* Approved for public release; distribution is unlimited

ORNL/TM-2016/365

# Evaluation of Variable Refrigerant Flow Systems Performance on Oak Ridge National Laboratory's Flexible Research Platform: *Part 2 Heating Season Analysis*


Piljae Im, PhD
Mini Malhotra, PhD
Jeffrey D. Munk

**August 2016**

OAK RIDGE NATIONAL LABORATORY
MANAGED BY UT-BATTELLE FOR THE US DEPARTMENT OF ENERGY

**ORNL/TM-2016/365**

Energy and Transportation Science Division

# Evaluation of Variable Refrigerant Flow Systems Performance on Oak Ridge National Laboratory's Flexible Research Platform: Part 2 Heating Season Analysis

Piljae Im, PhD
Mini Malhotra, PhD
Jeffrey D. Munk

Date Published: August 2016

Prepared by
OAK RIDGE NATIONAL LABORATORY
Oak Ridge, TN 37831-6283
managed by
UT-BATTELLE, LLC
for the
US DEPARTMENT OF ENERGY
under contract DE-AC05-00OR22725

# CONTENTS

## LIST OF FIGURES



## LIST OF TABLES

## EXECUTIVE SUMMARY

This report provides second-year project test results for the multi-year project titled "Evaluation of Variable Refrigeration Flow (VRF) system on Oak Ridge National Laboratory (ORNL)'s Flexible Research Platform (FRP)." The purpose of the second-year project was

- To evaluate the full- and part-load performance of a VRF system compared with that of the existing baseline heating, ventilation, and air-conditioning (HVAC) system, which is a conventional rooftop variable-air-volume (VAV) system with electric resistance heating
- To evaluate the energy savings potential of using VRF systems in major US cities using hourly building energy simulation

The second-year project performance period was from July 2015 through June 2016.

The performance of two HVAC systems was evaluated using ORNL's FRP, which is a two-story, 3,200 $ft^2$ (297.3 $m^2$) multi-zone unoccupied building that represents a typical low-rise, small office building common in the US existing building stock. The FRP is equipped with a conventional 12.5 ton (44 kW) RTU-VAV reheat system as the baseline system. For this study, a 12 ton (42 kW) VRF with a dedicated outdoor air system (DOAS) was installed to be compared with the baseline RTU system.

During the test period, full- and part-load conditions (i.e., 100, 75, and 50% loads) in the building were maintained alternately by conditioning either the entire building or selected zones, and emulating the occupancy accordingly. During the study period, each system was operated alternately under each of the three load conditions for 2–3 days; and the system parameters, indoor and outdoor conditions, loads, and energy use were monitored. The heating season performance and energy use of both systems was monitored in the winter of 2016. The system performance was evaluated in terms of weather-normalized HVAC energy consumption, the ability to maintain the desired indoor temperatures in the conditioned zones, and the seasonal average coefficient of performance (COP). Furthermore, the energy savings potential of using VRF systems in major US cities was evaluated using hourly building energy simulations calibrated with data from the measured data in the building. The following are the key findings and lessons learned from this case study. Three separate reports for cooling season analysis, heating season analysis, and simulation analysis are provided based on the analysis results. This report presents the heating season analysis results.

### Heating Season Analysis

The heating season analysis is based on the measured data from December 30, 2015, through March 6, 2016. Like the cooling season analysis, the analysis shows that the VRF system used less energy than the baseline RTU system for full and part load conditions. The energy savings for the VRF system compared with those for the RTU system for the cooling season are estimated to be 51, 47, and 27% under the 100, 75, and 50% load conditions, respectively.

In general, the RTU system provided better thermal control for most rooms during heating season because it can provide simultaneous cooling and heating for different rooms. Data for the VRF system show some rooms were overheated. As the installed VRF system cannot provide simultaneous cooling and heating, most of the rooms were overheated during periods when the room needed cooling.

During the heating season, the COPs of the VRF system ranged from 1.2 to 2.0, which were lower than expected but still substantially higher than the COPS of the RTU system. The RTU system COPs were less than 1, which was expected because the system’s two sources of heating are electrical resistance heat in the VAV boxes, with a COP of 1, and natural gas heating, with a COP of 0.8.

**Lessons Learned**

During the heating season test, it was found that the DOAS was turned off when the outdoor temperature was below 5°C and the system was operating in ventilation mode, or when the outdoor temperature dropped below −5°C and the system was operating in heating mode. The current DOAS cannot provide conditioned air to the space below a −5°C outside air temperature, and this is a potential issue for cold climates in the United States. It might be worthwhile to consider coupling the HVAC system with energy recovery ventilation/heat recovery ventilation so that unconditioned cold air can be preconditioned before entering the DOAS.

Although the discharge set point temperature for the DOAS in heating mode was 21°C, which was the same temperature as the room thermostat set point, the measured DOAS discharge temperature went up to 42°C. Since the DOAS provided warmer air than originally planned, sometimes rooms were overheated. Currently, the DOAS unit is not controlled based on room conditions but only based on heating/cooling mode and outdoor air temperature. Hence, more integrated control for the DOAS should be considered to respond to the room conditions. In addition, in heating mode, the DOAS should be able to provide air to the rooms at a neutral temperature (i.e., room set point temperature).

The target building, the FRP, is a small office building with a high percentage of glazing on all four wall orientations and high internal heat gains from emulated occupancy. With this setup, there were many times when the south-, east- and west-facing rooms needed cooling, even during heating season, while the other rooms still needed heating. As the currently installed VRF system is a heat pump–type system, it cannot provide simultaneous heating and cooling; and we observed that some room temperatures went over 30°C. Therefore, in this type of building (i.e., with many windows and high internal heat gain), a heat recovery type of VRF system is strongly recommended.

# 1. INTRODUCTION

## 1.1 BACKGROUND

This report provides test results from the second year of the multi-year project titled "Evaluation of Variable Refrigeration Flow system on Oak Ridge National Laboratory's Flexible Research Platform." The research for the first year of the 3 year project was performed from July 2014 through April 2015. The purposes of the first year were

- To design and install a variable refrigerant flow (VRF) system in the new Flexible Research Platform (FRP) test facility at Oak Ridge National Laboratory (ORNL)

- To evaluate its energy savings potential by comparing the energy use of the VRF system with that of the existing baseline heating, ventilation, and air conditioning (HVAC) system, a conventional rooftop unit (RTU) with a variable-air-volume (VAV) system and electric resistance heating

In addition, a newly developed enhanced control algorithm called CCM (the comfort control method) was implemented to estimate additional energy savings potential. The final report from the first year includes the background of the research, the FRP building characteristics, the baseline HVAC and VRF system characteristics, and the final results for the estimated energy savings of the VRF system and the CCM control algorithm (Im et al. 2015).

Based on the lessons and findings from the project's first year and further discussions with the manufacturer, the goal of the second year was developed. The first-year research confirmed the potential of a VRF system to reduce energy use and enhance indoor thermal comfort. At the same time, it emphasized the need to explore several other aspects of the performance of the VRF systems, including the following.

- **Analysis of part-load performance of VRF and baseline HVAC system:** VRF systems are known for their superior part-load performance compared with conventional HVAC systems. The future study could include an evaluation and comparison of the part-load performance of the VRF system and the baseline system in ORNL's FRP.

- **Analysis of VRF and baseline HVAC system performance in different climates:** To evaluate the performance of a VRF system in different climates, a simulation-based energy analysis could be performed. A building energy model could be developed, calibrated using the measured VRF and the baseline HVAC system performance data, and simulated with corresponding typical meteorological year weather files.

- **Application of a dedicated outdoor air system (DOAS):** After the first year, it was suggested that a DOAS be included with the VRF system to provide adequate fresh air to the indoor space according to the requirements of the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Standard 62.1 (ASHRAE 2013). In the first year, unconditioned outdoor air was introduced into the plenum space by continuous operation of an exhaust fan.

## 1.2 PURPOSE/OBJECTIVE

Therefore, for the second-year project, the following two main research objectives were defined.

1. To modify the previously installed VRF system and add a DOAS on the two-story FRP as one of multiple HVAC options and compare its full- and part-load performance with that of another baseline HVAC system (i.e., a rooftop packaged HVAC system with VAV reheat).

2. To evaluate the energy savings potential of using VRF systems in various US climate zones based on calibrated simulation modeling.

## 1.3 ORGANIZATION OF THE REPORT

Chapter 2 describes a methodology to evaluate the full- and part-load performance of VRF systems compared with the baseline RTU system and to use calibrated simulation modeling to evaluate nationwide the energy savings potential of VRF systems. Chapter 3 presents the heating season data analysis to evaluate the energy use and indoor conditions for different HVAC operation scenarios. In-depth performance analyses for each system for both seasons are also discussed. Chapter 4 concludes the study with a summary, findings, lessons, and discussion of future work.

# 2. METHODOLOGY

## 2.1 TEST FACILITY: TWO STORY FLEXIBLE RESEARCH PLATFORM (FRP)

The test facility is a two-story, 3,200 $ft^2$ (297.3 $m^2$) multi-zone unoccupied building that represents a typical low-rise, small office building common in the US existing building stock (Figure 1). The occupancy in the building can be simulated by process control of lighting and other internal loads. In this building, retrofits and alternative building components and systems can be implemented and their performance monitored. In addition, a dedicated weather station is installed on the roof that provides actual weather data for use in performance analysis and energy modeling. The building is equipped with a conventional 12.5 ton (44 kW) RTU-VAV reheat system. For this study, a 12 ton (42 kW) VRF with a DOAS was installed (Figure 1, center and right), and the existing RTU system served as the baseline system. Table 1 summarizes the baseline building and system characteristics.

**Figure 1. Test facility (left), VRF system outdoor unit (center), and indoor unit (right).**

**Table 1. Building characteristics**

| | |
|---|---|
| **Location** | Oak Ridge, Tennessee, USA |
| **Building size** | Two-story, 40×40 ft (12.2×12.2 m), 14 ft (4.3 m) floor-to-floor height |
| **Exterior walls** | Concrete masonry units with face brick, $R_{US}$-11 ($R_{SI}$-1.9) fiberglass insulation |
| **Floor** | Slab-on-grade |
| **Roof** | Metal deck with $R_{US}$ –18($R_{SI}$ –3.17) polyisocyanurate insulation |
| **Windows** | Double-pane clear glazing, 28% window-to-wall ratio |
| **Baseloads** | 0.85 W/$ft^2$ (9.18W/$m^2$) lighting power density, 1.3 W/$ft^2$ (14.04W/$m^2$) equipment power density |
| **Baseline HVAC system** | 12.5 ton, 9.7 EER rooftop unit; 81% AFUE natural gas furnace; VAV terminal units and electric reheat |
| **VRF system** | 12 ton (42 kW) VRF system with a DOAS |

## 2.2 HVAC SYSTEMS

Figure 2 shows the schematic of the two systems. The RTU provides direct expansion cooling, heating with a natural gas furnace, and electric resistance reheat at VAV terminal units. The return air is drawn from each room through an above-ceiling plenum on each floor. Fresh air is introduced through the fan of the DOAS to provide adequate ventilation in accordance with ASHRAE Standard 62.1-2013 (ASHRAE 2013). An exhaust fan is located on each floor and operates continuously. During the heating season, the RTU discharge air temperature was adjusted based on an outdoor air reset schedule (See Figure 3). The natural gas furnace engages if the return air temperature to the RTU drops below the discharge air set point temperature.

