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Cooling Season Full and Part Load Performance Evaluation of Variable Refrigerant Flow (VRF) System Using an Occupancy Simulated Research Building

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ABSTRACT

Variable refrigerant flow (VRF) systems are touted for their superior part-load performance compared to conventional HVAC systems. An on-going study evaluates the full and part-load performance of a VRF system compared with a conventional roof top unit (RTU) variable air volume (VAV) system in a 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 are maintained alternately by conditioning either the entire building or selected zones, and emulating the occupancy accordingly. To compare the performance of the two systems, each system is operated alternately for a weeklong period during which each of the three load conditions are maintained for 2-3 days, and the system parameters, indoor and outdoor conditions, loads, and energy use are monitored.

This paper presents the cooling season performance and energy use of both systems based on the monitored data during the summer of 2015. System performance is compared in terms of weather-normalized HVAC energy consumption and seasonal average COP. In addition, the ability of each system to maintain the indoor temperature in the conditioned zones is also evaluated. Based on the analysis, the energy savings for the VRF system compared with the RTU system for the cooling season are estimated to be 29%, 36%, and 46% under the 100%, 75%, and 50% load conditions, respectively. The average cooling COP was 4.2, 3.9, and 3.7 for the VRF system and 3.1, 3.0, 2.5 for the RTU system under the 100%, 75% and 50% load conditions, respectively. Both systems maintained the indoor temperature very well. However, the VRF system maintained the indoor temperature in a slightly tighter range compared to the RTU system.

1. INTRODUCTION

Variable refrigerant flow (VRF) heat pumps have been used in many countries in Europe and Asia for more than 30 years since it was invented in Japan in 1982 (Thornton 2012). Although the VRF market in U.S. is growing fast, they are still relatively new to the U.S. market. There are known benefits of VRF systems such as easier modular installation, space efficiency, responsiveness to fluctuating loads, and higher efficiency, but several studies (Goetzler 2007, Amarnath & Blatt 2008, Aynur et al 2009) show that there are concerns regarding the application of VRF systems in the U.S, including 1) lack of awareness of energy efficiency advantages, 2) higher first cost, 3) lack of understanding about the suitability of the VRF system for the building operation profile in the US. The performance of VRF systems in the US has been measured mainly in laboratories, which cannot represent its performance in a real building. On the other hand, performance measurement of VRF systems in real buildings is challenging due to the

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complexity of the system and occupancy related uncertainties. Addressing these concerns, a VRF system was installed in June 2014 in Oak Ridge National Laboratory's two-story Flexible Research Platform (FRP) – a multi-zone office building with simulated occupancy, to conduct a multi-year research to evaluate its performance in different operation modes (Im et al. 2015). During the alternate weeks of the entire study period, an existing RTU VAV system is also operated in the FRP to serve as the baseline system. In June 2015, the building and system operation was set to evaluate the part-load performance of the VRF system. To accomplish this, full and part-load conditions (i.e., 100%, 75% and 50% loads) in the building are maintained alternately by conditioning either the entire building or selected zones, and emulating the occupancy, accordingly. During the study period, each system is 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 are monitored. This paper discusses the cooling season energy use, thermal comfort and COP analysis of the VRF system as compared with an RTU system.

2. METHODOLGY

2.1 Test Facility

The test facility used for this study 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 1a). The occupancy in the building can be simulated by process control of lighting and other internal loads. On 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 system was installed (Figure 1b and c) and the existing RTU system served as the baseline system. Table 1 summarizes the baseline building and system characteristics.

Figure 1: Test facility (1a-left), VRF system's outdoor unit (1b-center) and indoor unit (1c-right)

Table 1**:** Building Characteristics

Location	Oak Ridge, Tennessee, USA
Building size	Two-story, 40 ft x 40 ft (12.2 m x 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 rooftop unit (RTU) provides DX cooling, heating with a natural gas furnace, and electric resistance reheat at VAV terminal units. The return air is drawn from each room through a plenum on each floor. Fresh air is introduced through the fan of the DOAS system 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. The RTU is programmed to maintain a constant discharge air temperature at 14ºC (57ºF). The natural gas furnace would engage if the building mixed air temperature dropped below 14ºC (i.e., during the winter). In the cooling mode (i.e., at discharge temperature at 14ºC), the electric resistance reheat in the VAV terminal boxes in individual zones activate to provide necessary heat to maintain the desired zone temperature.
The VRF system has a 12-ton (42 kW) outdoor unit and 10 indoor units with capacities ranging from half to 1.5 tons (1.8-5.3 kW). The 10 indoor units and the DOAS systems are connected to the same VRF outdoor condensing unit, and the DOAS system provides conditioned outdoor air to 10 zones. Note that the VRF system in this study is a heat pump type system which only provides cooling or heating at any single time, and cannot provide simultaneous heating and cooling for different thermal zones.

