAS reduced peak electric demand compared to the baseline VAV due to elimination of electric resistance heating. For colder climates, CCHP/DOAS had a higher winter peak demand than summer.

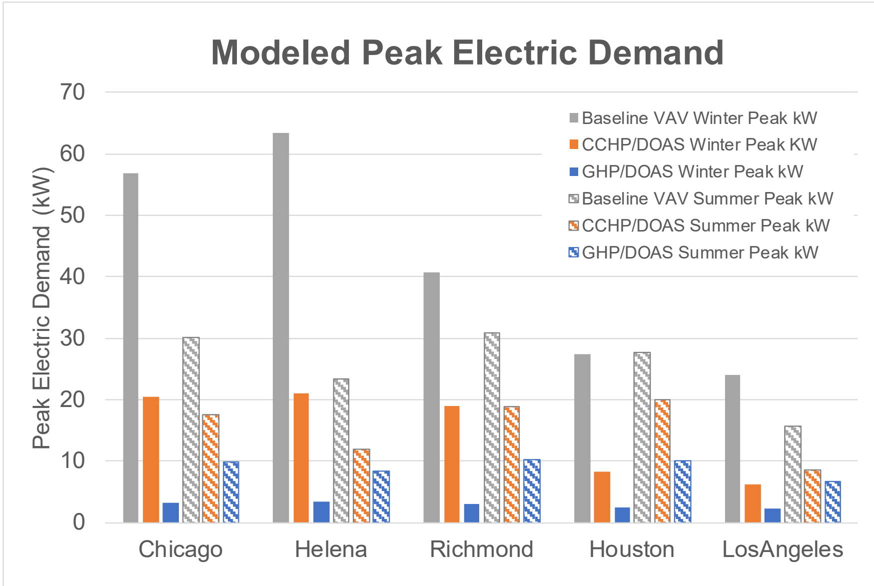



**Figure 36. Regional Modeled Comparisons for Peak Electric Demand**

### 6.2.2 Reduction in Annual Energy Costs

The GHP/DOAS met the performance objective of exceeding 25% reduction in annual energy costs compared to the CCHP/DOAS. Table 12 presents a comparison of projected annual energy costs based on both measured and modeled data for the field site. Based on measured data, the GHP/DOAS reduced energy costs by 71% compared to the baseline, and by 41% compared to the CCHP/DOAS. GHP/DOAS significantly reduced energy costs per square foot of the facility.

**Table 12. Comparison of Annual Energy Costs**

| Normalized Annual Energy Costs | Annual Fuel Costs | Demand Charge | Annual Total Energy Cost | Energy Cost $/s.f. |
|---|---|---|---|---|
| Baseline VAV with Electric Reheat | $9,714 | $8,086 | $17,800 | $2.21 |
| Total CCHP/DOAS System | $4,134 | $4,547 | $8,680 | $1.08 |
| Total GHP/DOAS System | $3,983 | $1,147 | $5,129 | $0.64 |
| CCHP Savings vs. Baseline | | | $9,120 (51% savings) | |
| GHP Savings vs. Baseline | | | $12,671 (71% savings) | |
| GHP Savings vs. CCHP | | | $3,551 (41% savings) | |
| **Modeled Annual Energy Costs** | **Annual Fuel Costs** | **Demand Charge** | **Annual Total Energy Cost** | **Energy Cost $/s.f.** |
| Baseline VAV with Electric Reheat | $6,481 | $7,031 | $13,511 | $1.68 |
| Total CCHP/DOAS System | $2,911 | $2,522 | $5,433 | $0.67 |
| Total GHP/DOAS System | $2,743 | $1,224 | $3,967 | $0.49 |
| CCHP Savings vs. Baseline | | | $8,078 (45% savings) | |
| GHP Savings vs. Baseline | | | $9,544 (71% savings) | |
| GHP Savings vs. CCHP | | | $1,466 (27% savings) | |

Assumptions: $0.0559/kWh; $10.3037/kW; $0.49/therm

Energy prices were based on incremental composite rates provided by the field site. Utility demand charges and rate structures can vary widely from state to state and will impact energy cost savings. Demand charges for the field site are based on the highest hourly peak kW during the past 12 months, whether summer or winter. For this demonstration, the highest peak electric demand occurred during the heating season for both the baseline VAV system (65 kW) and demonstration CCHP/DOAS (37 kW), resulting in significantly higher demand charges compared to the GHP/DOAS peak demand of 9.3 kW in the summer.

Modeled results for different regions show GHP/DOAS had the highest savings in annual energy costs for cold climates (Figure 37), and slightly lower savings for moderate and hot climates with high cooling loads (i.e. Houston and Los Angeles). As expected, modeled energy savings were lower than measured with respect to a more optimized VAV operation baseline.

Annual cost savings are a function of energy use as well as energy prices and maintenance costs. The magnitude of annual costs will vary with different energy pricing especially demand charges. Annual energy costs for CCHP/DOAS would increase if electric resistance heat was used for supplemental heating at low ambient temperatures. Based on this field site, the use of supplemental electric resistance heat at low ambient temperatures (i.e. daily HDD>30) would require an estimated 1591 kWh and increase peak electric demand to 60 kW, resulting in 34% higher annual energy costs ($2,936) largely due to higher demand charges.

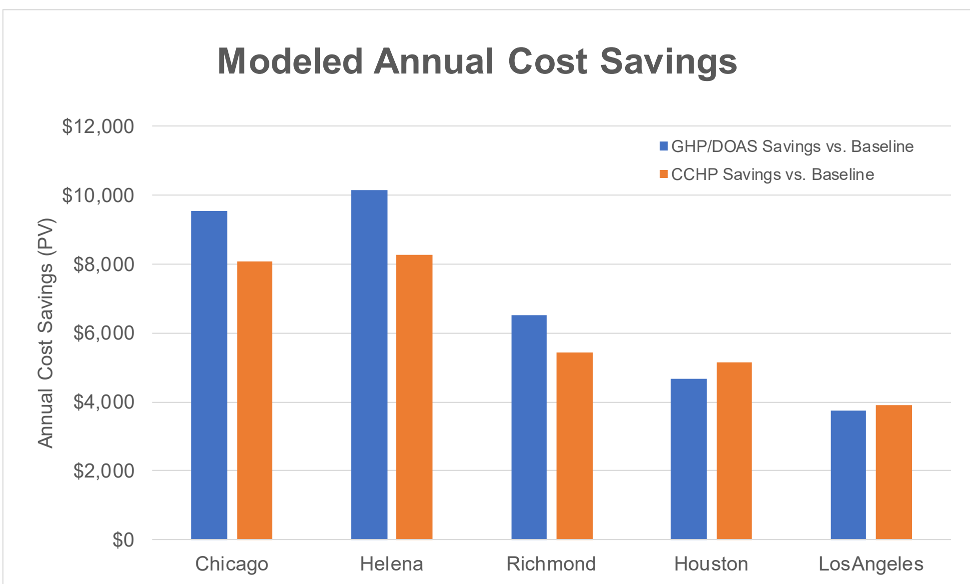



**Figure 37. Regional Modeled Comparisons for Annual Cost Saving (PV)**

#### 6.2.3 System Economics

The NIST Building Life-cycle Cost (BLCC) program was used to model savings in life-cycle cost impact (LCC) for each demonstrated technology. Input assumptions are described in detail in Section 7.0. As shown in Table 13, the GHP/DOAS met the performance objective of lower life-cycle costs compared to the CCHP/DOAS based on measured data. Modeled results for Chicago climate predicted a 4% increase in life-cycle costs for the GHP with respect to CCHP.

**Table 13. Comparison of Life-cycle Costs**

| Life-cycle Costs Based on Measured Data | LCCA Annual Savings (PV) | Lifecycle Cost Savings |
|---|---|---|
| CCHP Savings vs. Baseline | $8,888 | $84,660 (25% savings) |
| GHP Savings vs. Baseline | $11,937 | $96,241 (29% savings) |
| GHP Savings vs. CCHP | $3,048 | $11,581 (4% savings) |
| **Life-cycle Costs Based on Modeled Results** | **LCCA Annual Savings (PV)** | **Lifecycle Cost Savings** |
| CCHP Savings vs. Baseline | $7,785 | $71,424 (21% savings) |
| GHP Savings vs. Baseline | $8,970 | $60,640 (18% savings) |
| GHP Savings vs. CCHP | $1,185 | -$10,784 (-4% savings) |

Assumptions:
- 15-year equipment life with linear depreciation;
- Default BLCC inputs: Real Discount Rate: 3.1%; Nominal Discount Rate: 3.0%; Inflation Rate: 0.1% Investment Cost; Annual rate of increase: 0.9 %
- Energy prices: $0.0559/kWh; $10.3037/kW; $0.49/therm

Based on current energy pricing, modeled results for different regions show both VRF systems reduced life cycle costs as compared to the baseline VAV system. Savings in life-cycle costs were highest in cold climates (e.g. Chicago, Helena). The GHP/DOAS had savings in life-cycle costs for all regions except Los Angeles. (Figure 38) The CCHP/DOAS had the lowest life-cycle costs in all locations. As previously discussed, life-cycle costs for CCHP/DOAS would significantly increase if supplemental heating was added for operating low ambient temperatures.

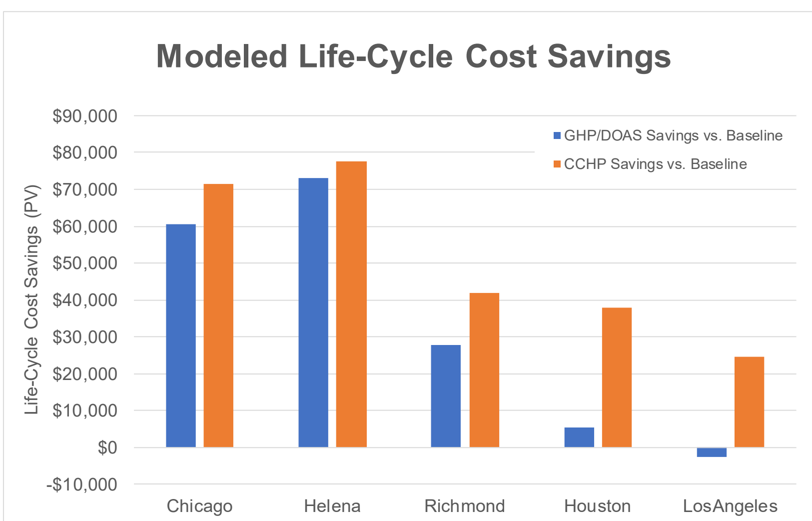



**Figure 38. Regional Modeled Comparisons for Savings in Life-Cycle Costs**

### 6.2.4 Reduction in Primary Energy Use

Primary energy use (i.e. full-fuel-cycle) accounts for all upstream energy used to generate power or to supply fuel to the building meter, i.e. includes all energy for fuel extraction (natural gas, oil, coal, renewables), conversion (e.g. power generation), and distribution (pipelines, power transmission lines). Primary energy is a more comprehensive approach to evaluate energy use and is more relevant to energy security for DoD facilities than the energy metered at the site. A growing number of codes and standards are shifting to the use of full fuel cycle or primary energy use as the metric to quantify the environmental impact of appliances, rather than site energy.