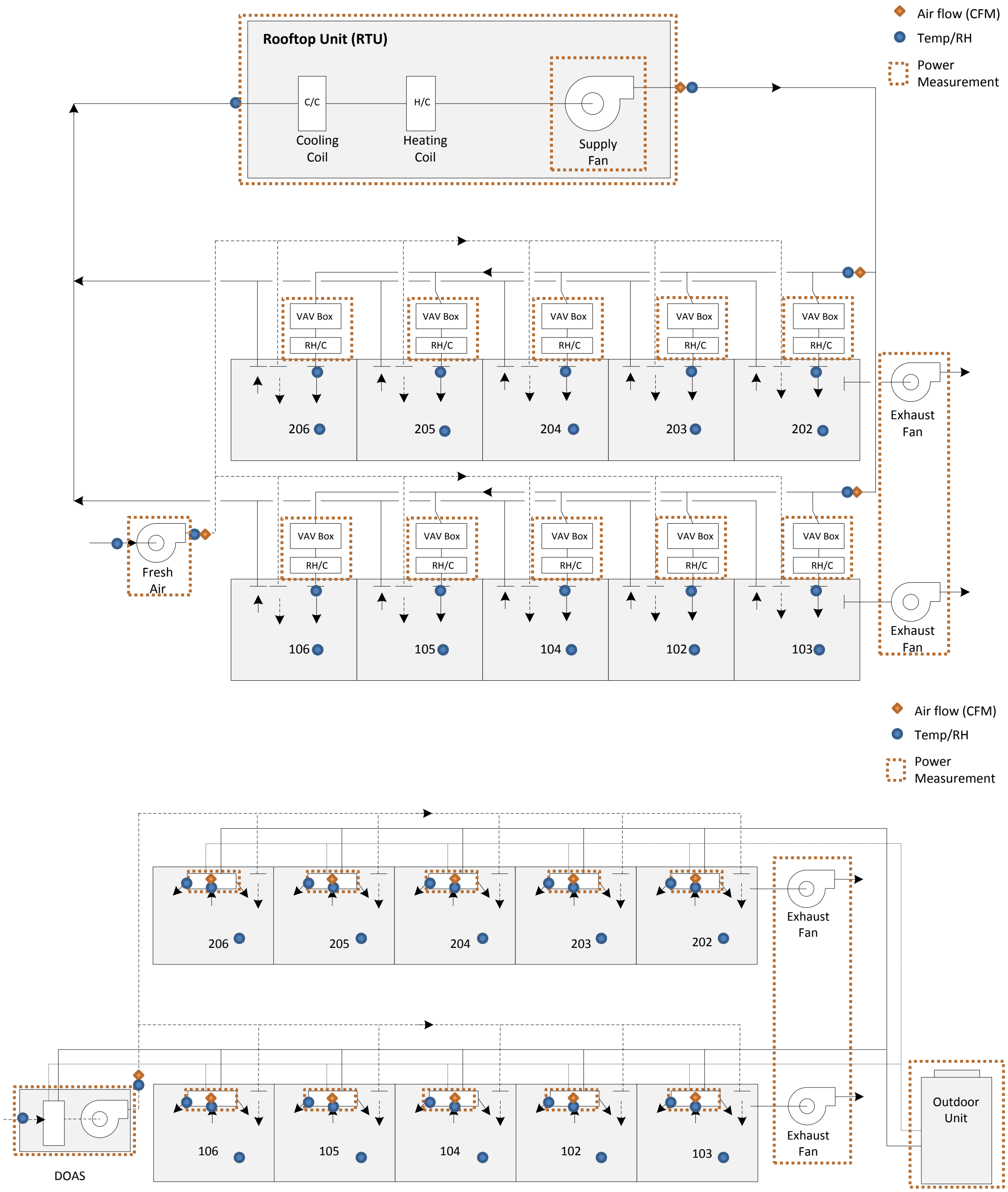



**Figure 2. System schematic and monitoring points for RTU system (above) and VRF system (below).**

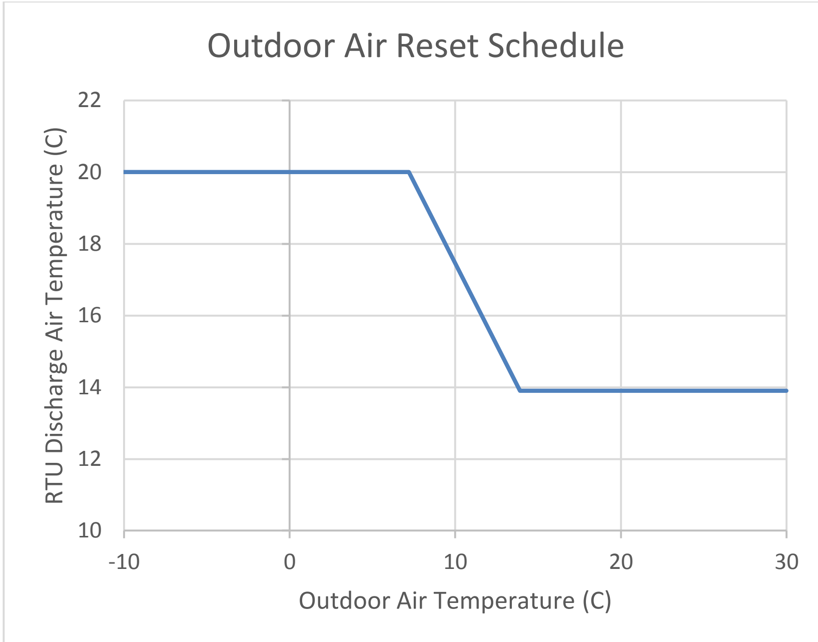



**Figure 3. Outdoor air reset schedule for RTU discharge temperature.**

The VRF system has a 12 ton (42 kW) outdoor unit, one DOAS unit, and ten indoor units with capacities ranging from 0.5 to 1.5 tons (1.8–5.3 kW). Appendix A provides a full specification of the outdoor and indoor units. The ten indoor units and the DOAS (Figure 4) are connected to the same VRF outdoor condensing unit, and the DOAS provides conditioned outdoor air to ten zones. Note that the VRF system in this test is a heat pump type of system that provides only cooling or heating at any single time and cannot provide simultaneous heating and cooling for different thermal zones.

## 2.3 BUILDING OPERATION

The VRF system and the baseline RTU system performance was evaluated by comparing their hourly and daily energy use and the indoor thermal conditions (i.e., temperature). In year 1, the entire building was conditioned for the performance evaluation, which had a limitation in part-load performance evaluation. In year 2, the building was partially and fully conditioned to compare the part-load performance of the systems in more detail. Figure 5 illustrates the schematic of operation to emulate (a) 50% load, (b) 75% load and (c) 100% load[1]. Each system was operated alternately for 8 consecutive days, with 3 days each for 50 and 75% and 2 days each for 100% loads. Unlike in year 1, there was no discrete weekday and weekend schedule, and the same occupancy schedule was used for all days.

Table 2 shows the schedule of the heating season operation of the RTU and VRF systems under different loads. The green-shaded cells indicate the days of VRF system operation, and the orange shaded cells indicate the days of VRF system operation. As shown in Table 2, the heating season analysis is based on measured data from December 30, 2015, through March 6, 2016. During this period, the RTU system was operated for 9 days at 50% load, 9 days at 75% load, and 13 days at 100% load. The VRF system was operated for 9 days at 50% load, 12 days at 75% load, and 11 days at 100% load. Based on the schedule, occupancy emulation also was controlled so that only conditioned rooms had emulated occupancy.

[1] The operation scenarios were called 50% and 75% loads based on the combined rated capacity of the indoor units.

**Figure 4. Installation of dedicated outdoor air system.**

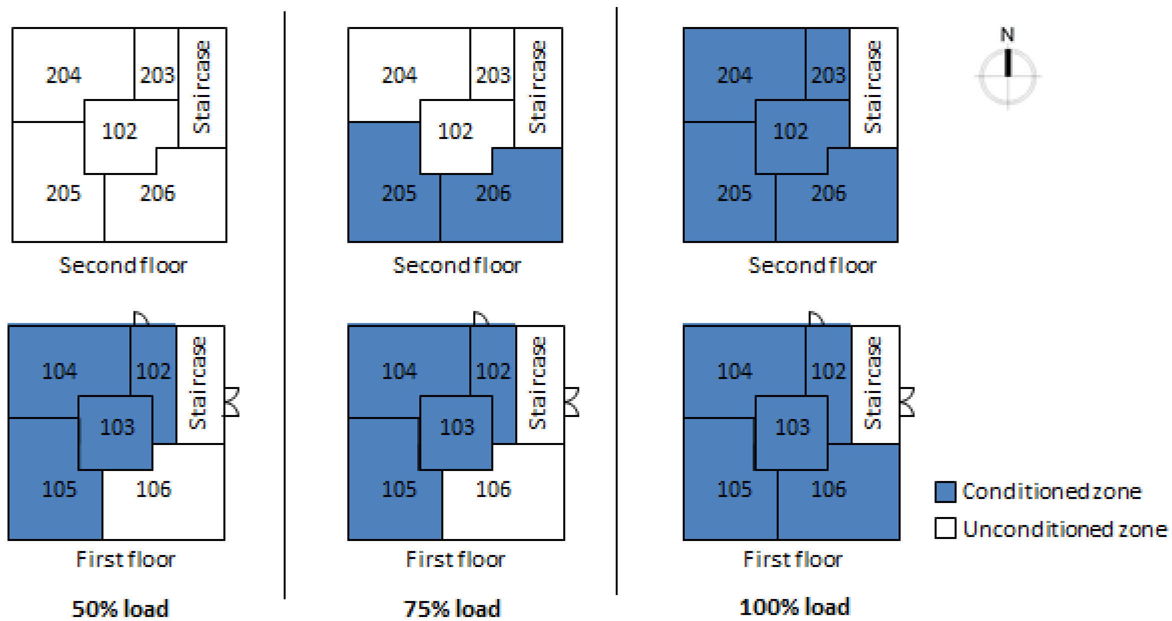



**Figure 5. Schematic of operation to emulate (a) 50% load, (b) 75% load, and (c) 100% load.**

**Table 2. Winter season operation schedule**

| Date | RTU50 | RTU75 | RTU100 | VRF50 | VRF75 | VFR100 | Date | RTU50 | RTU75 | RTU100 | VRF50 | VRF75 | VFR100 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| 30-Dec | | | | | | | 2-Feb | | | | | | |
| 31-Dec | | | | | | | 3-Feb | | | | | | |
| 1-Jan | | | | | | | 4-Feb | | | | | | |
| 2-Jan | | | | | | | 5-Feb | | | | | | |
| 3-Jan | | | | | | | 6-Feb | | | | | | |
| 4-Jan | | | | | | | 7-Feb | | | | | | |
| 5-Jan | | | | | | | 8-Feb | | | | | | |
| 6-Jan | | | | | | | 9-Feb | | | | | | |
| 7-Jan | | | | | | | 10-Feb | | | | | | |
| 8-Jan | | | | | | | 11-Feb | | | | | | |
| 9-Jan | | | | | | | 12-Feb | | | | | | |
| 10-Jan | | | | | | | 13-Feb | | | | | | |
| 11-Jan | | | | | | | 14-Feb | | | | | | |
| 12-Jan | | | | | | | 15-Feb | | | | | | |
| 13-Jan | | | | | | | 16-Feb | | | | | | |
| 14-Jan | | | | | | | 17-Feb | | | | | | |
| 15-Jan | | | | | | | 18-Feb | | | | | | |
| 16-Jan | | | | | | | 19-Feb | | | | | | |
| 17-Jan | | | | | | | 20-Feb | | | | | | |
| 18-Jan | | | | | | | 21-Feb | | | | | | |
| 19-Jan | | | | | | | 22-Feb | | | | | | |
| 20-Jan | | | | | | | 23-Feb | | | | | | |
| 21-Jan | | | | | | | 24-Feb | | | | | | |
| 22-Jan | | | | | | | 25-Feb | | | | | | |
| 23-Jan | | | | | | | 26-Feb | | | | | | |
| 24-Jan | | | | | | | 27-Feb | | | | | | |
| 25-Jan | | | | | | | 28-Feb | | | | | | |
| 26-Jan | | | | | | | 29-Feb | | | | | | |
| 27-Jan | | | | | | | 1-Mar | | | | | | |
| 28-Jan | | | | | | | 2-Mar | | | | | | |
| 29-Jan | | | | | | | 3-Mar | | | | | | |
| 30-Jan | | | | | | | 4-Mar | | | | | | |
| 31-Jan | | | | | | | 5-Mar | | | | | | |
| 1-Feb | | | | | | | 6-Mar | | | | | | |

## 2.4 EVALUATION METRICS

The performance of the RTU and VRF systems was compared in terms of (1) energy use, (2) ability to maintain room temperature, and (3) system efficiency. The energy use and thermal performance comparison were performed using measured hourly data for occupied hours only, excluding the startup hours (8 a.m. to 6 p.m.). The coefficient of performance (COP) analysis was performed using both hourly and 1-minute data. Figure 2 shows the air-side monitoring points, which include the room temperature and relative humidity; supply, return, and mixed-air temperatures; and relative humidity. The airflow rate was measured at the RTU supply, upstairs and downstairs of the supply duct, and at the fresh air supply for the RTU system. For the VRF system, one-time airflow measurements for each indoor unit were conducted, and the DOAS supply side for the VRF system was measured continuously. Power measurements were obtained separately for the RTU, supply fan, and DOAS fan for the baseline RTU system. Power consumption for the VRF outdoor unit, each VRF indoor unit, and the DOAS was measured as well. Table 3 lists the sensors that were installed for the VRF system. The measured supply, return temperature and relative humidity, and airflow were used to calculate the delivered heating and cooling loads to the building and system COP as well.