For this analysis, primary energy was calculated using estimated annual energy use and primary energy factors for electricity and natural gas. Electricity primary energy factors were based on published non-baseload generation mix for eGrid region RFCW corresponding to the demonstration site. Natural gas primary energy factors were based on published regional average. [SEEAT] As shown in Table 14, both VRF/DOAS system had significantly lower natural gas and electricity use compared to the baseline VAV, reducing primary energy use by 54-57%. This confirms published VRF modeling studies that report savings from 20% to 60% relative to VAV systems [EES Consulting]. Modeled results based on optimized equipment predict similar percent savings in primary energy.

Despite lower than expected part-load performance, the GHP/DOAS has similar primary energy use compared to the CCHP/DOAS but did not meet the performance objective of 10% savings in primary energy.

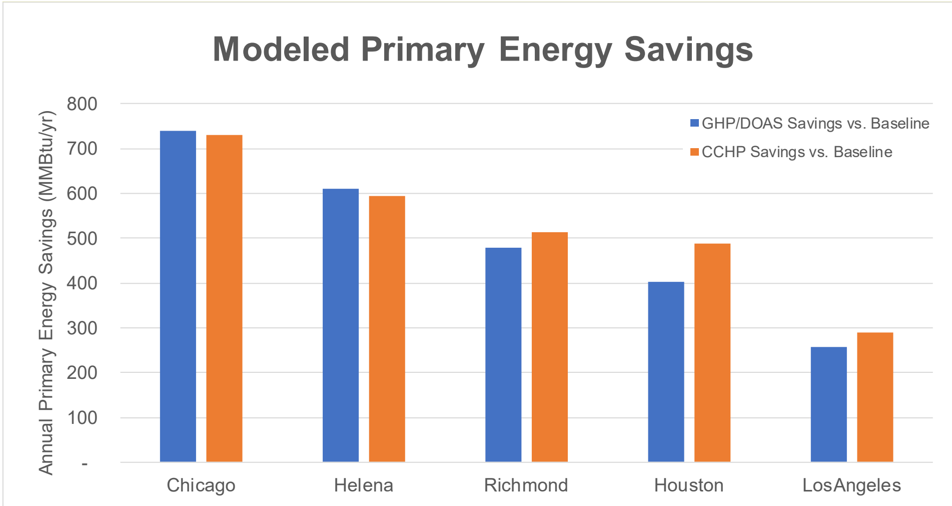
Modeled results show significant savings in primary energy for both VRF systems compared to the baseline VAV for all regions (Figure 39). Primary energy savings for the VRF/DOAS ranged from 47% to 62% compared to baseline. GHP/DOAS primary energy savings were slightly higher than the CCHP/DOAS in colder climates, and slightly lower in warmer regions. Primary energy was calculated using modeled energy use and regional primary energy factors based on non-baseload electricity mix per eGrid 2016.

**Table 14. Comparison of Annual Primary Energy Use**

| **Primary Energy Based on Measured Data** | **(MMBtu/yr)** |
|---|---|
| CCHP Savings vs. Baseline | 1142 (57% reduction |
| GHP Savings vs. Baseline | 1133(57% reduction |
| GHP Savings vs. CCHP | -9.4 (-1% reduction |
| **Primary Energy Based on Modeled Results** | **(MMBtu/yr)** |
| CCHP Savings vs. Baseline | 728 (55% reduction |
| GHP Savings vs. Baseline | 737 (56% reduction |
| GHP Savings vs. CCHP | 9.7 (2% reduction |

Assumptions: Natural gas: 1.09 Btu/Btu primary energy

Non-baseload electricity mix [eGrid RFCW 2016]: 3.26 Btu/Btu primary energy




Assumptions: Primary energy calculated based on regional primary energy factors (Btu/Btu): Non-baseload electricity mix per eGrid 2016: RFCW: 3.26; NWPP: 3.04; SRVC: 2.92; ERCT: 2.83; CAMX: 2.68; Natural gas: 1.09 [SEEAT]

**Figure 39. Regional Modeled Comparisons for Primary Energy Savings**

### 6.2.5 Direct Full Fuel Cycle GHG Emissions

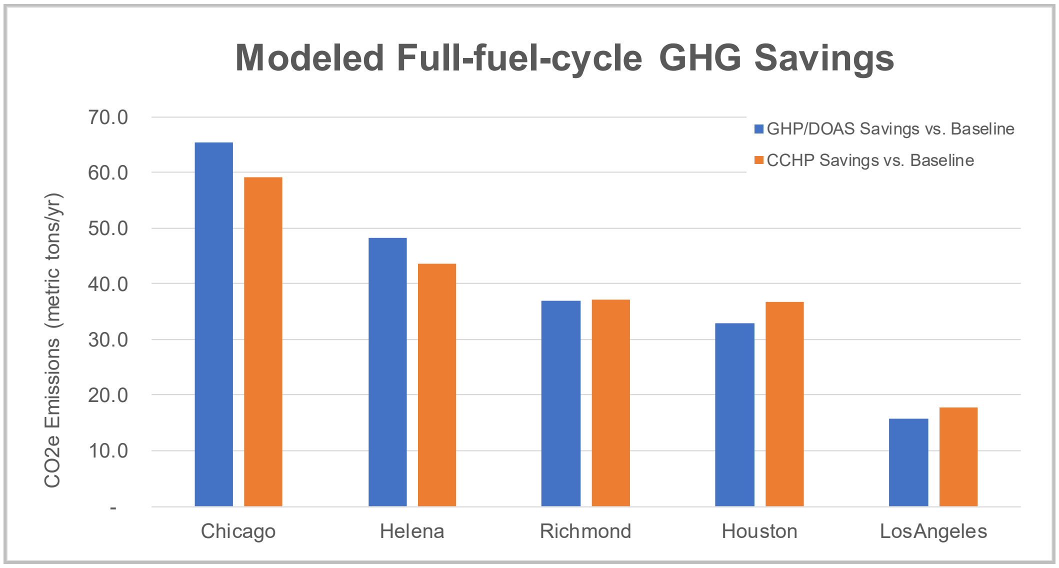
Both VRF/DOAS systems significantly reduced full-fuel-cycle GHG emissions up to 63% as shown in Table 15. Full-fuel-cycle GHG emissions were calculated based on estimated annual energy use and regional CO2e emission factors to estimate all upstream GHG emissions from power generation, transmission or distribution of the energy supply to the building meter. Emission factors for electricity were based on published non-baseload power generation mix for eGrid regions. Natural gas emission factors were also based on the published regional average. [SEEAT] The GHP/DOAS had similar full-fuel-cycle GHG emissions compared to the CCHP/DOAS but did not meet the performance objective of 20% GHG reduction due to lower than expected part-load performance. GHP engine exhaust emissions were also not included due to lack of available data but would slightly increase full-fuel-cycle GHG.

**Table 15. Full-Fuel-Cycle GHG Emissions**

| Annual CO2e Based on Measured Data | (Metric tons/yr) |
|---|---|
| CCHP Savings vs. Baseline | 91.5 (55% reduction) |
| GHP Savings vs. Baseline | 100.2 (63% reduction) |
| GHP Savings vs. CCHP | 8.7 (13% reduction) |
| **Annual CO2e Based on Modeled Results** | **(Metric tons/yr)** |
| CCHP Savings vs. Baseline | 59.1 (55% reduction) |
| GHP Savings vs. Baseline | 65.4 (61% reduction) |
| GHP Savings vs. CCHP | 6.3 (13% reduction) |

Assumptions: Full-fuel-cycle CO2e emissions based on regional emission factors (lbs/MWh): Non-baseload electricity mix per eGrid 2016: RFCW: 2138; Natural gas: 147 lb/MMBtu [SEEAT]

Modeled results show significant savings in full-fuel-cycle GHG for both VRF systems compared to the baseline VAV for all regions (Figure 40). VRF/DOAS generated savings in GHG emissions from 51% to 62% compared to baseline. For the GHP/DOAS, GHG savings were slightly higher than the CCHP/DOAS in colder climates, and slightly lower in warmer regions. Regional GHG emission factors were used to estimate full-fuel-cycle GHG emissions based on non-baseload electricity mix per eGrid 2016.




Assumptions: Full-fuel-cycle CO2e emissions based on regional emission factors (lbs/MWh): Non-baseload electricity mix per eGrid 2016: RFCW: 2138; NWPP: 1799; SRVC: 1660; ERCT: 1625; CAMX: 1235; Natural gas: 147 lb/MMBtu [SEEAT]

**Figure 40. Regional Modeled Comparisons for Full-fuel-cycle GHG Emissions**

### 6.2.6 Comfort & Reliability

The heating capacity and efficiency of all air source heat pumps decrease with lower ambient temperatures. For this demonstration, the CCHP was unable to meet the heating load for several days when the daily average temperatures were at or below 16°F, indicating the need for supplemental heating. The Daikin CCHP is designed for cold climates without supplemental heating and has a minimum rated temperature of -4°F. For this location, the ASHRAE 99% design temperature is -5°F; however, during the demonstration period, minimum temperatures exceeded -5°F on several occasions including historic cold temperatures of -23°F during the January 2019 Polar Vortex.

CCHP was unable to meet the heating load for seven days when daily average temperatures were below 16°F. Figure 41 shows the decrease in heating delivered (circled) at very cold temperatures. Under the coldest conditions, indoor zones ranged from 53°F to 62°F. This demonstrates that supplemental heating is needed for cold climate (ASHRAE Zone 5) despite the product cold climate specifications and the design approach to oversizing the outdoor unit.

During this period, both the DOAS and GHP were not operational due to unrelated installation and component issues, so we were unable to assess the cold climate performance of this equipment under the same conditions. Two weeks prior, refrigerant leaks from the threaded meter connections were repaired and both heat pumps were charged with new refrigerant. Subsequently, GHP fault codes indicated low oil, likely due to the earlier refrigerant leaks, and a possible faulty compressor discharge temperature sensor. The GHP sensor was ordered and replaced as soon as possible; however, this occurred after the Polar Vortex period. The DOAS was out of service due to a failed ignition board, diagnosed prior to the extreme cold. A new ignition board, spark rod, and flame sensor were ordered and replaced under equipment warranty. DOAS operation supplying 64°F ventilation would have provided some supplemental heat and increased zone temperatures.

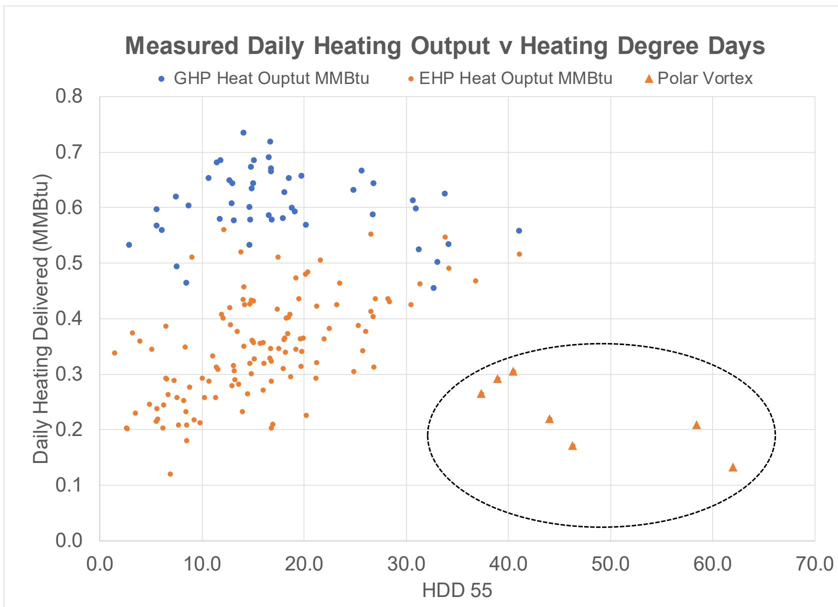



**Figure 41. CCHP Was Unable to Meet the Heating Load for Several Days.**

During the demonstration, the GHP had lower than expected performance at low part-load operation which impacted comfort and reliability. VRF systems were carefully sized for the field site; however, by design, VRF systems frequently operate at low part-loads, modulating to meet the temperature setpoints for multiple zones. GHP technologies use a variable-speed natural gas engine which can modulate to meet variable heating or cooling loads. Based on performance data from other GHP products, decreased performance at low part-load operation is not an inherent issue for GHPs and may be due to product-specific control strategies.

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## 7.0 COST ASSESSMENT

GTI developed a cost assessment for replacing an existing VAV system at end of equipment life with either a GHP VRF system combined with a DOAS or an electric CCHP VRF and DOAS system. The cost assessment also compared the performance of the gas and electric heat pump VRF systems. Cost models were based on measured data as well as modeled simulations to determine the economic benefits for the demonstration technologies in different climate regions.

### 7.1 COST MODEL

The cost elements for VRF/DOAS systems are listed in Table 13. Facility operational costs were estimated based on estimated annual energy use and NSGL utility rates. Hardware capital costs are based on actual equipment costs for the demonstration. Installation cost are based on rule of thumb estimates for VRF systems. Manufacturers estimate VRF installation costs at $4000/ton, while the design firm estimated installation at $8000/ton reflecting Chicago area labor rates. This cost assessment assumed the mid-point $6000/ton. DOAS installation costs were based on RS Means 2016 estimates for an RTU retrofit to existing ductwork. Consumables for the installation including refrigerant (R410A) and oil, are typically included in the installation costs, but may be required for service or maintenance. Estimated costs for consumables are based on project costs.

**Table 16. Cost Elements for Demonstration VRF/DOAS**

| Cost Element | Data Tracked During the Demonstration | Estimated Costs |
|---|---|---|
| Hardware capital costs | NextAire™ 8-ton electric cold climate heat pump with ten VRF fan coil units based on actual costs including $2500 scheduled maintenance plan | $44,283 |
| | Daikin 12-ton electric cold climate heat pump with 10 VRF fan coil units based on actual costs. Note capital costs include iTouch controller and 12-ton heat pump oversized for approx. 10-ton load | $20,680 |
| | Standard efficiency DOAS (6-ton) retrofit to existing ductwork | $19,715 |
| Installation costs | Both VRF systems based on mature market rule-of-thumb ranging from $4K-$8K/ton | $6000 per ton |
| | DOAS installation costs similar to RTU retrofit to existing ductwork est. $2070/ton [RS Means 2016 Section D3050 150 Single Zone Office] | $2,070 per ton |
| Consumables | Refrigerant R410A<br>Compressor oil | $845 for 20RT VRF<br>$212 per 0.5L |
| Facility operational costs | Reduction in energy required vs. baseline data | $5K-$7K per year |
| Maintenance | Frequency of required maintenance | Annual |
| | Assume similar maintenance costs for GHP, CCHP, and VAV:<br>- Baseline: Inspect/adjust VAV boxes (19); replace filters<br>- Both VRF/DOAS: Check VRF fan coil units (20); clean condensate; replace filters<br>- GHP engine maintenance at 10K, 20K, and 30K runtime hours included in scheduled maintenance plan in capital costs ($2500) | $ 100 material<br>$1100 labor<br>**$1200 Total** |
| Hardware lifetime | Equipment life:<br>- VAV<br>- VRF/DOAS<br>- GHP/DOAS: est. 30,000 hours or 15 years | 15 years |
| Operator and service training | Manufacturer training costs for service providers<br>(2-day training and commissioning) | $5,287 |

GHP natural gas engine has additional service requirements. To address customers' concern about unexpected maintenance costs, one manufacturer offers a service agreement included with the equipment purchase ($2500). This scheduled maintenance plan provides engine maintenance (oil change, belts, etc.) at intervals of 1000, 2000, and 3000 runtime hours. This option was selected for the demonstration. Alternatively, incremental maintenance costs for GHP can be estimated at $1000/year. This assessment assumes engine maintenance is coved by the service plan which is included in the GHP capital equipment costs.

Aside from the engine maintenance, both VRF/DOAS maintenance tasks differ from more central systems such as VAV. Some sources predict VRF/DOAS have higher maintenance costs than conventional equipment [Thorton], while other publications expect similar or lower maintenance costs [Goetzler]. Typical baseline maintenance tasks include annually inspecting or adjust VAV boxes and replacing filters. The VRF/DOAS maintenance task include checking VRF indoor fan coil units, cleaning condensate tubing, replacing filters. Increasing the number of fan coil units can significantly increase maintenance costs; however, the design for this field site included twenty VRF fan coil units, comparable to the baseline nineteen VAV-boxes. This economic assessment assumed similar costs for baseline VAV and VRF maintenance. Based on conversations with the manufacturer and facility staff, it was assumed no other repairs or refrigerant replacement were needed over the 15-year equipment lifetime.

## 7.2 COST DRIVERS

### 7.2.1 Installed Costs

Cost drivers for the VRF/DOAS equipment include the existing HVAC equipment. For the demonstration, the DOAS was able to use existing ductwork significantly reducing installation costs. New installations will have lower incremental costs than retrofit applications. Based on this demonstration, electric CCHP VRF systems will require supplemental heat in cold climate applications, which will increase installed costs. Based on conversations with the manufacturer, electric resistance heating is not available with the cassette-style indoor fan coils used in the demonstration. Optional electric resistance heating is available for some ducted VRF fan coils at approximately $300 per unit. In addition, any electric service upgrades required to meet the higher peak electric demand (60 kW) would significantly increase installed costs. Electric CCHP VRF systems require 460V 3Ph power with capacity of 20 to 30 kW. On the other hand, GHP VRF systems require only 208V 1Ph power with much lower electric demand offering added benefits for sites with power constraints.

### 7.2.2 Energy Prices

Figure 42 shows a sensitivity analysis of energy savings with respect to natural gas prices, electricity prices, and demand charge. Energy prices for the demonstration site are highlighted. Negative costs savings (i.e. increased energy costs) are shown in red, with net energy cost savings shown in green. The top table shows the GHP annual energy cost savings are positive relative to the baseline. Energy costs savings increase with higher electricity pricing and demand charges. As shown in the lower table, compared to the CCHP, demand charges must be present for the GHP to generate energy cost savings. This illustrates how both demand charges and electricity rates relative to gas pricing are key factors in the GHP annual energy cost savings.

### 7.2.3 Climate Impact

As discussed in Sections 6.2.2 and 6.2.3, both VRF/DOAS systems have potential to reduce annual energy costs and life-cycle costs compared to the baseline VAV. Modeled results show the GHP/DOAS had higher savings in annual energy costs for cold and moderate climates (Chicago, Helena, Richmond) than the CCHP/DOAS, but not for warmer climates with high cooling loads (Houston, Los Angeles). Likewise, the GHP/DOAS had lower life-cycle costs in colder and the CCHP/DOAS had lower life-cycle costs in Richmond, Houston, and Los Angeles. The GHP provides higher efficiency heating on a primary energy basis, and the CCHP provides higher efficiency cooling. As discussed in previous sections, annual cost savings and life-cycle costs will be highly dependent relative electric/gas pricing and demand charges.

**Net Annual Energy Cost Savings**

| GHP v Baseline | | Electricity ($/kWh) | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | 0.04 | | | 0.056 | | | 0.08 | | | 0.10 | | | 0.12 | | | 0.14 | |
| Winter Demand ($/kW) | | $0 | $5 | $10 | $0 | $5 | $10.3 | $0 | $5 | $10 | $0 | $5 | $10 | $0 | $5 | $10 | $0 | $5 | $10 |
| Gas Cost | 0.40 | 4,069 | 7,593 | 11,117 | 5,780 | 9,304 | 13,042 | 8,373 | 11,897 | 15,421 | 10,525 | 14,049 | 17,573 | 12,677 | 16,201 | 19,725 | 14,829 | 18,353 | 21,877 |
| ($/therm) | 0.49 | 4,017 | 7,541 | 11,064 | 5,728 | 9,251 | 12,989 | 8,321 | 11,844 | 15,368 | 10,472 | 13,996 | 17,520 | 12,624 | 16,148 | 19,672 | 14,776 | 18,300 | 21,824 |
| | 0.60 | 3,952 | 7,476 | 11,000 | 5,663 | 9,187 | 12,925 | 8,256 | 11,780 | 15,304 | 10,408 | 13,932 | 17,456 | 12,560 | 16,084 | 19,608 | 14,712 | 18,236 | 21,760 |
| | 0.70 | 3,894 | 7,418 | 10,941 | 5,604 | 9,128 | 12,866 | 8,198 | 11,721 | 15,245 | 10,349 | 13,873 | 17,397 | 12,501 | 16,025 | 19,549 | 14,653 | 18,177 | 21,701 |
| | 0.80 | 3,835 | 7,359 | 10,883 | 5,546 | 9,070 | 12,808 | 8,139 | 11,663 | 15,187 | 10,291 | 13,815 | 17,339 | 12,443 | 15,967 | 19,490 | 14,595 | 18,119 | 21,642 |
| | 0.90 | 3,777 | 7,300 | 10,824 | 5,487 | 9,011 | 12,749 | 8,080 | 11,604 | 15,128 | 10,232 | 13,756 | 17,280 | 12,384 | 15,908 | 19,432 | 14,536 | 18,060 | 21,584 |
| | 1.00 | 3,718 | 7,242 | 10,766 | 5,429 | 8,953 | 12,690 | 8,022 | 11,546 | 15,069 | 10,174 | 13,698 | 17,221 | 12,326 | 15,849 | 19,373 | 14,478 | 18,001 | 21,525 |