**Table 3. List of sensors for VRF systems**

| Function | Sensor | Sensor model | Location | Quantity | Accuracy |
|---|---|---|---|---|---|
| **Indoor thermal condition heating and cooling capacities** | Temp/RH | Campbell Sci HC2S3-L | All air side | 20 | ±0.1°C and ±0.1% RH @ 23°C |
| | Air flow sensors (for DOAS) | Air monitor fan evaluators paired to DPT2500 Plus transmitters | DOAS | 1 | DTP2500: 0.25% of natural span, including hysteresis, deadband, nonlinearity, and nonrepeatability; fan evaluator: ±2% |
| **Power** | Wattnode | Continental Controls WNB-3D-240P, 100 Hz option | ID units | 11 | ±0.5% of reading |
| | CT | Continental Controls ACT-0750-005 | ID units | 22 | ±0.75% of reading |
| | Wattnode | Continental Controls WNB-3D-240P, 100 Hz option | OD unit | 1 | ±0.5% of reading |
| | CT | Continental Controls ACT-0750-020 | OD unit | 3 | ±0.75% of reading |

# 3. HEATING SEASON ANALYSIS

The heating season analysis is based on the measured data from December 30, 2015, through March 6, 2016. During this period, the RTU and VRF systems were operated under 50, 75, and 100% load conditions alternately. The performance of the RTU and VRF systems was compared in terms of (1) energy use, (2) the ability to maintain room temperature, and (3) system efficiency. The energy use and thermal performance comparison were performed using measured hourly data for occupied hours only. The COP analysis was performed using 1-minute data.[2]

## 3.1 HOURLY THERMAL CONDITION ANALYSIS—OCCUPIED HOURS ONLY

Figure 6 shows the measured hourly room temperature statistics (minimum, first quartile, median, third quartile, and maximum) for the occupied hours (8 a.m. to 6 p.m.; i.e., excluding the startup hours) in all rooms during RTU and VRF system operation at the three capacities during heating season. Outdoor air temperature (OAT) statistics are also plotted. The thermostat heating set point is marked as a red line across the plot. Temperature statistics in unconditioned rooms are shaded in gray.

During the heating season, in general, the RTU system provided better thermal control for most of the rooms; rooms 205 and 206, which were overheated, were the exception. Those two rooms face south, east and west, and it was observed that they were overheated mainly as a result of high solar heat gain through the windows during early morning and late afternoon.

For all three operation modes, the VRF system overheated rooms. There are a few reasons for this. As the installed VRF system cannot provide simultaneous cooling and heating, rooms that required cooling during the heating season were overheated. In most cases, the room temperatures stayed below 27°C. The VRF system also experienced the same solar gains as the RTU, leading to overheating of rooms with south-, east-, and west-facing glazing. Additionally, the DOAS is supposed to provide conditioned fresh air at a temperature close to the room temperature. However, the measured data show that the discharge temperature of the DOAS is close to 40°C, which could overheat the rooms (Figure 7). Therefore, it is strongly recommended that the DOAS be able to provide conditioned air close to the room air temperature to avoid overheating rooms. In addition, this type of small office building with a high window area should have a heat recovery–type VRF system so that the system can meet the building's simultaneous cooling and heating demands during the heating and shoulder seasons.

[2] Unlike in the cooling season analysis, hourly COP analysis was not done. In VRF heating operation, indoor unit fans are not running continuously, so it is hard to measure a precise delta T (i.e., supply minus return air) and airflow rate to calculate the delivered heating load.

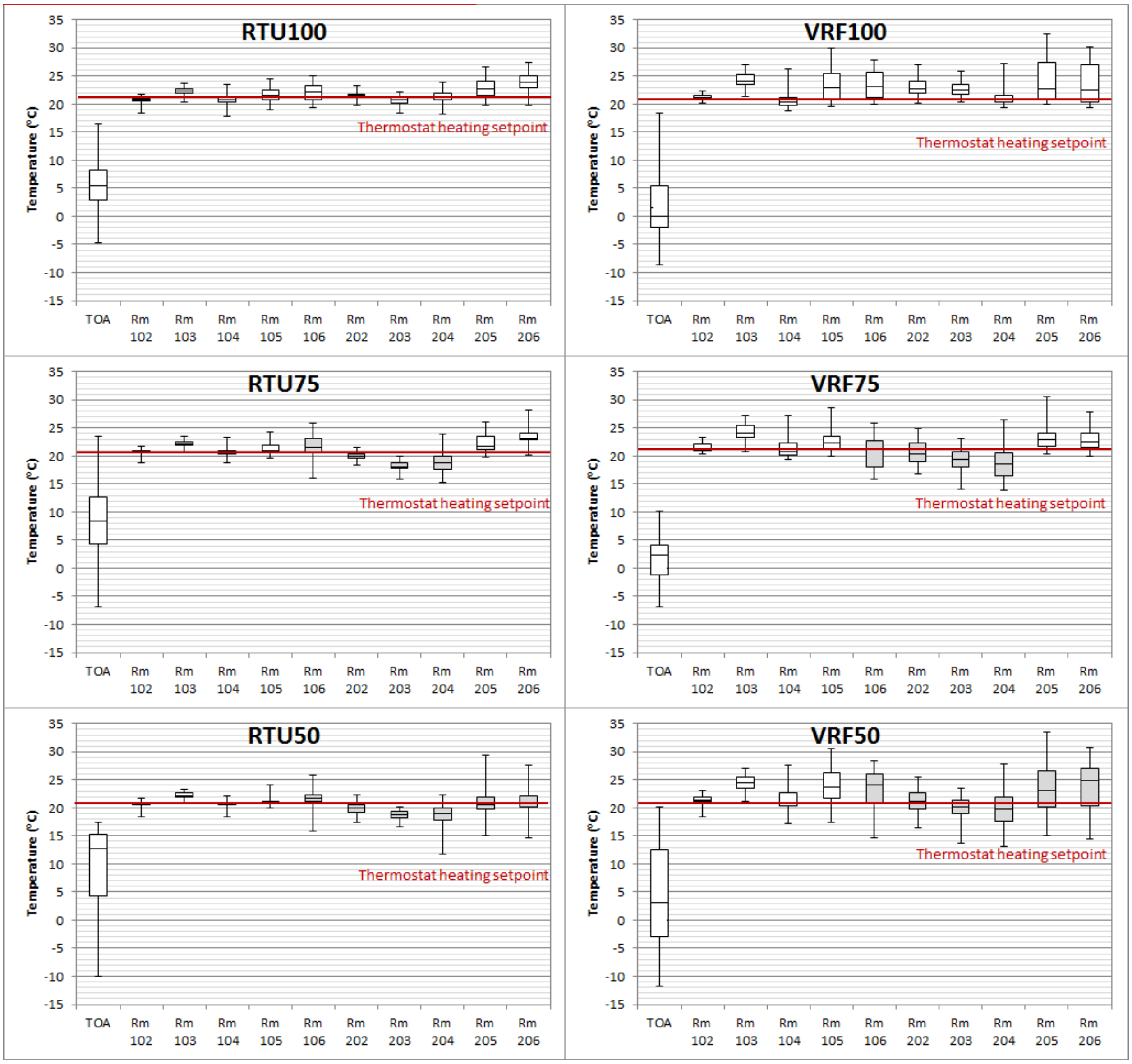



**Figure 6. Room temperature during RTU and VRF system operation at different capacities.**

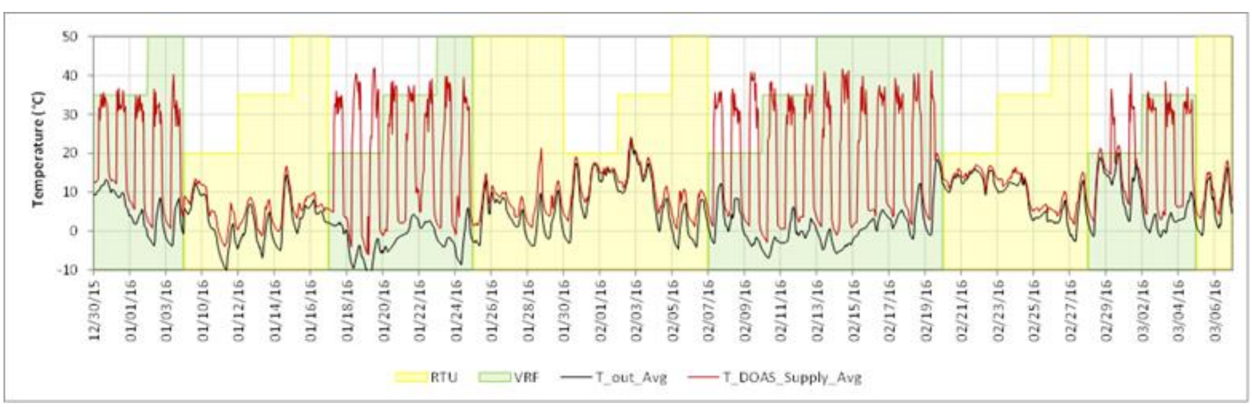



**Figure 7. DOAS discharge temperature during heating season (green-shaded area shows the discharge temperature during VRF operation).**

## 3.2 PERFORMANCE DURING TYPICAL WINTER DAYS

Figure 8 shows the hourly energy consumption and room temperatures for typical winter days during RTU and VRF system operation at the three capacities. In a similar fashion to the cooling season analysis, days were selected that had relatively similar OAT profiles, and temperatures were plotted only for rooms 102, 103, 104 and 105, which were conditioned in all control modes.

The data analysis shows that both systems exhibited a similar energy use pattern of high energy use during startup at 6:00 a.m. to reach the thermostat set point (21°C) as the occupancy mode changed from "unoccupied" to "occupied," and then a drop in energy use as the room temperature reached the thermostat set point. In addition, the analysis shows that the RTU system used natural gas during or before morning startup to satisfy the RTU discharge set point temperature as the return temperature from the zones to the RTU air handling unit (AHU) dropped below the set point temperature.

In general, the RTU system used far more startup energy than the VRF system in the mornings. In the afternoons, the energy use for the RTU and the VRF system appeared to be similar, except during 50% operation. For the VRF 75 and 100% operation modes, some rooms appeared to be overheated mainly for the reasons explained in Section 3.1.

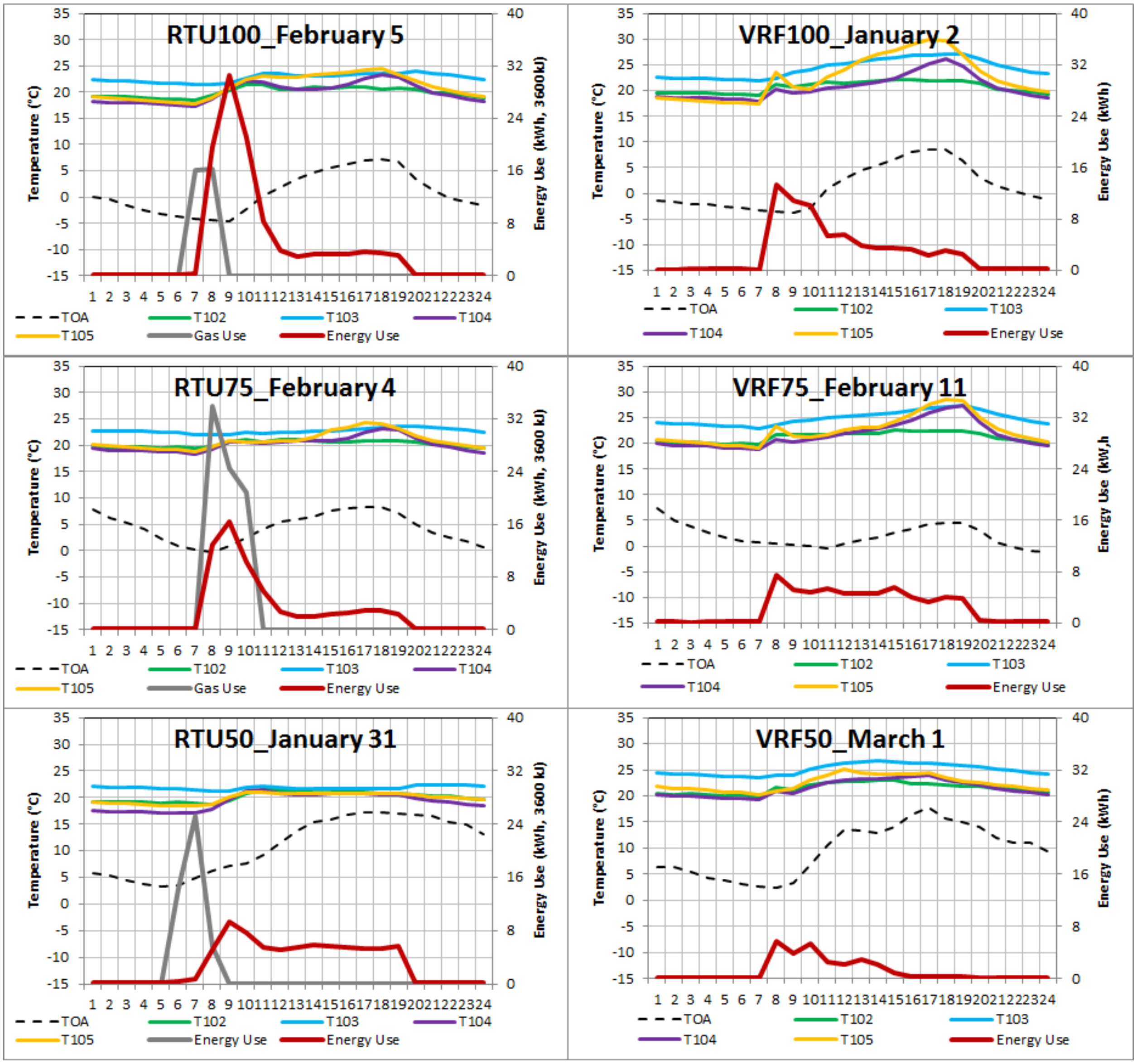



**Figure 8. Room temperature and energy use for a typical day during RTU and VRF system operation at different loads.**

### 3.3 POWER CONSUMPTION ANALYSIS

Figure 9 shows a comparison of power consumption during the occupied hours when the RTU and VRF systems were operated under different loads. The electricity used by the RTU operation is a sum of the electricity used by the RTU (compressor + condenser fan) and by the VAV electric reheating equipment. In addition, natural gas was used by the RTU gas furnace. The VRF system shows a relatively strong correlation between energy use and the OAT, whereas the RTU total energy does not show a strong trend with the OAT. Hence, developing a weather-normalized model for RTU energy use requires considering many more issues than does the RTU model for cooling season. The RTU natural gas heating and direct exchange cooling operate to maintain the desired supply air temperature defined in the outdoor air reset schedule (Figure 3). The VAV boxes and reheating equipment then operate independently to achieve the desired room conditions. In addition, the fresh air unit for the RTU did not operate below 5°C, which may have led to underestimating the heating energy use. Therefore, the additional heating energy required for conditioning the fresh air should also be calculated and included in the RTU model during the heating season. In light of these issues, a new methodology of developing an RTU model was considered.

### 3.3.1 RTU Power Consumption Analysis

For the RTU power consumption analysis, first, a delivered building load was calculated according to Eq. (1):

$$\text{Building Load (kW)} = RTU_{mc}(\text{Tin-Tout}) + VAV_{Reheating} + M\_DOAS_{add} \ , \tag{1}$$

where

$RTU_{mc}$ (Tin-Tout)(kW): RTU delivered load (measured),
$VAV_{Reheating}$ (kW): VAV reheating energy (measured),
$M\_DOAS_{add}$ (kW): Ventilation load that should be added below 5C of OAT (modeled).

For RTU operation, an additional ventilation load needs to be calculated and added when the OAT is below 5°C, as the DOAS fan did not operate below 5°C. We assumed the additional ventilation load was met by additional VAV reheating. The regression model developed for additional VAV reheating due to an additional DOAS load is shown in Table 4. The positive and negative building loads represent net heating and net cooling loads, respectively.

Figure 10 shows the calculated hourly building loads for all three operations as outdoor temperature changes. For the building loads, regression models were developed by using multivariable regression including a combination of OAT, $OAT^2$, solar radiation, hour of day, and hour of $day^2$. Table 4 also shows the developed regression models for RTU building loads for all three operations.

As a next step, hourly regression models for RTU hourly energy use ($RTU_{DX+Fan}$), RTU cooling loads ($RTU_{cooling}$), and hourly natural gas use ($NG_{heating}$) were developed separately. Figure 11 shows the regression models developed for $RTU_{DX+Fan}$, $RTU_{cooling}$, and $NG_{heating}$, respectively.

As a final step, the hourly VAV reheating energy, which could not be modeled easily, can be calculated as in Eq. (2). In this way, the VAV reheat is responding to the unmet building load similarly to the way it operates in the real world.

$$VAV_{Reheating} = M\_\text{Building Load} + M\_RTU_{Cooling} - M\_NG_{heating} \ , \tag{2}$$

where

M_Building Load (kW): Regression model for building load,
$M\_RTU_{cooling}$ (kW): Regression model for RTU compressor work (kWh) * COP,
$M\_NG_{heating}$ (kW): Regression model for natural gas use * AFUE (80%).

Figure 12 shows the calculated VAV reheating energy based on Eq. (2), and Figure 13 shows the hourly total RTU energy use, which is the sum of RTU energy, VAV reheating energy, and natural gas use. The final $R^2$ values from the model, compared with the measured energy uses for RTU 100, 75, and 50% operation, were 0.88, 0.83, and 0.79, respectively.