| GHP v CCHP | | Electricity ($/kWh) | | | | | | | | | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | 0.04 | | | 0.056 | | | 0.08 | | | 0.10 | | | 0.12 | | | 0.14 | |
| Winter Demand ($/kW) | | $0 | $5 | $10 | $0 | $5 | $10.3 | $0 | $5 | $10 | $0 | $5 | $10 | $0 | $5 | $10 | $0 | $5 | $10 |
| Gas Cost | 0.40 | (61) | 1,745 | 3,552 | 457 | 2,263 | 4,179 | 1,241 | 3,048 | 4,854 | 1,892 | 3,699 | 5,505 | 2,543 | 4,350 | 6,157 | 3,195 | 5,001 | 6,808 |
| ($/therm) | 0.49 | (368) | 1,439 | 3,245 | 150 | 1,956 | 3,873 | 934 | 2,741 | 4,547 | 1,586 | 3,392 | 5,199 | 2,237 | 4,043 | 5,850 | 2,888 | 4,694 | 6,501 |
| | 0.60 | (743) | 1,064 | 2,870 | (225) | 1,581 | 3,498 | 559 | 2,366 | 4,173 | 1,211 | 3,017 | 4,824 | 1,862 | 3,668 | 5,475 | 2,513 | 4,319 | 6,126 |
| | 0.70 | (1,084) | 723 | 2,529 | (566) | 1,240 | 3,157 | 219 | 2,025 | 3,832 | 870 | 2,676 | 4,483 | 1,521 | 3,327 | 5,134 | 2,172 | 3,979 | 5,785 |
| | 0.80 | (1,425) | 382 | 2,188 | (907) | 900 | 2,816 | (122) | 1,684 | 3,491 | 529 | 2,335 | 4,142 | 1,180 | 2,987 | 4,793 | 1,831 | 3,638 | 5,444 |
| | 0.90 | (1,766) | 41 | 1,848 | (1,248) | 559 | 2,475 | (463) | 1,343 | 3,150 | 188 | 1,995 | 3,801 | 839 | 2,646 | 4,452 | 1,490 | 3,297 | 5,103 |
| | 1.00 | (2,106) | (300) | 1,507 | (1,589) | 218 | 2,134 | (804) | 1,002 | 2,809 | (153) | 1,654 | 3,460 | 498 | 2,305 | 4,111 | 1,149 | 2,956 | 4,763 |

**Figure 42. Sensitivity Analysis of Energy Prices and Demand Charges**

### 7.2.4 Propane Operation

While the majority of DoD installations have natural gas utility infrastructure and can use natural gas for space conditioning, about a dozen installations in the U.S. use propane instead of natural gas. Based on the 2018 DoD Annual Facilities Energy Management Report, propane or LPG makes up only 0.45% of the total installation energy consumption. Installations without natural gas service include a few locations such as Alaska, Florida, Georgia, Nevada, North Carolina, Utah and Hawaii. (See Appendix E). For sites without natural gas service, the GHP can be operated on propane instead of natural gas. Propane operation is not expected to impact maintenance costs or equipment life.

For propane operation, the NextAire™ GHP uses an exhaust aftertreatment system (referred to as a deodorizer) which reacts with the unburned hydrocarbons and NOx to reduce emissions. The manufacturer reports no change in performance with propane for this model GHP; however, other gas engine-drive heat pump models, such as the packaged rooftop unit, report 5-10% lower efficiency. The most significant difference would be higher propane fuel costs compared to natural gas. For 2019, U.S. wholesale propane weekly prices ranged from $6.39 to $9.91 per MMBtu ($0.584 to $0.906 per gallon) compared to the U.S. average for natural gas at $7.46 per MMBtu [EIA].

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# 8.0 COST ANALYSIS AND COMPARISON

## 8.1 DEMONSTRATION COST ASSESSMENT

The NIST Building Life-cycle Cost (BLCC) program was used to model the building specific simple payback, savings to investment ratio (SIR), and life-cycle cost impact (LCC) for each demonstrated technology. A list of input assumptions is shown in Table 17. Energy prices were based on incremental composite rates provided by the field site. Utility demand charges and rate structures can vary widely from state to state and will impact energy cost savings. Demand charges for the field site were applied monthly based on the highest hourly peak kW during the past 12 months, whether summer or winter.

**Table 17. Input Assumptions for BLCC Cost Assessment**

| BLCC Assumptions: |
|---|
| - 15-year equipment life (2034)<br>- Default BLCC inputs: Real Discount Rate: 3.1%; Nominal Discount Rate: 3.0%; Inflation Rate: 0.1% Investment Cost; Annual rate of increase: 0.9 %<br>- NSGL Utility Rates: Electricity \$0.0559/kWh; \$10.3037/kW demand; Natural Gas \$ 4.90/MMBtu<br>- U.S. Average 2019 Propane \$0.0774/gal; \$8.47/MMBtu |

Table 18 presents the life-cycle assessment for both VRF/DOAS systems compared to the baseline VAV system based on measured data. For this demonstration, both VRF systems (CCHP and GHP) were cost effective in reducing life cycle costs with simple paybacks under five years and SIR over 4.

**Table 18. Life-cycle Assessment of VRF/DOAS Systems with Respect to Baseline VAV Based on the Demonstration Measured Data**

| | Equipment Premium | First Cost Premium | Annual Savings (PV) | Life-Cycle Cost Savings (PV) | | Simple Payback (years) | SIR |
|---|---|---|---|---|---|---|---|
| GHP / DOAS wrt VAV Baseline | \$32,000 | \$47,000 | \$11,937 | \$96,241 | 29% | 4 | 4.0 |
| Electric CCHP / DOAS wrt VAV Baseline | \$9,600 | \$22,000 | \$8,888 | \$84,660 | 25% | 3 | 4.9 |

Table 19 presents the life-cycle assessment comparing the GHP/DOAS to the CCHP/DOAS based on measured data. Despite lower than expected part-load performance, the GHP/DOAS had higher life-cycle costs than the CCHP/DOAS. This comparison does not include supplemental heat for the CCHP/DOAS during extreme cold in terms of additional capital costs or energy consumption which would negatively impact the CCHP/DOAS economics.

**Table 19. Life-cycle Assessment of GHP/DOAS with Respect to Electric CCHP/DOAS Based on the Demonstration Measured Data**

| | Equipment Premium | First Cost Premium | Annual Savings (PV) | Life-Cycle Cost Savings (PV) | | Simple Payback (years) | SIR |
|---|---|---|---|---|---|---|---|
| GHP / DOAS wrt CCHP / DOAS | $22,400 | $25,000 | $3,048 | $11,581 | 4% | 8 | 1.5 |

The addition of CCHP supplemental electric resistance heat was estimated to increase annual energy costs by $2937 and add $3000 to installed cost. This increases CCHP life-cycle costs by 13%, reducing the life-cycle costs savings relative to the baseline from 25% savings to 13% savings.

## 8.2 REGIONAL COST ASSESSMENT

Hourly energy simulations were conducted for five different climates to predict regional energy and economic benefits for the VRF/DOAS systems. In addition, EnergyPlus equipment models were adapted to measured field data from the demonstration and optimized to be more representative of typical HVAC operation. Modeled results in Table 20 show GHP/DOAS compared to baseline VAV was cost-effective savings in moderate and cold climates (e.g. Chicago, Helena). GHP/DOAS reduced annual cost in all regions and need additional development to reduce first costs to achieve adequate payback.

**Table 20. Life-cycle Assessment of GHP/DOAS with Respect to Baseline VAV Based on Modeled Data**

| | Equipment Premium | First Cost Premium | Annual Savings (PV) | Life-Cycle Cost Savings (PV) | | Simple Payback (years) | SIR |
|---|---|---|---|---|---|---|---|
| **Chicago** | $32,000 | $47,000 | $8,970 | $60,640 | 18% | 5 | 2.3 |
| **Helena** | $32,000 | $47,000 | $10,015 | $73,181 | 22% | 5 | 2.6 |
| **Richmond** | $32,000 | $47,000 | $6,229 | $27,747 | 8% | 8 | 1.6 |
| **Houston** | $32,000 | $47,000 | $4,376 | $5,513 | 2% | 11 | 1.1 |
| **Los Angeles** | $32,000 | $47,000 | $3,710 | ($2,485) | -1% | Not Achieved | 1.0 |

Table 21 presents life-cycle assessment for the CCHP/DOAS with respect to the baseline VAV system. The CHHP has potential to reduce life-cycle costs in all regions, but with longer payback periods in warm/hot climates.

**Table 21. Life-cycle Assessment of CCHP/DOAS with Respect to Baseline VAV Based on Modeled Data**

| | Equipment Premium | First Cost Premium | Annual Savings (PV) | Life-Cycle Cost Savings (PV) | | Simple Payback (years) | SIR |
|---|---|---|---|---|---|---|---|
| **Chicago** | $9,600 | $22,000 | $7,785 | $71,424 | 21% | 3 | 4.3 |
| **Helena** | $9,600 | $22,000 | $8,309 | $77,708 | 23% | 3 | 4.5 |
| **Richmond** | $9,600 | $22,000 | $5,329 | $41,947 | 12% | 5 | 2.9 |
| **Houston** | $9,600 | $22,000 | $4,984 | $37,806 | 11% | 5 | 2.7 |
| **Los Angeles** | $9,600 | $22,000 | $3,887 | $24,638 | 7% | 6 | 2.1 |

Based on modeled simulations, the GHP/DOAS generated life-cycle savings with respect to the CCHP/DOAS only for the Chicago region. This analysis is based on the incremental utility rates provided by NSGL which are significantly lower than commercial market rates. Economic benefits will vary with regional energy prices and demand charges, so a regional assessment should be conducted based on local energy prices to determine if the GHP/DOAS or CCHP/DOAS is cost-effective for a specific location.