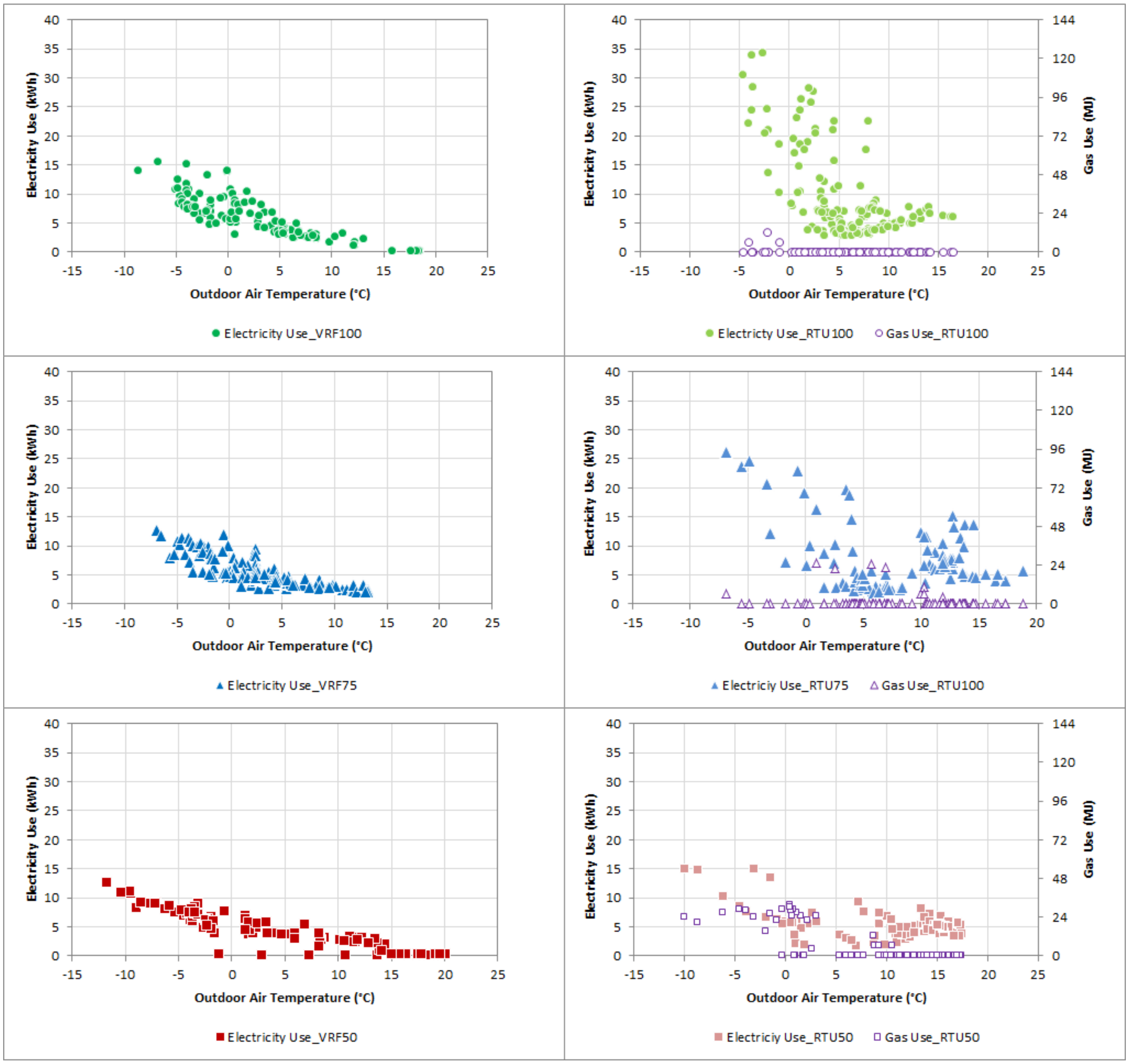



**Figure 9. Comparison of hourly power consumption during occupied hours at varying loads.**

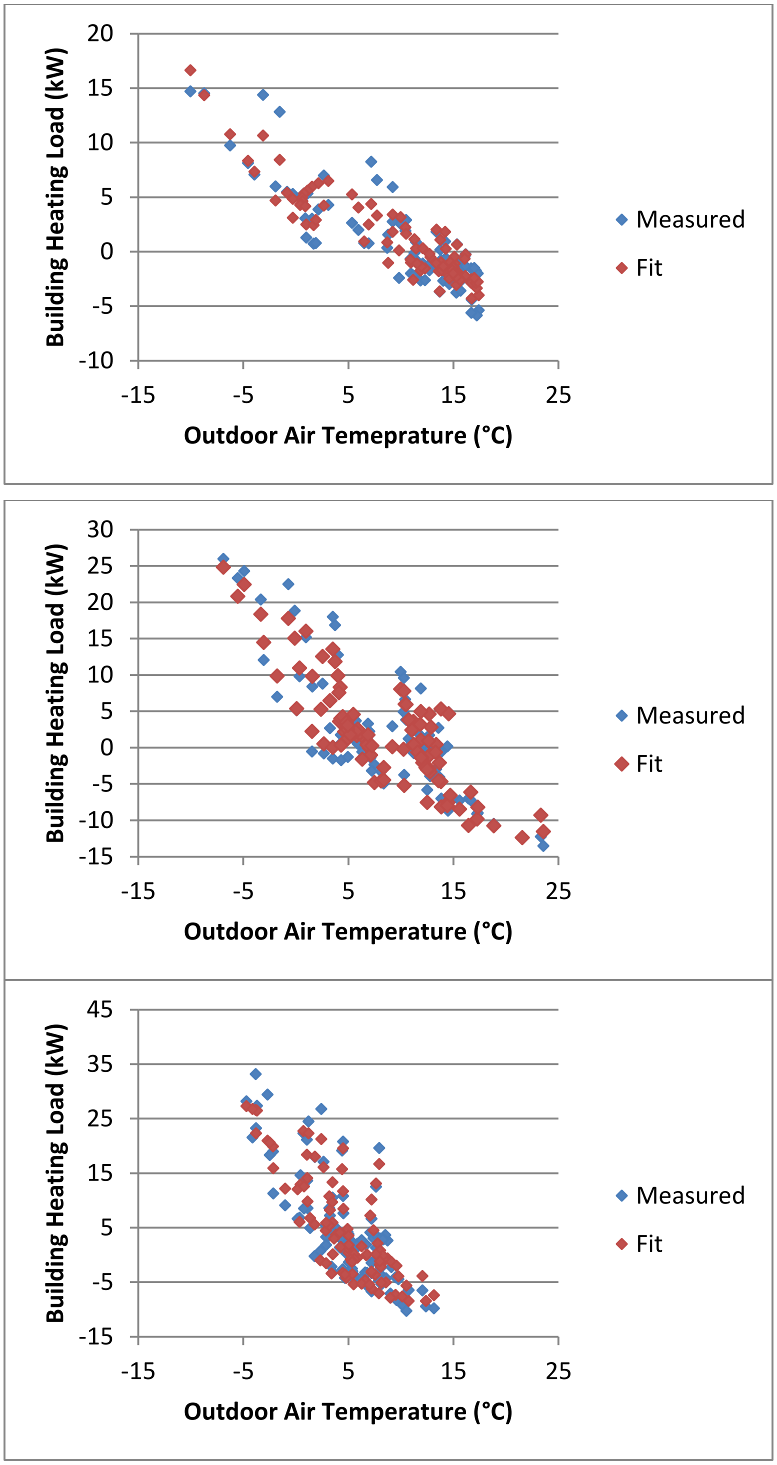


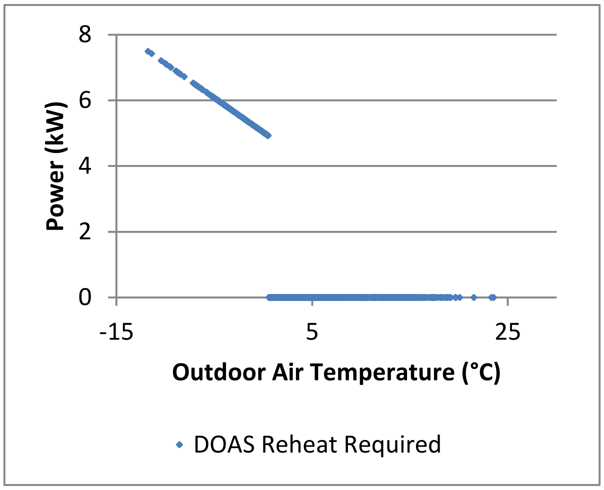

**Figure 10. Calculated hourly RTU building loads.**

**Table 4. Regression models for RTU power consumption for heating season**

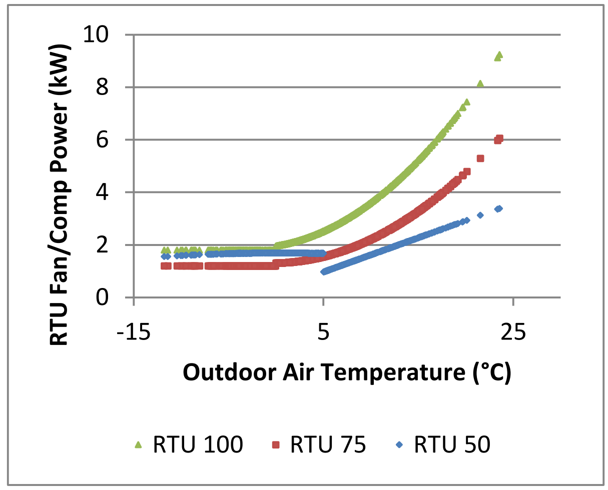
| Category | Operation | Regression models | R2 |
|---|---|---|---|
| Building load (kW) | RTU100 | $-0.8459a - 0.0002b + 0.2256c^2 - 7.5075c + 68.98$ | 0.92 |
| | RTU75 | $0.01338a^2 - 1.0334a - 0.0002b + 0.0488c^2 - 2.1728c + 31.317$ | 0.90 |
| | RTU50 | $0.01458c^2 - 0.6714a - 0.000066b + 0.0434c^2 - 1.4859c + 17.5349$ | 0.89 |
| $M\_DOAS_{add}$ (kW) | RTU100 | $-0.1891a + 5.1779$ | N/A |
| | RTU75 | $-0.1891a + 5.1779$ | N/A |
| | RTU50 | $-0.1891a + 5.1779$ | N/A |
| $M\_RTU_{DX+Fan}$ (kW) | RTU100 | $0.0135a^2 + 0.0261a + 1.91$ | 0.89 |
| | RTU75 | $0.00881a^2 + 0.00381a + 1.22$ | 0.86 |
| | RTU50 (If a<5°C) | $0.00078a^2 + 0.0125a + 1.68$ | 0.34 |
| | RTU50 (If a>5°C) | $0.1304a + 0.315$ | 0.82 |
| $M\_RTU_{cooling}$ (kW) | RTU100 | $-0.1119a - 0.3402d^2 + 6.8119d - 13.0405$ | 0.98 |
| | RTU75 | $2.6080d - 2.8632$ | 0.92 |
| | RTU50 | $-0.09736a - 0.92115d^2 + 7.4314d - 6.0403$ | 0.91 |
| $M\_NG_{heating}$ (kW) | RTU50 (If a<4.4°C) | $0.0586a + 7.3972$ | 0.07 |
| | RTU50 (If 4.4°C<a <10°C) | $-0.7315a + 9.0857$ | 0.91 |
| | RTU50 (If a>10°C) | Zero | N/A |

where

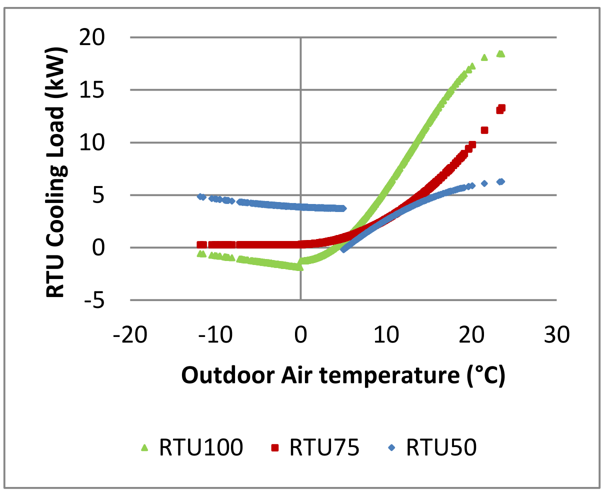
*a*: Outdoor air temperature (°C)
*b*: Global horizontal solar radiation (W/m2)
*c*: Time of day
*d*: $M\_RTU_{DX+Fan}$



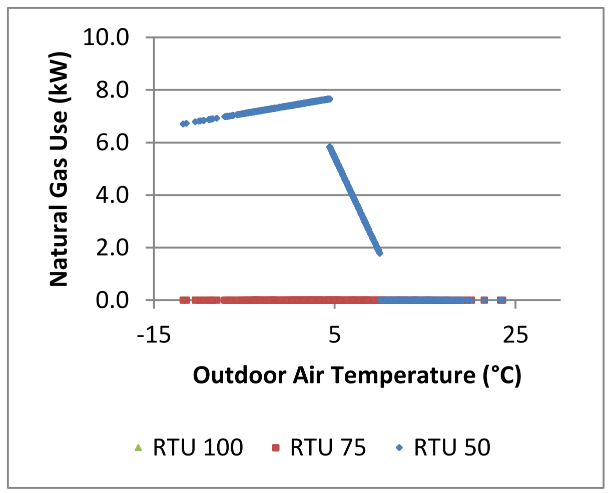

*Additional hourly VAV reheating due to additional ventilation load*




*Hourly RTU energy use (DX cooling + fan)*




*Hourly delivered RTU cooling loads*




*Hourly natural gas use*

**Figure 11. Regression models for RTU energy use.**

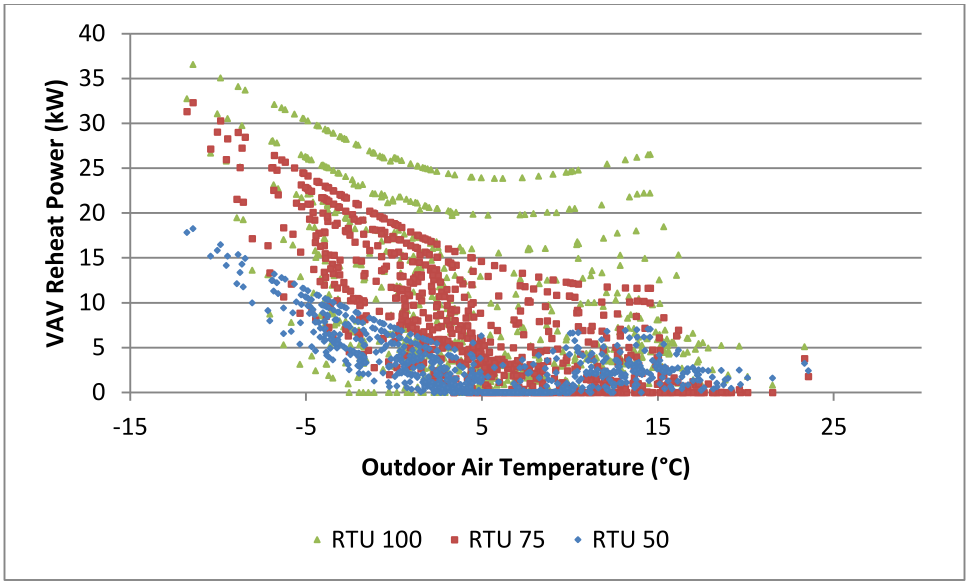



**Figure 12. Calculated hourly RTU VAV reheating energy.**

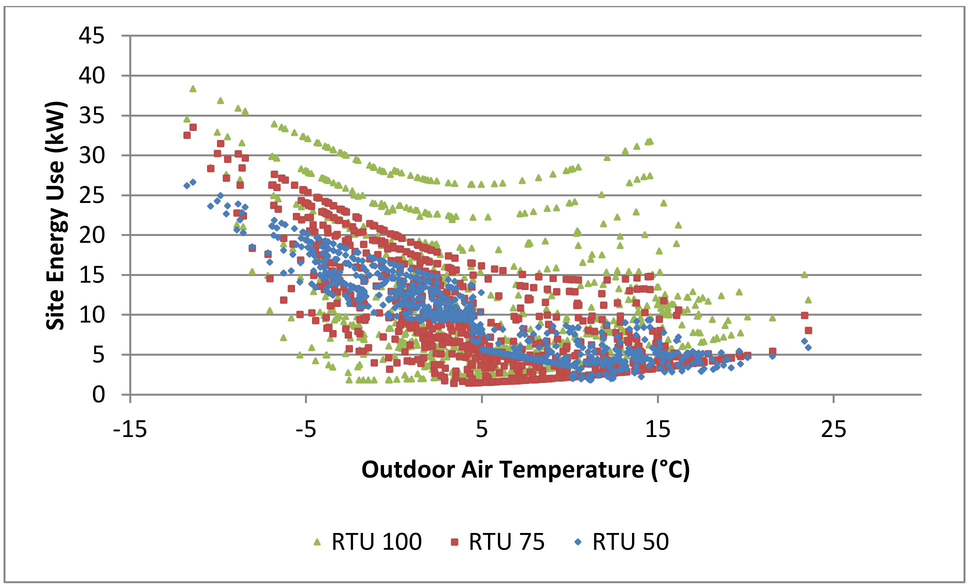



**Figure 13. Hourly total RTU source energy (RTU+VAV+NG).**

### 3.3.2 VRF Power Consumption Analysis

Regression models for hourly VRF energy use were also developed using multivariable regression, including a combination of OAT, $OAT^2$, hour of day, and hour of $day^2$. Hourly solar radiation was also considered as a variable, but the regression model with this additional variable did not show significant improvement in the model fit. The developed models were as shown in Table 5 and Figure 14.

**Table 5. Regression models for VRF power consumption for heating season**

| Category | Operation | Regression models | R2 |
|---|---|---|---|
| VRF hourly energy use (W) | VRF100 | $1.4a^2 - 371.2a + 74c^2 - 2303c + 24391.7$ | 0.83 |
| | VRF75 | $19.7a^2 - 561.6a + 22c^2 - 659c + 11092.5$ | 0.73 |
| | VRF50 | $10.8a^2 - 345.3a + 28c^2 - 907c + 12286.8$ | 0.92 |

where

a: Outdoor air temperature (°C)
c: Time of day

Figure 15 shows the VRF energy use predicted for the entire heating season for 50, 75, and 100% VRF operation. Note that we assumed the minimal energy use of VRF system during heating season to be about 2.2 kW, which is mainly used for the DOAS. In addition, it is assumed that this minimal VRF energy use would be seen for all heating operation above outdoor temperatures of 13°C.

### 3.3.3 Predicted Heating Season Energy Use

Using the regression models for the RTU and VRF, weather-normalized energy use for the period of December 30, 2015, through March 6, 2016 was predicted. Unlike in the cooling season energy comparison, source energy savings were calculated, as the RTU energy use includes both electricity and natural gas use. For the conversion of site energy to source energy, factors of 3.365 and 1.092 were used for electricity and natural gas, respectively.

The final calculation shows that the VRF system saved 51, 47, and 27% of the source heating energy use at 100, 75, and 50% loads, respectively (Figure 16). Unlike in cooling season operation, it appears that the RTU energy use dramatically dropped as the building was used partially (i.e., 75 and 50%). However, the energy use for the VRF in part-load operation did not show much difference, relatively, resulting in lower energy savings in part-load operation. For the RTU, since zone-level reheating represented the main use of energy for heating, partial room operation showed proportionally reduced reheating energy use as well as reduced RTU fan energy. For the VRF, the slight difference in energy use for full- and part-load operation was mainly due to the high heating energy use of the DOAS. As mentioned earlier, the DOAS supply temperature is higher than the discharge set point temperature, and the DOAS was running continuously in all ten zones even during partial room operation. As the DOAS uses a major portion of the total heating energy, the VRF system energy use for all three operation modes is similar. It is expected that partial room operation would show lower energy use if there were no DOAS running or if the DOAS could provide conditioned air only to conditioned rooms.

Additional saving analysis was performed assuming the VAV reheating would be provided by hot water generated by NG boiler. Given the 80% efficient boiler and a generic part load performance curve, the hourly HW reheating energy was calculated and replaced the modeled electric resistant heating. The savings calculation with this scenario shows that the VRF system saved 18, 9, and 8% of the source heating energy use at 100, 75, and 50% loads, respectively. Given the relatively small size of the building and ease of installation for electric reheating, however, the HW reheating with boiler room would not be ideal for this building.

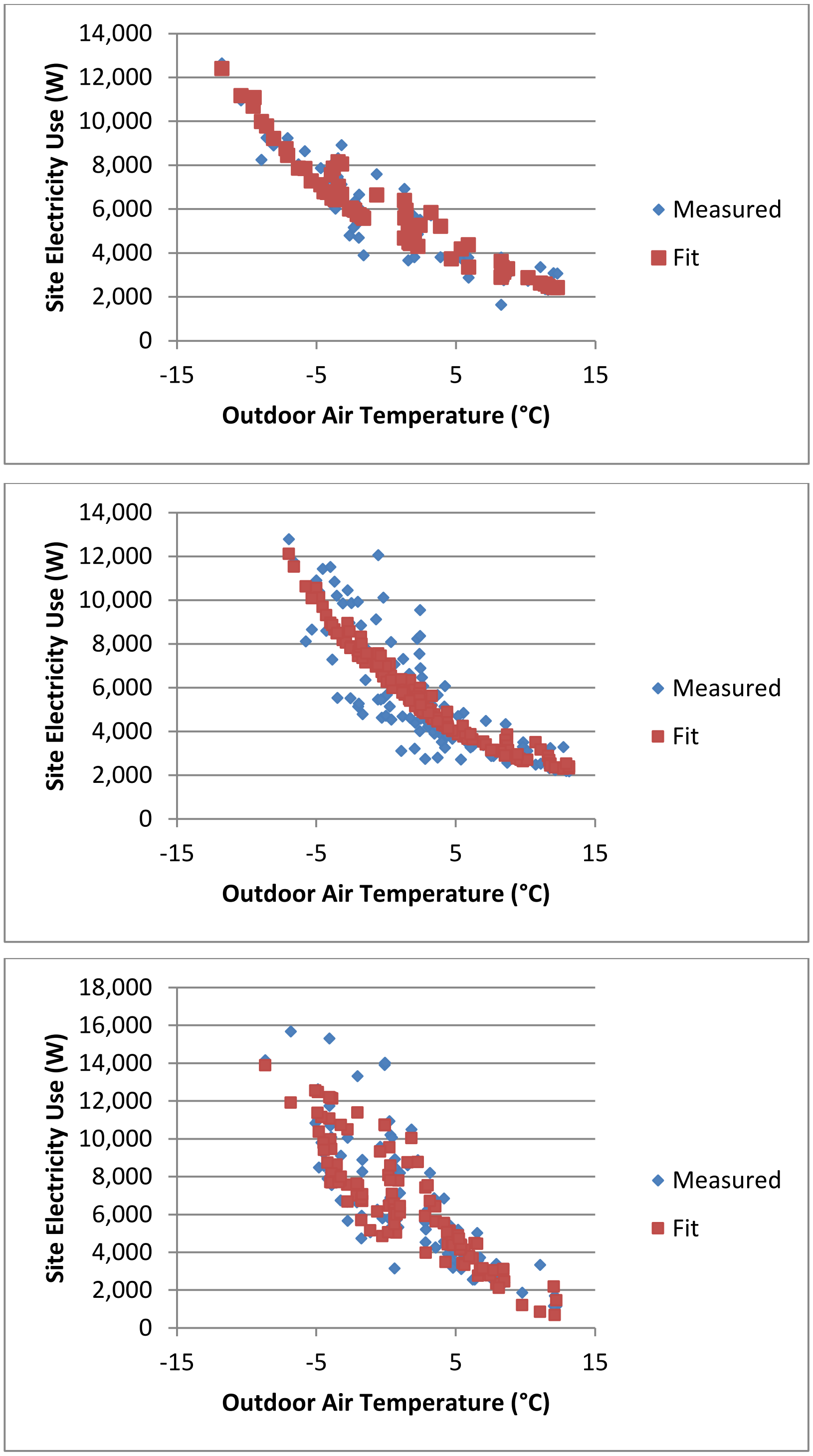



**Figure 14. Hourly VRF energy and regression models.**

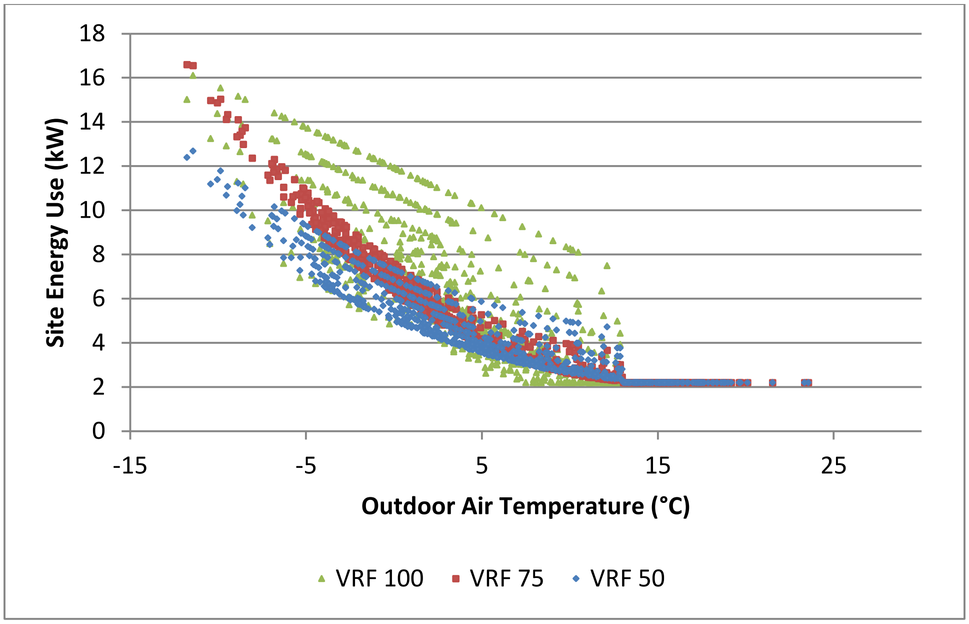



**Figure 15. Hourly VRF energy predicted over the entire heating season.**

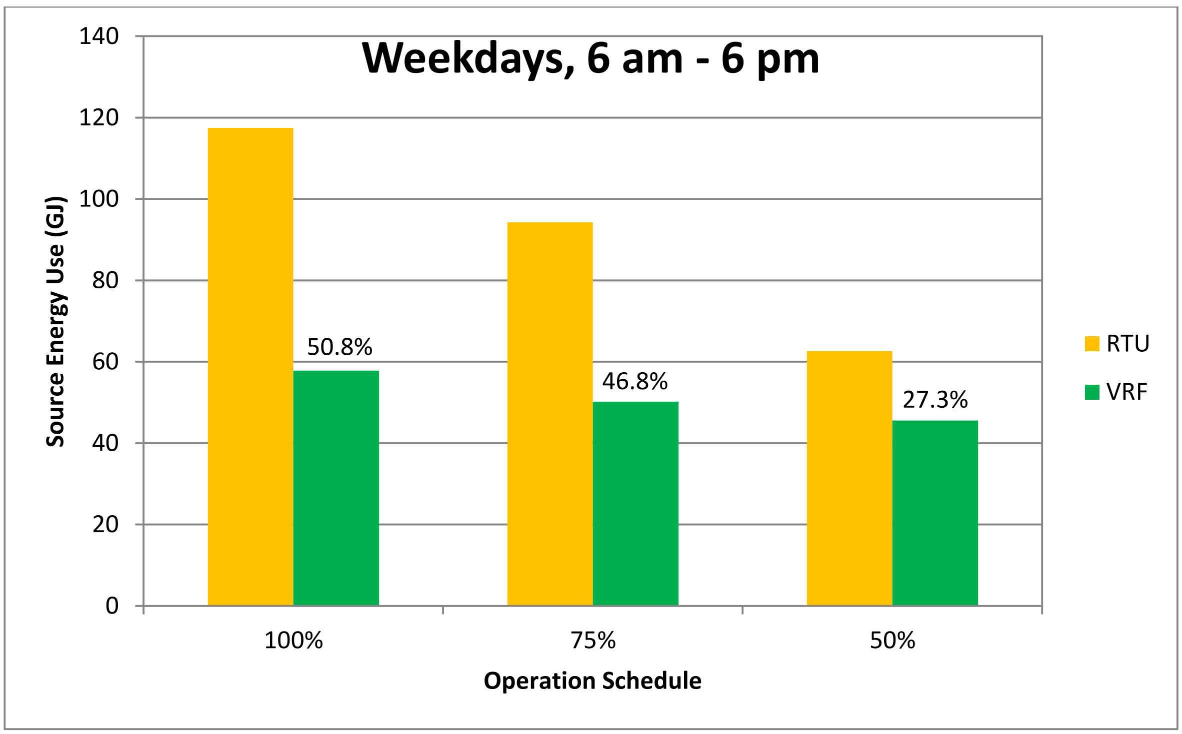



**Figure 16. Comparison of VRF and RTU energy use over the entire heating season.**

## 3.4 QUASI–STEADY-STATE COP ANALYSIS

### Performance analysis

The performance of both the RTU and VRF systems was analyzed using 1-minute resolution data. Periods of time when the RTU system was providing net cooling were filtered out of the data to leave only heating data. The DOAS did not operate when the OAT was less than 4.5°C and the RTU was operating. Therefore, the additional heating requirement due to ventilation was calculated and added into the data based on Eq. (3). This ensured that the building loads during the operation of the RTU and VRF were as close as possible. The energy use of the RTU was adjusted based on the measured COP of the system for that temperature bin. Thus, the performance of the system was not affected by the addition of the ventilation load.

$$Q = \dot{m}c(T_{in} - T_{out}) \ . \tag{3}$$

The average heating capacities of the RTU and the VRF systems relative to the outdoor ambient temperatures for the different operation schedules are shown in Figure 17. As expected, the heating capacity was reduced for the 75 and 50% schedules because the unoccupied zones did not require heating. The VRF system showed a lower heating capacity for the 75 and 100% schedules at low outdoor temperatures, less than 0°C, and a higher heating capacity for all schedules at mild outdoor temperatures. Comparing the average mean radiant temperature for all of the rooms, not just those being conditioned, shows that the VRF system was heating the building more at mild ambient temperatures (Figure 18). This explains the higher heating capacity of the VRF system at mild temperatures. However, the mean radiant temperatures for the VRF and RTU system at low outdoor temperatures are very similar, indicating that the heating capacities of the systems should be similar despite the measured capacity indicating otherwise.