Current low energy prices impact the economic payback of higher first cost energy efficient technologies such as the GHP. These results illustrate the current economic barrier for GHP technology due to high first costs, particularly in applications with relatively low utility rates. The GHP is an emerging technology being compared to a more mature technology (CCHP) with multiple manufacturers and decades of optimization and lower first cost. For wider adoption, this indicates that GHP/DOAS may need additional development to optimize performance and reduce first costs.

**Table 22. Life-cycle Assessment of GHP/DOAS with respect to CCHP/DOAS Based on Modeled Data**

| | Equipment Premium | First Cost Premium | Annual Savings (PV) | Life-Cycle Cost Savings (PV) | | Simple Payback (years) | SIR |
|---|---|---|---|---|---|---|---|
| **Chicago** | $22,400 | $25,000 | $1,185 | ($10,784) | -4% | Not Achieved | 0.6 |
| **Helena** | $22,400 | $25,000 | $1,706 | ($4,527) | -2% | Not Achieved | 0.8 |
| **Richmond** | $22,400 | $25,000 | $900 | ($14,200) | -5% | Not Achieved | 0.4 |
| **Houston** | $22,400 | $25,000 | ($607) | ($32,294) | -12% | Not Achieved | N/A |
| **Los Angeles** | $22,400 | $25,000 | ($177) | ($27,124) | -10% | Not Achieved | N/A |

Table 23 presents the life-cycle assessment for a propane-fired GHP/DOAS with higher fuel prices compared to the previous natural gas assessment. Assumptions include no change in GHP first costs or maintenance, as well as no change in performance per the GHP manufacturer. Lifecycle costs for the propane GHP/DOAS were compared to the baseline VAV system, also assumed to operate with propane. While annual cost savings listed in Table 23 were slightly higher than the natural gas results listed in Table 20, LCC savings and simple paybacks were similar. Based on these modeled results for propane heating systems, the GHP/DOAS would be a cost-effective option, relative to the baseline VAV system, only in heating dominated climates.

**Table 23. Life-cycle Assessment of Propane GHP/DOAS with Respect to Baseline VAV Based on Modeled Data**

| | Equipment Premium | First Cost Premium | Annual Savings (PV) | Life-Cycle Cost Savings (PV) | | Simple Payback (years) | SIR |
|---|---|---|---|---|---|---|---|
| **Chicago** | $32,000 | $47,000 | $9,393 | $65,000 | 19% | 5 | 2.0 |
| **Helena** | $32,000 | $47,000 | $10,079 | $73,000 | 21% | 5 | 2.6 |
| **Richmond** | $32,000 | $47,000 | $6,334 | $29,000 | 10% | 7 | 1.5 |
| **Houston** | $32,000 | $47,000 | $4,180 | $3,000 | 1% | 11 | 1.0 |
| **Los Angeles** | $32,000 | $47,000 | $3,705 | (-$3,000) | -1% | Not Achieved | 1.0 |

# 9.0 IMPLEMENTATION ISSUES / LESSONS LEARNED

## 9.1 VRF REGULATIONS FOR DOD FACILITIES

Several regulations summarized in the *Unified Facilities Criteria UFC 3-410-01 Heating, Ventilating, and Air Conditioning Systems* apply to the use of VRF systems for DoD facilities. Previous versions of UFC 3-410-01 (e.g. Change 3, January 25, 2017) limited or discouraged the use of VRF in various DoD facilities. The most recent version (Change 4, November 01, 2017) presents several requirements for VRF systems per Section 3-5.16 and *Best Practices for Variable Refrigerant Flow (VRF) Systems* in Appendix B-11. These concerns are addressed in detail in Appendix D.

The first requirement is to meet Open Control System Requirements per United Facilities Guide Specifications (UFGS) 23 09 00 and either UFGS-23 09 23.01 or UFGS-23 09 23.02. Other military codes that may apply including: 10 U.S. Code § 2867. *Energy monitoring and utility control system specification for military construction and military family housing activities*; UFGS 23 09 23.01 *LonWorks Digital Control for HVAC and Other Building Control Systems*; UFGS 23 09 23.02 *BACnet Digital Control for HVAC and Other Building Control Systems*.

This demonstration plan was reviewed by NSGL Asset Management group and was issued a Site Approval Review Checklist (SA17-0004). The following HVAC requirements outlined in UFC 4-010-01 were identified as applicable for this demonstration (Figure 43):

- DoD Standard 16 (1) *Air intakes requirements to heating, ventilating, and air conditioning (HVAC) systems*.

  (a) HVAC Replacements and Upgrades. Where air handling equipment in heating, ventilating, and air conditioning systems is being replaced or when they are being upgraded, need to ensure that air intakes are either 10 feet / (3 meters) above the ground or extend the intakes to that height.

  <u>Status</u>: DOAS air intake was extended to 10 feet above ground.

- DoD Standard 18 (1) *Emergency Air Distribution Shutoff for HVAC systems*.

  (a) HVAC systems being installed for new and existing buildings are required to have Emergency Air Distribution Shutoff for HVAC systems IOT immediately shut down the air distribution and exhaust systems throughout the building and that'll close all dampers leading to the outside except where interior pressure and airflow control would more efficiently prevent the spread of airborne contaminants and/or ensure the safety of egress pathways.

  (b) HVAC shutoff switches must be located to be easily accessible by building occupants so that the travel distance to the nearest shutoff switch will not be in excess of 200 feet (61 meters). They must also be well labeled, and of a different color than fire alarm pull stations.

  <u>Status</u>: Emergency shutoff switches was installed to shutoff air intake at the DOAS.

Other pertinent regulations include ASHRAE Standards 15 and 34 which provide safety guidance for refrigerant systems. Since VRF systems often do not supply ventilation, ASHRAE Standard 62 can provide guidance on the required ventilation provided by auxiliary equipment such as a DOAS.

- ANSI/ASHRAE Standard 15-2019 (packaged w/ Standard 34-2019) *Safety Standard for Refrigeration Systems and Designation and Safety Classification of Refrigerants*.
- ANSI/ASHRAE Standard 62.1-2016 *Ventilation for Acceptable Indoor Air Quality*.

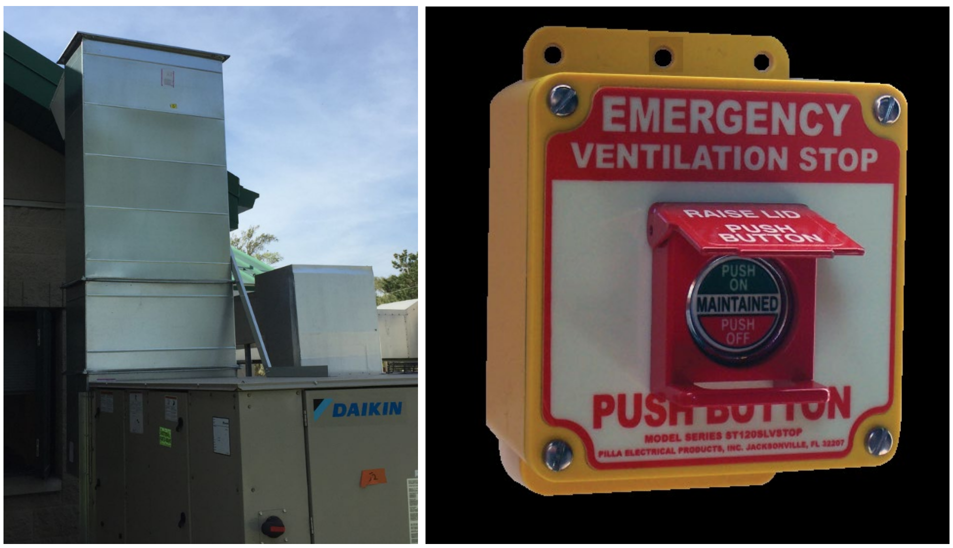


**Figure 43. DOAS Installation Complied with DoD Standards 16 and 18**

## 9.2 LESSONS LEARNED

Baseline monitoring highlighted some operational issues with the existing baseline VAV system. As confirmed by the manufacturer, a BAS is required to integrate the controls for the gas burner and VAV-boxes. Without a BAS at the field site, the VAV gas burner and the indoor VAV-boxes operated independently resulting in excess electric resistance heating with high peak electric demand along with higher energy costs. Although the baseline system did not operate as designed, this may be a typical situation for smaller buildings or sites without a central BAS. In addition, due to comfort issues, several occupants at the field site relied on electric resistance space heaters to provide supplemental heating. Since VAV systems are widely used for multi-zone applications, such as office buildings, retro-commissioning and/or retrofitting integrated controls may improve efficiency and reduce energy costs for existing equipment.

Based on this demonstration, VRF systems improved comfort along with significant energy savings, lower peak electric demand, and lower annual costs compared to conventional VAV systems. Both CCHP and GHP systems have potential to reduce life-cycle costs depending on regional heating/cooling loads and energy prices. The demonstration also identified operational issues and limitations for both VRF technologies in this application. During the demonstration, the field site experienced a number of outages of both VRF equipment and conventional HVAC equipment (e.g. the DOAS igniter). Several outages were due to equipment installation issues, highlighting the importance of well-trained service providers which is a common concern for emerging technologies.

For this demonstration in climate ASHRAE Climate Zone 5, the electric CCHP was unable to meet the heating load for several days. During the monitoring period (February 2018 and 2019), ambient temperatures dropped below winter design conditions. During extreme cold, the CCHP continued to operation but was unable to maintain zone temperatures and operated at very low efficiencies. This CCHP model was specifically designed for cold climate applications without supplemental heating. For this demonstration, the CCHP was over sized based on de-rated heating capacity at low temperatures to match the estimated peak heating load. As weather conditions become less predictable with more extreme temperatures, the CCHP will require a means of supplemental heating for reliable cold climate operation.

As a key finding, this demonstration highlights how VRF heat pumps regularly operate at very low part-loads, even when sized appropriately. This is amplified when paired with a DOAS which can reduce the building heating and cooling loads. Currently, GHP manufacturer rating is based on full-load capacity and efficiency at select rating conditions; however, depending on the climate, the heat pump might never operate at those specific conditions. By design, VRF systems usually operate at lower part-loads by modulating to meet the multi-zone heating and cooling loads. Existing GHP performance ratings are not a good indicator of seasonal performance. Current efforts to update GHP standards will include development of VRF performance metrics that reflect actual install conditions including part-load operation. The parallel evolution of GHP performance standards will support the development of more optimized designs.

Part-load operation adversely impacted the performance of both heat pumps; however, this specific GHP model had much lower than expected performance at low part-load operation. This extent of decreased part-load performance is not inherent in this class of technology and could be due to product-specific controls or engine sizing. GHPs use variable-speed engines to enable them to closely follow the load and maintain efficiency. This is an important finding and warrants further investigation to optimize part-load performance.