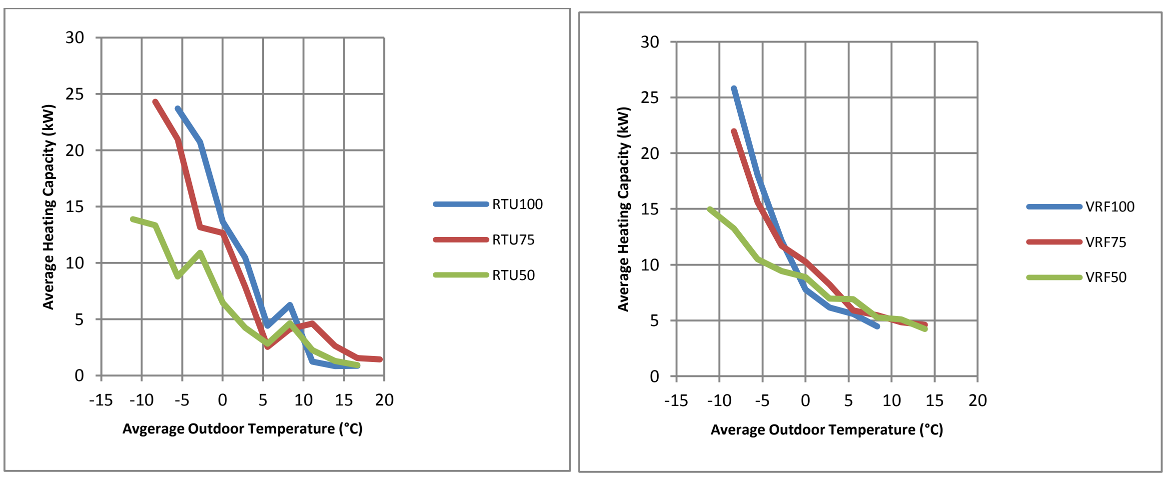



**Figure 17. Average heating capacities of the RTU and VRF systems relative to the outdoor ambient temperature for the different operation schedules.**

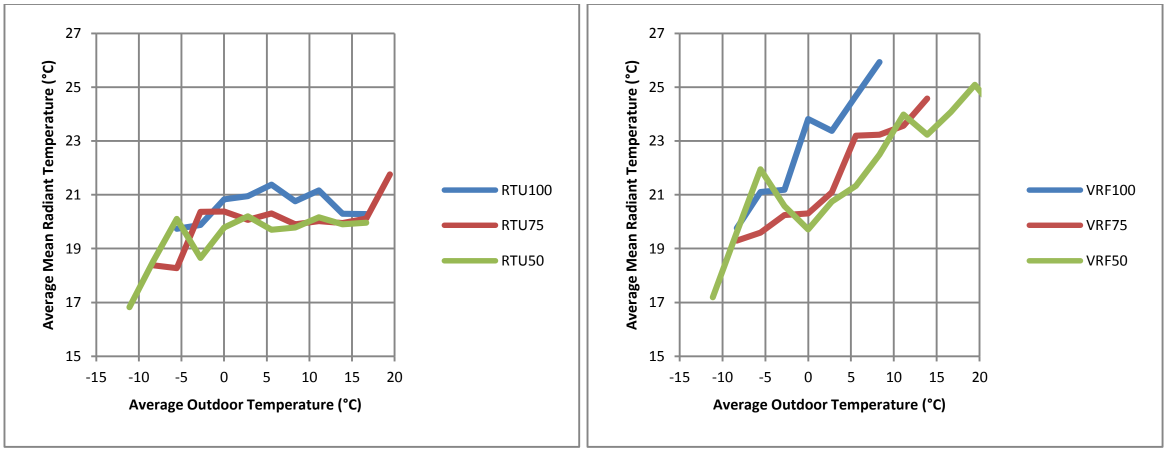



**Figure 18. Average mean radiant temperature of all rooms in building relative to outdoor air temperature for both VRF and RTU systems.**

The site energy use of the systems is compared in Figure 19. The VRF system used significantly less site energy across the entire range of temperatures. However, the RTU is equipped with natural gas heating, which resulted in reduced source energy use relative to electricity use. Figure 20 shows the source energy use of both systems, assuming a 3.2 factor for converting site electricity use to source electricity use and no losses for natural gas use. The only significant change in the shape of the RTU energy use was during the 50% schedule, which had the only significant natural gas use. The use of natural gas was likely due to the reduced return air temperature caused by pulling a portion of the return air from unconditioned rooms. This reduced the return air temperature below the supply air temperature set point of 19.4°C, causing the RTU to use natural gas for heating. However, the heating capacity of the RTU is quite large at 23.8 kW (low stage); the large heating capacity often caused a spike in the supply air temperature that then triggered the RTU to provide cooling to bring the supply air temperature back down. This ping-pong effect was limited to the 50% schedule. The increase in energy use for the RTU system at temperatures between 5 and 20°C was caused by the RTU providing cooling and the VAV boxes reheating that air to provide net heating. The RTU and the VRF system used a similar amount of source energy at around 5°C, but the RTU used significantly more energy at outdoor temperatures above and below that point.

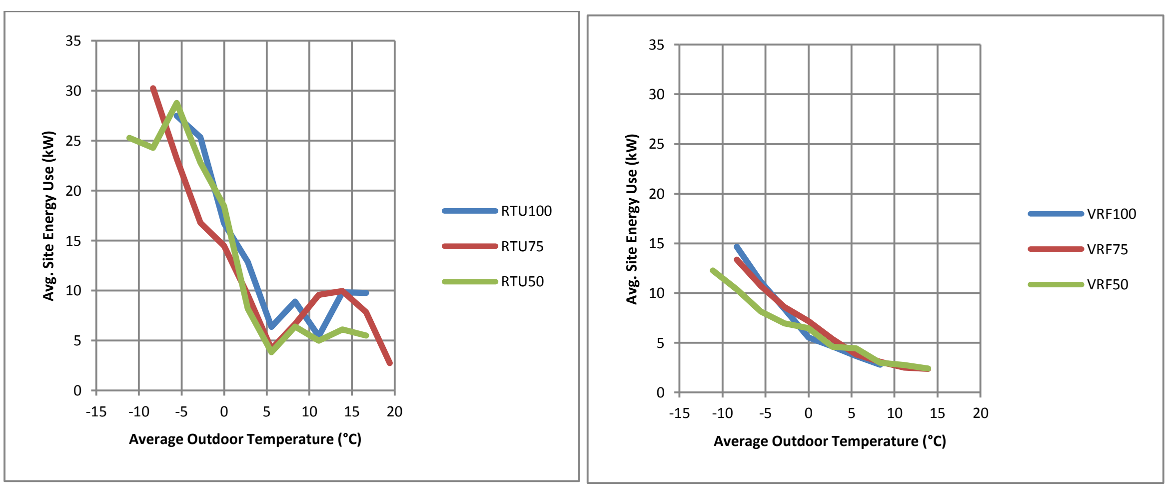



**Figure 19. Site energy use of the RTU and VRF systems relative to outdoor air temperature for all operating schedules.**

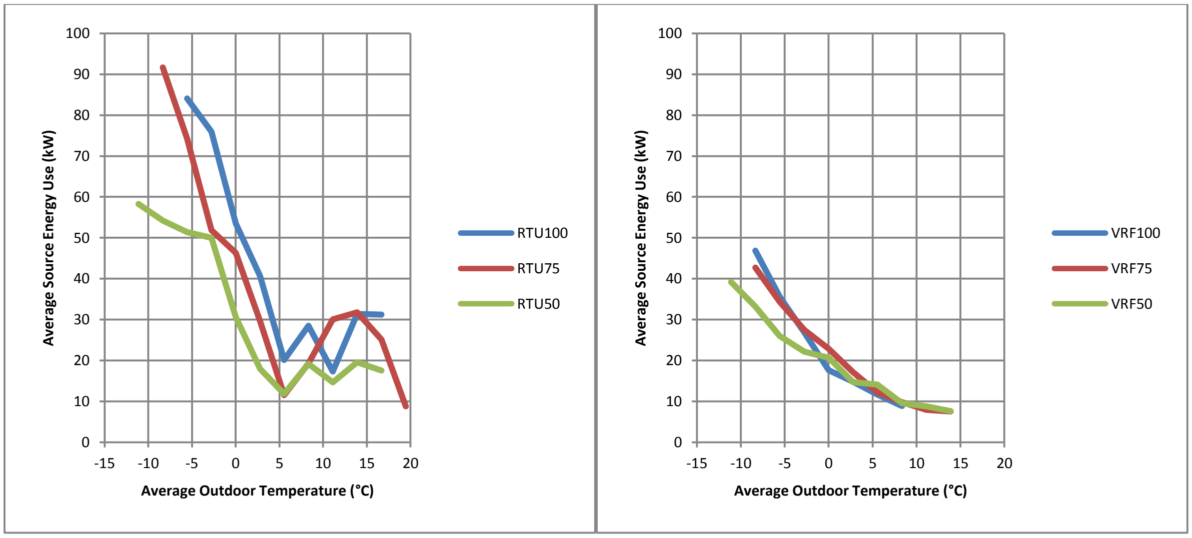
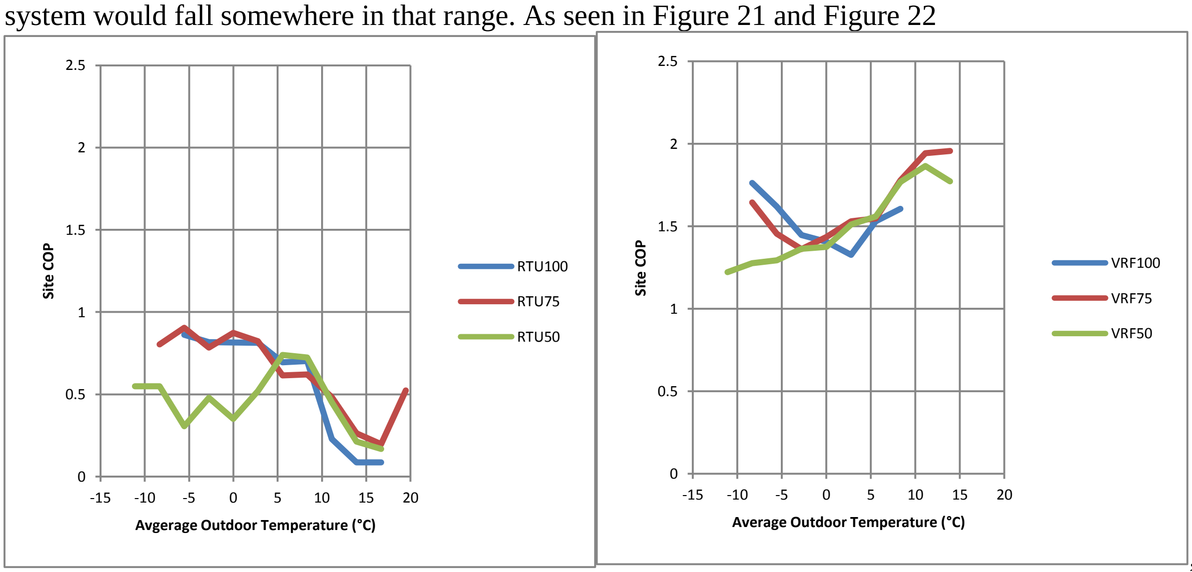



**Figure 20. Source energy use of RTU and VRF systems relative to outdoor temperature for all operating schedules.**

Since the RTU system's two sources of heating are electrical resistance heat in the VAV boxes with a COP of 1, and natural gas heating with a COP of 0.8, it was expected that the overall site COP of this system would fall somewhere in that range. As seen in Figure 21 and Figure 22




, this was true for the 75 and 100% operating schedules for temperatures below approximately 5°C. Above 5°C, the RTU compressor ran to provide cooling that was then offset by the resistance heating in the VAV boxes, reducing the COP of the system. Similarly, the 50% operating schedule also switched between natural gas heating and cooling with reheating at temperatures below 5°C while trying to maintain the appropriate supply air temperature from the RTU. The COPs of the VRF system ranged from 1.2 to 2.0, which are lower than expected but still substantially higher than the COPS of the RTU system.

Since the RTU used very little natural gas, the source COPs shown in Figure 22 tell a similar story as the site COPs shown in Figure 21.