This demonstration compares an emerging technology (GHP) with a more mature technology (CCHP) that has multiple manufacturers, decades of design optimization, and lower first cost. These results suggest the GHP may need additional research and development to optimize performance and to reduce equipment and installation costs in order to improve regional economics and support broader market adoption.

## 9.3 SUMMARY OF KEY FINDINGS AND RECOMMENDATIONS

- Both CCHP and GHP VRF systems improved comfort compared to the baseline VAV system along with significant energy savings, lower peak electric demand, and lower energy costs.
- Both CCHP and GHP VRF systems had lower life-cycle costs compared to the baseline VAV system for this field site. Life-cycle cost savings for other locations are dependent on regional heating/cooling loads and energy prices.
- The electric CCHP was unable to meet the facility heating load for several days without supplemental heating, despite oversizing based on the estimated peak heating load.
- VRF heat pumps regularly operate at very low part-loads, below 40% specified capacity, even when sized appropriately. To improve overall efficiency, VRF systems should be optimized for low part-load operation. Alternatively, if combined with a DOAS able to provide supplemental heating and cooling, VRF design capacity could be reduced leading to lower first costs and improved efficiency by operating closer to full load.
- While low part-load operation adversely impacted the performance of both heat pumps; the GHP model used in this demonstration had lower than expected performance under these conditions. This specific GHP design may need additional development to optimize performance and reduce equipment costs in order to improve regional economics and broader market adoption.
- Without a central building automation system (BAS), the baseline VAV system used excess electric resistance heating resulting in a higher than expected winter peak electric demand and energy costs. Despite high energy use, the baseline system had difficulty maintaining comfort in the facility. The addition of integrated controls for existing VAV systems may reduce peak electric demand and lower energy costs.
- Several regulations in *Unified Facilities Criteria (UFC)* apply to the use of VRF systems for DoD facilities. This demonstration provides measured energy savings and a detailed life-cycle cost assessment for VRF systems which validates previous modeled estimates. In light of the significant energy savings potential and comfort improvements, we recommend the UFC review current regulations for VRF systems.

# 10.0 REFERENCES


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# APPENDIX A POINTS OF CONTACT

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|---|---|---|---|
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| Jacquelynn Kelly | NAVFAC Mid-Atlantic<br>Capital Improvements<br>310 Seabee Way,<br>Building 1H<br>Great Lakes, IL 60088 | 847/688-5395, ext. 258<br>DSN 792-5395, ext. 258<br>jacquelynn.kelly@navy.mil | Financial POC accepting MIPR |
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# APPENDIX B M&V SPECIFICATIONS

**Table B-1. Baseline DAS Equipment Specifications**

| Monitored Data Point | | Equipment | Manufacturer/ Model | Accuracy |
|---|---|---|---|---|
| GP | Gas Use | Gas Meter with pulser | American Meter<br><br>AC-425-TC-P | 20 pulses per cubic feet |
| W1 | VAV total electricity use | Watt meter with split-core current transformers | Continental Controls<br><br>RWNB-3D-480-P<br><br>ACT-0750-### Opt C0.6 | ± 0.5% |
| W2 | VAV supply fan power | | | |
| W3 | VAV exhaust fan power | | | |
| W4 | VAV reheat boxes VAV-1 through VAV-19 | | | |
| T1, RH1 | Outdoor Air Temperature | humidity/temp sensor | Dwyer WHT-311 | ± 3% |
| T2 | Supply Air Temperature | Thermocouple | Omega<br>5TC-TT-T-24-72 | ± 1ºF |
| T3 | Return Air Temperature | | | |
| T4 | Exhaust Temperature | | | |
| CS-EC | Economizer on/off | Current Switch | Setra CSCGFN015NN | N/A |
| Data Logger | Records Data | Intellilogger | Logic Beach IL-80 | |
| Cell Modem | Connects Data Logger to Internet | Cell Modem | Sierra Wireless Raven XE | |

### Table B-2. Instrumentation for Monitoring Performance of the Demonstration Equipment

| Instrumentation | | Accuracy | Units | Signal Type |
|---|---|---|---|---|
| Watt meter with split-core current transformers | Measure electricity use:<br>- GHP and CCHP outdoor units<br>- VRF fan coils<br>- DOAS | ± 0.5% | kWh | Analog |
| Current Switch | DOAS economizer operation | | | Digital |
| Gas meter with pulse output | Measure gas consumption for GHP and DOAS<br>(temperature-compensated volume flow) | 20 pulses per cubic feet | Cubic foot | Digital |
| Pressure transducer | Measure:<br>- Supply gas pressure (GHP)<br>- Refrigerant R410A vapor and liquid pressure (GHP, CCHP)<br>- Barometric pressure | | Psig | Analog |
| Type T thermocouples | Measure dry-bulb temperature:<br>- Outside air<br>- DOAS supply<br>- Refrigerant R410A vapor and liquid temperature (GHP, CCHP)<br>- Sample fan coil return/supply | ± 1°F | ºF | Analog |
| Humidity sensors | Measure relative humidity of ambient, supply air and return air at each air handler | | % RH | Analog |
| Coriolis meter | Measure refrigerant R410A liquid mass flow rate for GHP and CCHP | ± 0.10% | kg/hr | Analog |

# APPENDIX C MODELING APPROACH AND ASSUMPTION

The simulations in this study were performed using OpenStudio 2.6[1] and EnergyPlus 8.8[2]. Both are open source tools developed by the US Department of Energy (DOE) for the purpose of estimating and optimizing the energy consumption of buildings on an annual basis. A 3-dimensional representation of the building was created with Trimble Sketchup[3], using Google Earth satellite image of the building for the footprint and external shading. The building model as simulated is shown in Figure C-1. Construction details for the building were not available except for the vintage. The building was assumed to be constructed similar to the DOE Commercial Reference Buildings of the 1980+ vintage[4]. Building internal loads, schedules, furnishings, lighting, and occupancy were assumed to be DOE Reference Medium Office type. Where more accurate information was available, e.g., Outside Air (OA) and setpoint setback, custom schedules were created. Annual simulations of the building energy consumption were performed for the Typical Meteorological Year 3 [5] weather data in each location.

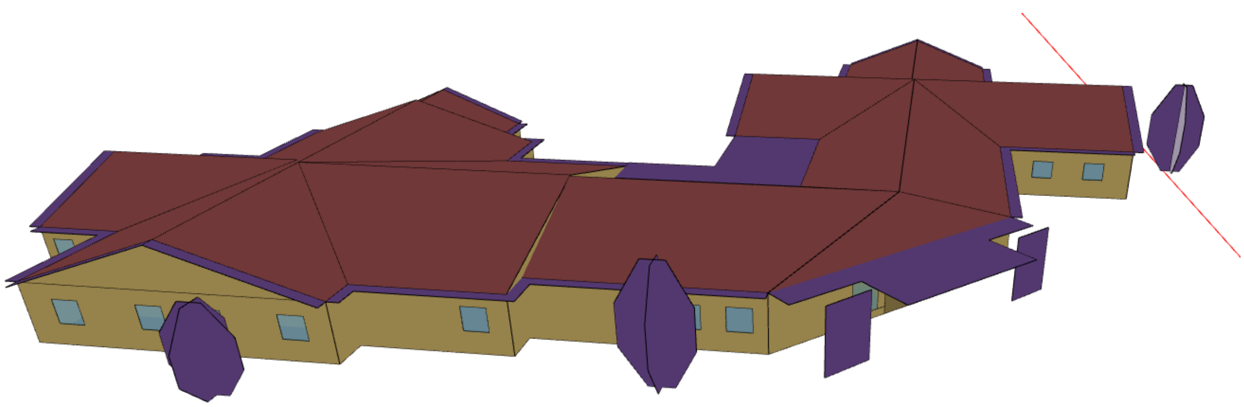


**Figure C-1 Three-dimensional rendering of the building as simulated. Purple geometry represents shading surfaces.**

The internal layout of space types and thermal zones was developed from the mechanical layout of the building. Figure C-2 illustrates the level of granularity used for space types and thermal zones (different colors represent unique space types and thermal zones). The spaces in the model that were outside the conditioned space in study were setup with "ideal loads" systems, i.e., the model maintained setpoints without consideration for the type of HVAC system used.

1 Guglielmetti, R., Macumber, D. and Long, N., 2011. OpenStudio: an open source integrated analysis platform (No. NREL/CP-5500-51836). National Renewable Energy Lab.(NREL), Golden, CO (United States).
2 Crawley, D.B., Lawrie, L.K., Winkelmann, F.C., Buhl, W.F., Huang, Y.J., Pedersen, C.O., Strand, R.K., Liesen, R.J., Fisher, D.E., Witte, M.J. and Glazer, J., 2001. EnergyPlus: creating a new-generation building energy simulation program. Energy and buildings, 33(4), pp.319-331.
3 SketchUp: https://www.sketchup.com/
4 Existing Commercial Reference Buildings Constructed In or After 1980 - https://www.energy.gov/eere/buildings/existing-commercial-reference-buildings-constructed-or-after-1980
5 National Solar Radiation Data Base, 1991- 2005 Update: Typical Meteorological Year 3 - https://rredc.nrel.gov/solar/old_data/nsrdb/1991-2005/tmy3/

For the spaces included in the study, ~7600 sq-ft, the HVAC systems modeled included the VAV baseline, CCHP/DOAS, and GHP/DOAS.

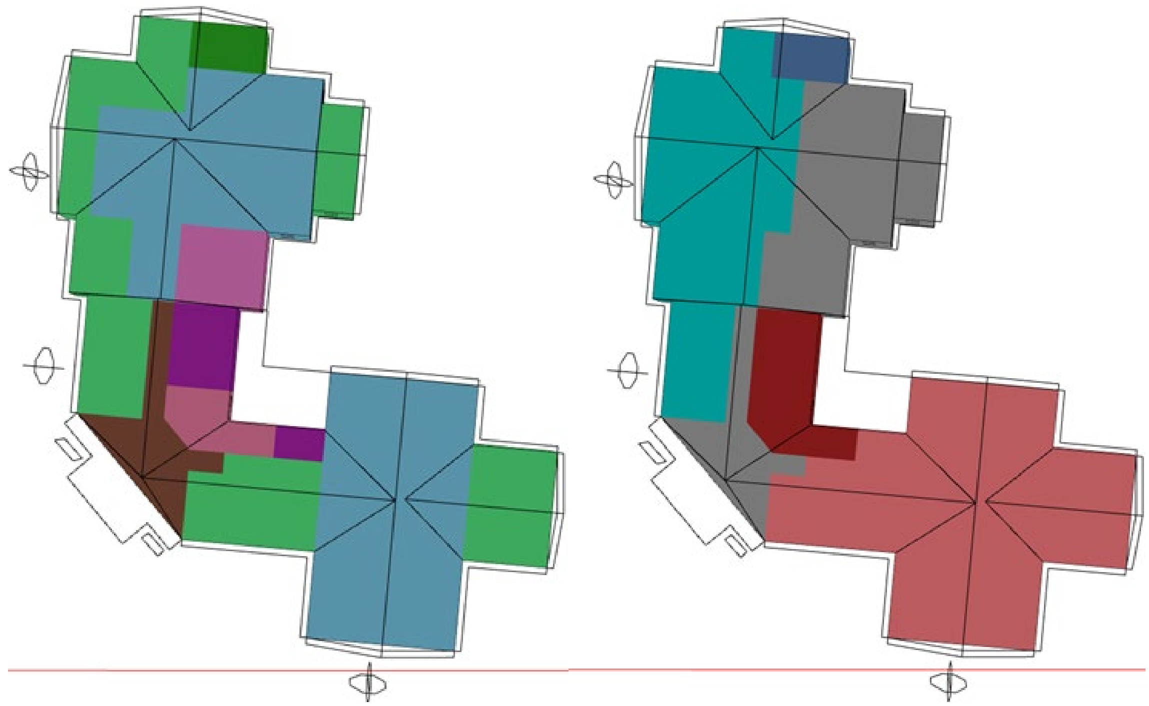


**Figure C-2. Granularity Level in Space Types (Left) and Thermal Zones (Right) Used in the Model.**

*Space types included open and closed offices, conference rooms, a mechanical room, break rooms, restrooms, and a lobby.*

To validate the building model, both the baseline case and the CCHP/DOAS, and GHP/DOAS were simulated as installed. Design loads predicted by the model were similar to the design loads calculated by the mechanical contractor using Carrier HAP software. The model used in this study tended towards higher heating loads and lower cooling loads, e.g., ~100 MBH design heating vs ~85 MBH, and ~4 tons for cooling vs 6 tons from Carrier HAP, per zone. A limitation of the model and tools used in this study is that duct losses (heat loss and leakage) cannot be captured in a straightforward manner. These losses were accounted for in a simplified manner by applying a 20% efficiency penalty to the heating and cooling coils of the VAV and DOAS systems since both used the same ducting.

The final building model underestimated gas consumption for the VAV baseline case, even with the duct loss penalty. The exact operating characteristics of the VAV system were unknown, e.g., setpoints, reset schedule (if any), and volume of outdoor air. The baseline simulations were based on the name-plate specs for the VAV. However, they may not be representative of how the real system operated. The predicted energy consumption was very sensitive to the amount of outdoor air treated. In contrast, detailed info was available for the VRF/DOAS case. When the VRF/DOAS system was simulated as installed, the model provided good agreement with respect to gas and electricity consumption, plotted in Figure C-3.

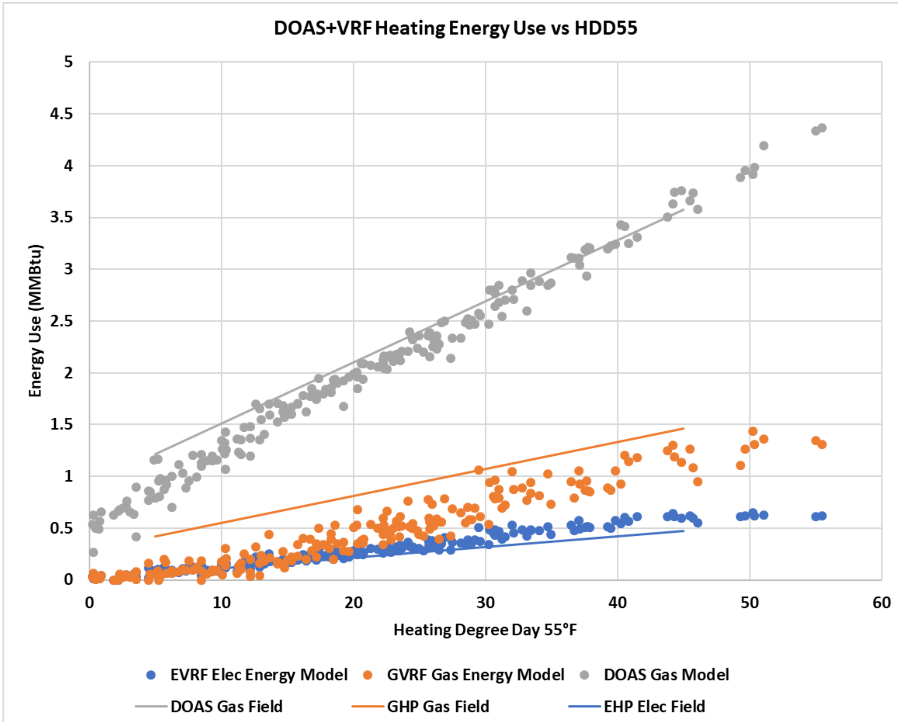



**Figure C-3. Predicted vs Actual Energy Consumption for the VRF/DOAS Case (as installed).**

No further modifications were made to the baseline model. For the economic analysis, the VAV baseline case represents a more optimized VAV system. It used recommended setpoints and control strategy (55-68°F supply air temperature setpoint based on the cooling needs of the warmest zone). For the final economic analysis in different climates, the model was simulated as having only a single system, VAV, CCHP/DOAS, or GHP/DOAS. The amount of outdoor air was identical in each case, and the capacities of each system were auto-sized by EnergyPlus for each climate.

For the VAV baseline and DOAS gas heating, in addition to the duct loss efficiency penalty, a part-load efficiency curve was defined as Eff_gas = Eff_(steady state) * (0.8 + 0.2*PLR) where PLR is the part-load ratio, defined as PLR= load/capacity. The cooling system (DX cooling coil) in the VAV and DOAS systems used the default performance curves that are available in OpenStudio. Performance for the Daikin heat pump (CCHP) was modeled using the "performance curve" based method available in EnergyPlus. Manufacturer data was used to derive performance curves for the heat pump capacity and energy input ratio as a function of indoor and outdoor temperatures, as well as part-load ratio. The curves were derived using the method developed by Raustad[6]. Performance data from Daikin was only available down to a part-load ratio of 0.5. The curves developed were used to extrapolate the performance of the heat pump to lower part-load ratios. Comparison with available field measurements provided reasonable agreement at low part-loads. However, the form of the model used in EnergyPlus is limited and exact reproduction of the performance as specified Daikin was not possible.

6 Raustad, R., 2012. Creating performance curves for variable refrigerant flow heat pumps in EnergyPlus. Florida Solar Energy Center. FSEC-CR-1910-12

Figure C-4, provides a comparison of the predicted model Heating COP vs outdoor wet-bulb temperature. CCHP performance during defrost and any loss in capacity was not described by the manufacturer. To account for it, a 10% energy input penalty was applied to the CCHP performance at outdoor temperatures between 20-40°F.

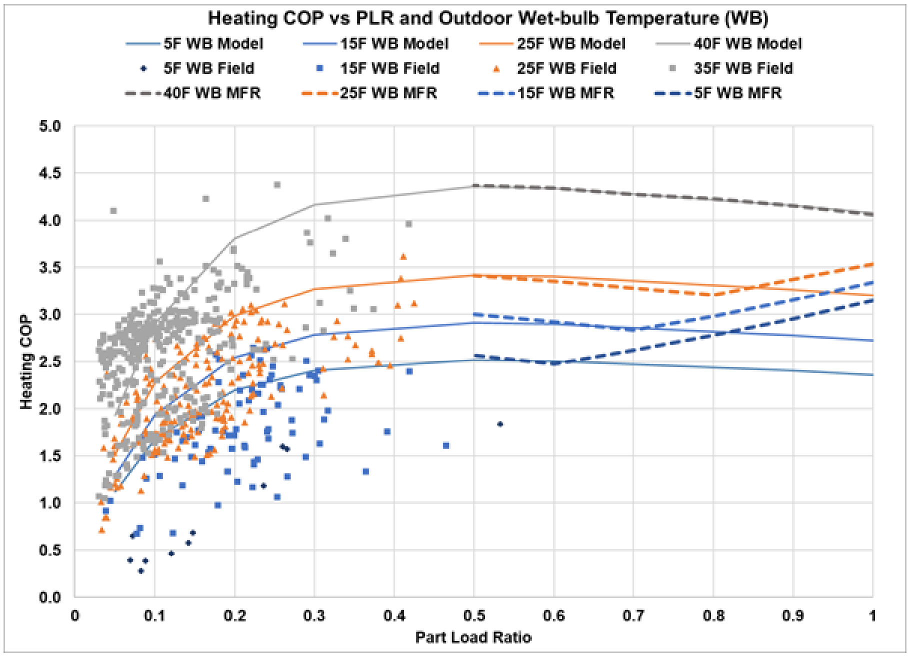



**Figure C-4 Model predicted Heating COP compared to manufacturer specs and field measurements (unfiltered)**

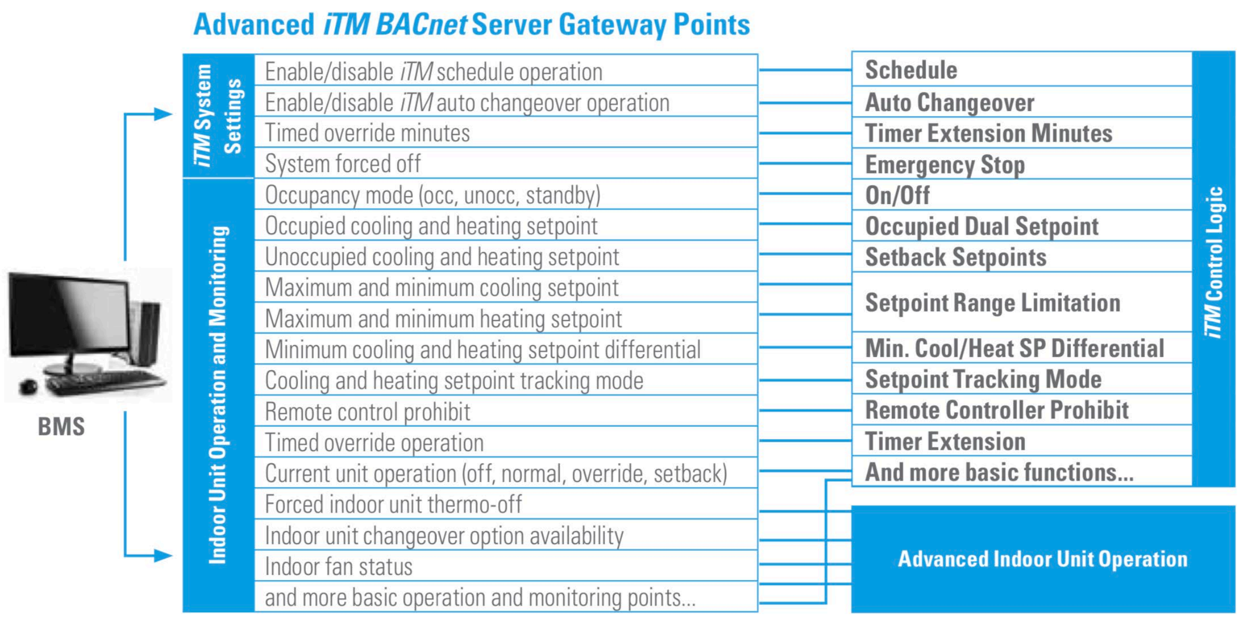
For the GHP, a similar procedure was used to develop capacity and energy input ratio performance curves. Performance data from a different GHP manufacturer was used to develop the curves since the demonstration GHP performed below expectations. The alternative GHP is was monitored by GTI as part of a separate study and demonstrated better performance at part-load. This data shared with GTI is proprietary and cannot be reproduced here for comparison.