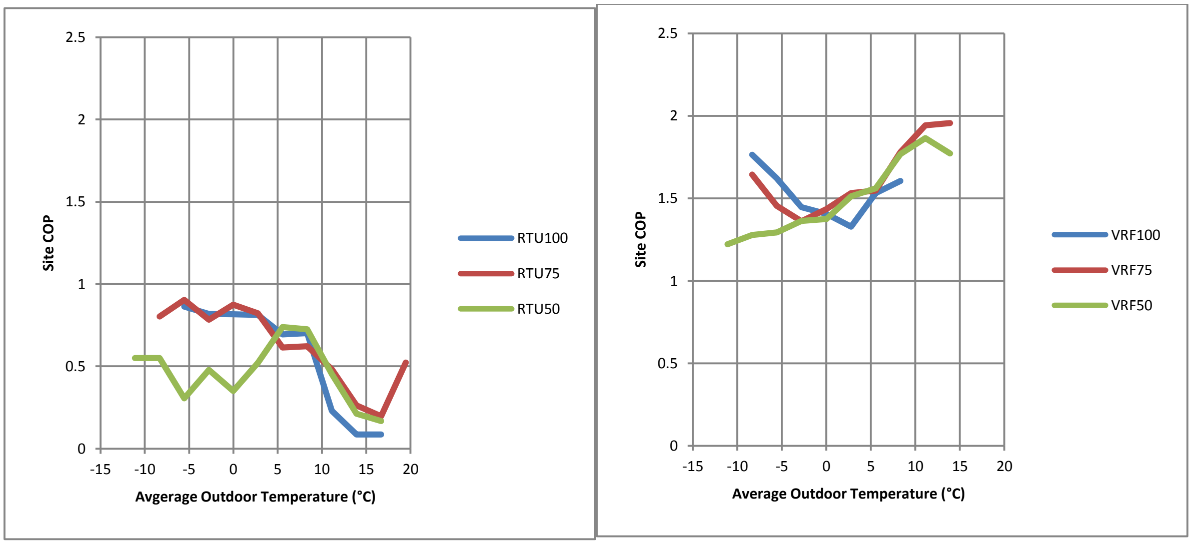



**Figure 21. Site COPs of RTU and VRF systems relative to outdoor air temperature for all operating schedules.**

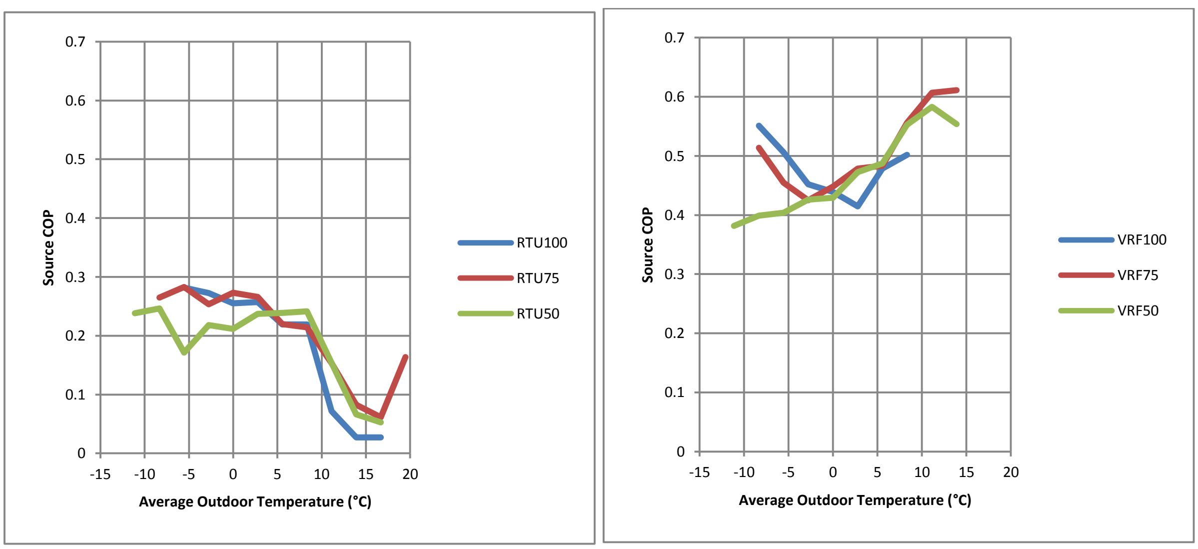



**Figure 22. Source COPs of RTU and VRF systems relative to outdoor air temperature for all operating schedules.**

The VRF system consists of 11 indoor units, 10 to condition the rooms and a DOAS unit to condition ventilation air. Figure 23 shows the fraction of the rated capacity of these units that was actively heating relative to the OAT for the three different operating schedules. It is interesting to note that the DOAS unit capacity was ~16% of the total rated indoor capacity, and it was generally the only unit running for all schedules at outdoor temperatures of 5°C or higher. The 100% schedule also had fewer units actively heating than did the other two schedules in the 0 to 5°C range. This is also reflected in a lower heating capacity than for the other two schedules in this temperature range in Figure 17.

It is also of interest to examine how the VRF system efficiency varied with the fraction of indoor rated capacity that was actively heating (Figure 24). In Figure 24, the marker color indicates the COP, along with the data labels, and the marker shape differentiates among the three operating schedules. It can be seen that, generally, the more units that are actively heating, the better the efficiency of the system. The more active units there are, the larger is the indoor heat exchanger area, which should increase the efficiency. There also appears to be a more significant drop in efficiency below the 0.4 fraction of indoor rated capacity, which may correlate to the minimum compressor operating speed. Below this point, the compressor may not be able to reduce its speed any further, and the use of a hot gas bypass may be required for the outdoor unit to match the required capacity of the indoor units. The reduced efficiency might also be due to heat that was delivered to inactive units and therefore was not measured in this study. It was observed, through elevated supply and return air temperatures, that hot refrigerant was still passing through the indoor coils of units that were not actively heating and did not have their indoor fans running. This would provide some amount of heat to the room via natural convection and radiation. The amount of heat probably would not be large, but since most of the indoor units were not active most of the time, it could add up to significant unmeasured, albeit unrequested, heating capacity. These factors could explain the lower than expected COPs seen in the VRF system.

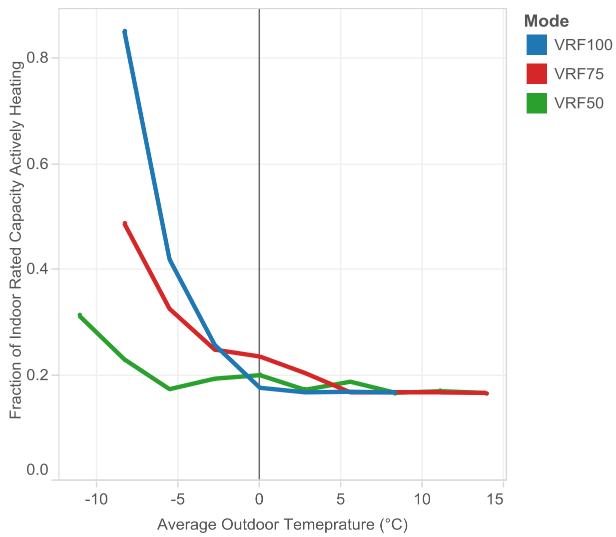



**Figure 23. Fraction of indoor rated capacity of VRF system actively heating relative to outdoor temperature for all operating schedules.**

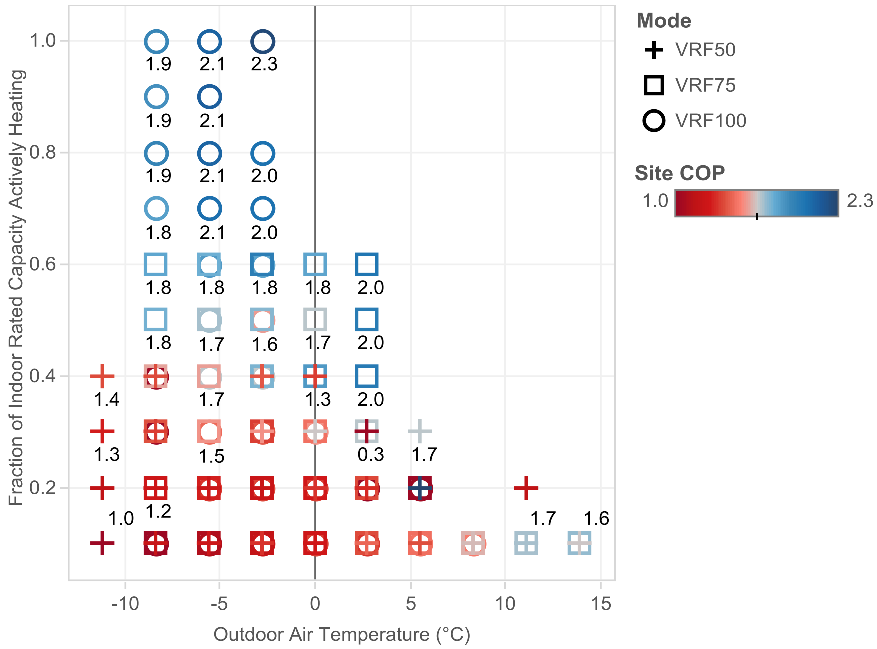



**Figure 24. Site COPs of VRF system relative to outdoor temperatures and fraction of indoor rated capacity actively heating for all operating schedules.**

# 4. CONCLUSION AND LESSONS LEARNED

## 4.1 SUMMARY AND CONCLUSION

This study compares the full- and part-load performance of a VRF system with the baseline RTU/VAV system in a two-story, 300 $m^2$ multi-zone building with emulated office occupancy. To accomplish this, full and part-load conditions (i.e., 100, 75, and 50% loads) in the building were maintained alternately by conditioning either the entire building or selected zones and emulating the occupancy accordingly. During the study period, each system was operated alternately under each of the three load conditions for 2–3 days, and the system parameters, indoor and outdoor conditions, loads, and energy use were monitored. The heating season performance and energy use of both systems was monitored in the winter of 2016. The system performance was evaluated in terms of weather-normalized HVAC energy consumption, the ability to maintain the desired indoor temperatures in the conditioned zones, and the seasonal average COP. Furthermore, the energy savings potential of using VRF systems in major US cities was evaluated using hourly building energy simulations calibrated with data from the measured data in the building. The following are the key findings and lessons learned from this case study.

**Heating Season Analysis**

- In general, the RTU system provided better thermal control for most rooms during heating season since the RTU system can provide simultaneous cooling and heating for different rooms. The analysis shows that the VRF system overheated some rooms. As the installed VRF system cannot provide simultaneous cooling and heating, the majority of the rooms were overheated when they might need cooling.

- Multi-variable regression models were developed for the RTU and VRF hourly energy use. The energy savings for the VRF system compared with the RTU system for the heating season are estimated to be 51, 47, and 27% under the 100, 75, and 50% load conditions, respectively.

- During the heating season, the COPs of the VRF system ranged from 1.2 to 2.0, which were lower than expected but still substantially higher than the COPS of the RTU system. The RTU system COPs were less than 1, which was expected because the system's two sources of heating are electrical resistance heat in the VAV boxes, with a COP of 1, and natural gas heating, with a COP of 0.8.

## 4.2 LESSONS LEARNED

- During the heating season tests, it was found that the DOAS was turned off when the outdoor temperature was below 5°C and the system was operating in ventilation mode, or when the outdoor temperature was below −5°C and the system was operating in heating mode. The current DOAS system cannot provide conditioned air to the space below −5°C OAT, and this is a potential issue for cold climates in the United States. It might be worthwhile to consider coupling the HVAC system with energy recovery ventilation/heat recovery ventilation so that unconditioned cold air could be preconditioned before entering the DOAS systems.

- Although the discharge set point temperature for the DOAS system in heating mode was 21°C, which is the same temperature as the room thermostat set point, the measured DOAS discharge temperature went up to 42°C. Since the DOAS system provides warmer air than was originally planned, sometimes rooms were overheated. Currently, the DOAS unit is not controlled based on room conditions, but only based on heating/cooling mode and OAT. Hence, more integrated control for the

DOAS should be considered to respond to room conditions. In addition, in heating mode, the DOAS should be able to provide air to the rooms at a neutral temperature (i.e., room set point temperature).

- The target building, the FRP, is a small office building with a high percentage of glazing on all four wall orientations and high internal heat gains from emulated occupancy. With this setup, there were many times when south-, east-, and west-facing rooms needed cooling, even during heating season, while other rooms still needed heating. As the currently installed VRF system is a heat pump type of system, it cannot provide simultaneous heating and cooling; and we observed that some room temperatures went over 30°C. Therefore, in this type of building (i.e., with many windows and high internal heat gain), a heat recovery type of VRF system is strongly recommended.

# REFERENCES

ASHRAE. 2013. ANSI/ASHRAE Standard 62.1-2013. Ventilation for Acceptable Indoor Air Quality, American Society of Heating, Refrigeration and Air-Conditioning Engineers, Atlanta.

ASHRAE. 2013. ANSI/ASHRAE Standard 55-2013. Thermal Environmental Conditions for Human Occupancy, American Society of Heating, Refrigeration and Air-Conditioning Engineers, Atlanta.

Im, P., J. Munk, A. Gehl 2015. Evaluation of Variable Refrigerant Flow Systems Performance and the Enhanced Control Algorithm on Oak Ridge National Laboratory's Flexible Research Platform, ORNL/TM-2015/225

# APPENDIX A. VRF System Specification

| Location | | Model name | Cooling capacity (Btu/h) | Heating capacity (Btu/h) |
|---|---|---|---|---|
| Ground (outside) | | AM144FXVAFH/AA | 144,000 | 162,000 |
| DOAS | | AM140HNEPCH/MG | 47,800 | 30,400 |
| Second floor | Rm 202 | AM012FNNDVCH/AA | 12,000 | 13,500 |
| | Rm 203 | AM009FNIDCH/AA | 7,500 | 8,500 |
| | Rm 204 | AM018FN4DCH/AA | 18,000 | 20,000 |
| | Rm 205 | AM018FN4DCH/AA | 18,000 | 20,000 |
| | Rm 206 | AM018FN4DCH/AA | 18,000 | 20,000 |
| First floor | Rm 102 | AM009FNIDCH/AA | 7,500 | 8,500 |
| | Rm 103 | AM009FNIDCH/AA | 7,500 | 8,500 |
| | Rm 104 | AM012FNNDVCH/AA | 12,000 | 13,500 |
| | Rm 105 | AM018FN4DCH/AA | 18,000 | 20,000 |
| | Rm 106 | AM018FN4DCH/AA | 18,000 | 20,000 |