# APPENDIX D UFC/UFGS ISSUES ON VRF

During this project, after the installation of the VRF systems at the demonstration site, the DoD Unified Facilities Criteria issued a revision to yjr UFC 3-410-01 Heating, Ventilating, and Air Conditioning Systems presenting additional design considerations regarding the use of VRF in DoD facilities. The section below summarizes and addresses each of these considerations.[7]

1) *Unified Facility Guide Specifications require the installation of non-proprietary control networks down to the level of each individual device in the system. As of the publication date of this UFC, all known commercially-available VRF systems rely on a proprietary network that does not comply with the UFGS requirements.*

    i. Daikin offers a BACnet/Lon interface option which can communicate with non-proprietary building automation systems (BAS). The cost of the BACnet/Lon interface depends on the number of points and system size but is estimated at $3,000 and $6,000. The BAS interface is hard-wired access only, so security would be handled on the BAS side. At the local level the system can be password protected.




2) *The practical implications of a proprietary control network are that only factory-authorized technicians are allowed to install and service VRF systems, cutting down on the ability of a building owner to shop around for better prices, and leading to higher life-cycle operating costs.*

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[7] Nelson, L. 2018. *DoD Issues Guidance on VRF Systems*. International Association of Plumbing and Mechanical Officials. IAPMOnline http://www.iapmonline.org/Documents/archive/20180604_DoD_VRF.aspx

i. DoD facility staff can be trained to service VRF equipment transferring this expertise in house for faster response and lower life-cycle operating costs. As part of the Technology Transfer task for this project, NAVFAC facility staff were provided two days training by GHP manufacturer to service the VRF systems and routine engine maintenance. Feedback following the training indicated NAVFAC staff felt confident they could transfer their expertise from conventional air conditioning and vehicle engine maintenance to providing technical support for the VRF systems. As another training alternative, Daikin factory authorization for VRF equipment consists of one to two-day training for install and commissioning with an additional two-day training for advance service.

3) *VRF systems piping/tubing must have all brazed connections…list of fittings and joints that are prohibited include but are not limited to the following: push-on fittings, extruded fittings, flare fittings, press-connect fittings, mechanical joints and groove joints.*

i. Brazed fittings are commonly used for refrigeration piping. Likewise, for VRF systems, brazed fittings create very reliable leak-free joints to withstands high and low temperatures and pressures and tend to be lower in cost. During this demonstration, threaded fittings were required to install flow meters in the refrigerant lines. While threaded fitting were used successfully at other field sites, this site experienced multiple refrigerant leaks due in part to the contractor's lack of experience using threaded fittings instead of brazed. This supports the UFC requirement for the use of brazed fittings.

4) *A recent EPA rule (40 CFR § 82.157, 11/18/16) extended refrigerant management requirements to common substitutes like hydrofluorocarbons (HFCs) refrigerant systems if 50 lbs or greater of refrigerant to take effect Jan 2019.*

i. R-410A (which contains only fluorine) does not contribute to ozone depletion, and is therefore becoming more widely used, as ozone-depleting refrigerants like R-22 are phased out. Although R410a has a higher GWP than R-22, it allows for higher SEER ratings reducing overall power consumption. https://en.m.wikipedia.org/wiki/R-410A

ii. EPA issued a proposed rule Protection of Stratospheric Ozone: Revisions to the Refrigerant Management Program's Extension to Substitutes on 10/01/2018. This action proposes to revisit the Agency's recent approach to regulating appliances containing substitute refrigerants such as HFCs and rescind the November 18, 2016 extension of the leak repair provisions to appliances using substitute refrigerants. This proposal also requests public comment on rescinding other provisions that were extended to substitute refrigerants. This proposal would not affect the requirements for ozone-depleting refrigerants. If finalized as proposed, this action would rescind the leak repair and maintenance requirements at 40 CFR 82.157 for substitute refrigerants, and therefore, appliances with 50 or more pounds of substitute refrigerants would not be subject to the following requirements. https://www.epa.gov/section608/revised-section-608-refrigerant-management-regulations

iii. If the EPA does not rescind the November 18, 2016 rule, it will be an added burden for reporting purposes. At NSGL, the total R410a refrigerant charge for the 8-ton Nextaire GHP is 36 lbs. 8.4 oz and for the 12-ton Daikin IV is 35.7 lbs. While each heat pump VRF system in this demonstration is under the 50 lbs. limit, typically a single VRF system would be sized for this building which would exceed the 50 lbs. refrigerant limit.

5) *VRF refrigerant is heavier than air, and "puddles" on the floor of a room, displacing breathable air*

   i. Safety issues regarding refrigerant are successfully addressed by ASHREA Standards 15 and 34. https://www.daikinac.com/content/assets/DOC/White-papers-/TAVRVUSE13-05C-ASHRAE-Standard-15-Article-May-2013.pdf

6) *VRF system is designed around the thermodynamic properties of a specific refrigerant type, this means that when a refrigerant type is phased out in favor of a more environmentally friendly formulation, as is happening today, the price of the refrigerant increases rapidly, and the VRF system itself will likely need extensive modification or even replacement in order to function properly*

   i. OEMs are investigating refrigerants with lower global-warming-potential (GWP) with a focus on maintaining safety and performance along with minimal changes in equipment; however, this may impact equipment design. R-410a is used in air conditioning as well as VRF systems, so all cooling equipment will be impacted by refrigerant changes. R-410a is a blend of multiple refrigerants, and some are considering components of the blend, such as R32 which has a lower GWP (approx. 700) compared to R410a GWP of 1980. Most manufacturers have placed plans on hold regarding the switch to a new refrigerant or blend due to the change in regulations by the current administration.

   ii. Daikin is evaluating R32 as well as other refrigerants including R466A and DR-55. Daikin VRF fan coil units are used for Daikin heat pumps as well as Yanmar and Aisin GHPs. https://www.daikin.com/corporate/why_daikin/benefits/r-32/

   iii. Honeywell recently unveiled Solstice® N41 (provisional R-466A), a nonflammable and lower GWP refrigerant for use in stationary air conditioning systems. In addition, early testing indicates that switching to Solstice N41 would require <u>minimal changes to equipment and no additional training for installation and repair technicians</u>. Preliminary data indicates that the refrigerant may allow OEMs to easily convert from R-410a https://www.achrnews.com/articles/137328-honeywell-announces-nonflammable-refrigerant-that-can-replace-r-410a

   iv. Carrier reports that R466a could be a viable R410a alternative and a potential rival for R32 refrigerant. R466a has a lower GWP (733) vs R401a 1980 GWP. R32 has a similar GWP, but R32 is mildly flammable. R466a has the added benefit of being nonflammable. R466a is expected to be available commercially sometime in 2019. Reference: Carrier Enterprise News. November 27, 2018. Is There A Nonflammable Refrigerant That Can Replace R410a? http://news.carrierenterprise.com/r410a-nonflammable-refrigerant-replacement/

7) *Life-cycle Cost analysis comparing VRF with traditional systems can be difficult given the relative newness of VRF systems and given that many VRF systems can only be serviced by factory-trained technicians which affects the maintenance costs for the system compared to more traditional systems.*

   i. The complexity and customized design of VRF systems for specific buildings make it difficult to predict energy savings relative to baseline HVAC systems. Due to limited field data available for VRF systems, energy savings are often based on energy modeling or laboratory data obtained under controlled conditions. This demonstration provides a unique opportunity to directly compare measured performance data for a VRF system to conventional equipment. Combined with existing EnergyPlus models, adapted for cold climate performance, this data and cost assessment provides life-cycle cost assessment for VRF systems which validate previous modeled estimates.

   ii. The growing market for VRF in the U.S. is expanding the regional service networks for qualified technicians and contractors. Daikin Service has hundreds of service technicians throughout the U.S., Mexico, and Canada. In addition, several VRF systems including the two GHP manufacturers (Yanmar and Blue Mountain Energy, formerly IntelliChoice Energy) use Daikin controllers and fan coils. Other manufacturers such as Panasonic and Mitsubishi make their own fan coil designs.

   iii. As an alternative, DoD facility staff can attend brief training sessions to provide VRF maintenance and service. (See Section 1(ii))

## APPENDIX E DOD INSTALLATIONS

Based on a search of public information, approximately 193 DoD installations are located throughout the U.S. Of these installations, about twelve do not have natural gas utility service. The majority of installations, about 70%, are located in warm or hot climates and 30% are in located in cold climates. About 55 DoD installations are locate in cold climates with natural gas service.

**Table E-1. DoD Installations per State**

| | | | |
|---|---|---|---|
| Alabama | 3 | Montana | 1 |
| Alaska | 5 | Nebraska | 1 |
| Arizona | 5 | Nevada | 3 |
| Arkansas | 1 | New Jersey | 1 |
| California | 24 | New Mexico | 3 |
| Colorado | 4 | New York | 4 |
| Connecticut | 1 | North Carolina | 8 |
| Delaware | 1 | North Dakota | 2 |
| District of Columbia | 3 | Ohio | 1 |
| Florida | 12 | Oklahoma | 4 |
| Georgia | 7 | Pennsylvania | 1 |
| Hawaii | 8 | Rhode Island | 1 |
| Idaho | 1 | South Carolina | 9 |
| Illinois | 2 | South Dakota | 1 |
| Indiana | 1 | State | 1 |
| Kansas | 3 | Tennessee | 1 |
| Kentucky | 2 | Texas | 14 |
| Louisiana | 3 | Utah | 2 |
| Maryland | 11 | Virginia | 17 |
| Massachusetts | 2 | Washington | 11 |
| Mississippi | 4 | Wisconsin | 1 |
| Missouri | 2 | Wyoming | 1 |