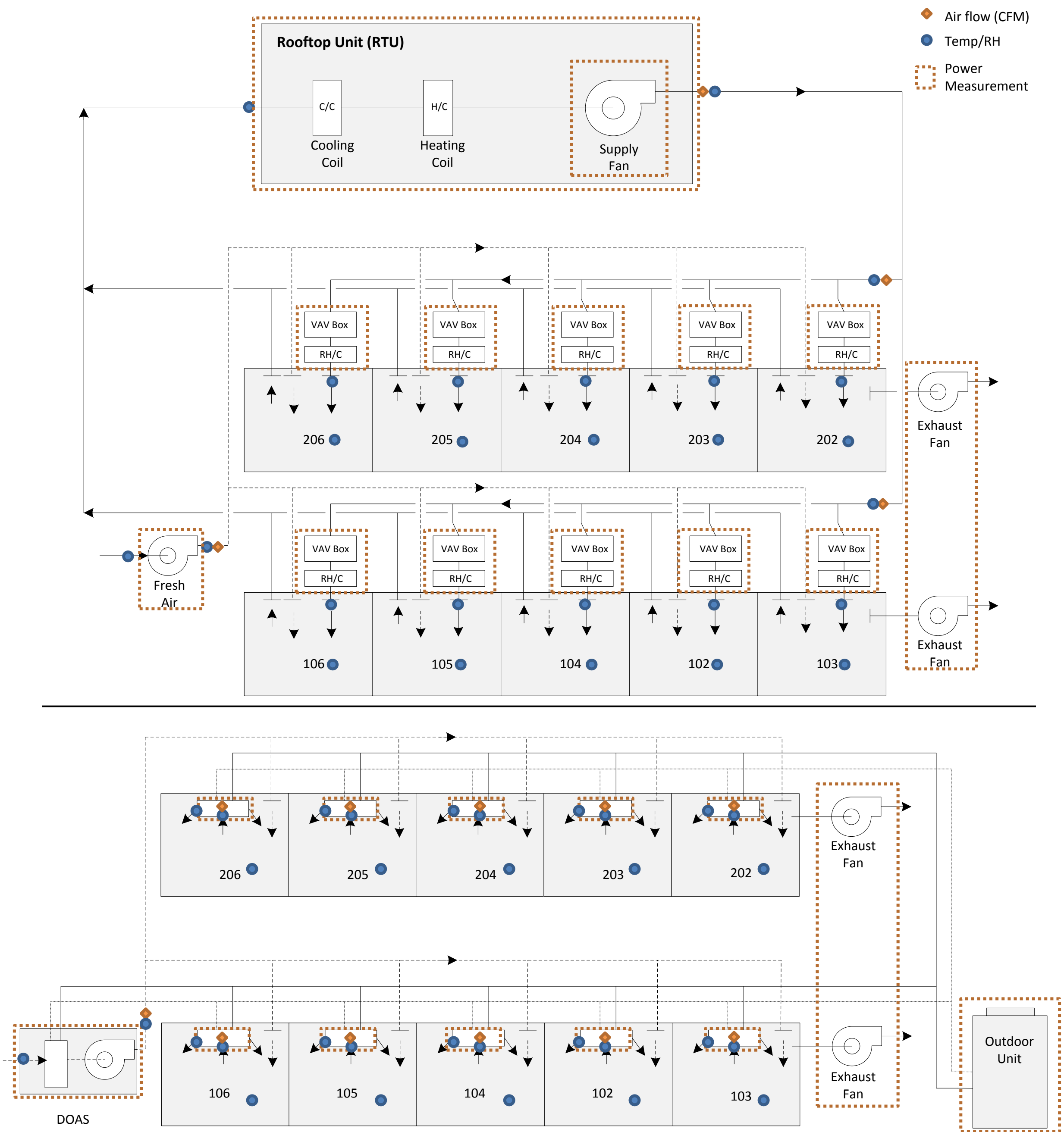

Figure 2: System schematic and monitoring points for RTU system (above) and VRF system (below)

2.3 Building Operation

The building's HVAC and occupancy emulating systems were operated to simulate occupied hours from 6 am to 6 pm. The thermostat cooling setpoint was maintained at 24ºC (75.2ºF) during the occupied hours and 30ºC (86ºF) during the unoccupied hours. The full and part-load conditions in the building were maintained alternately by conditioning the selected zones, and emulating the occupancy in the conditioned zones, accordingly. Figure 3 marks the conditioned zones in the building to emulate (a) 50% load, (b) 75% load and (c) 100% load‡. During the study period, each system was operated alternately for a weeklong period during which each of the three load conditions were maintained for 2 to 3 days, and the system parameters, indoor and outdoor conditions, loads, and energy use were monitored.

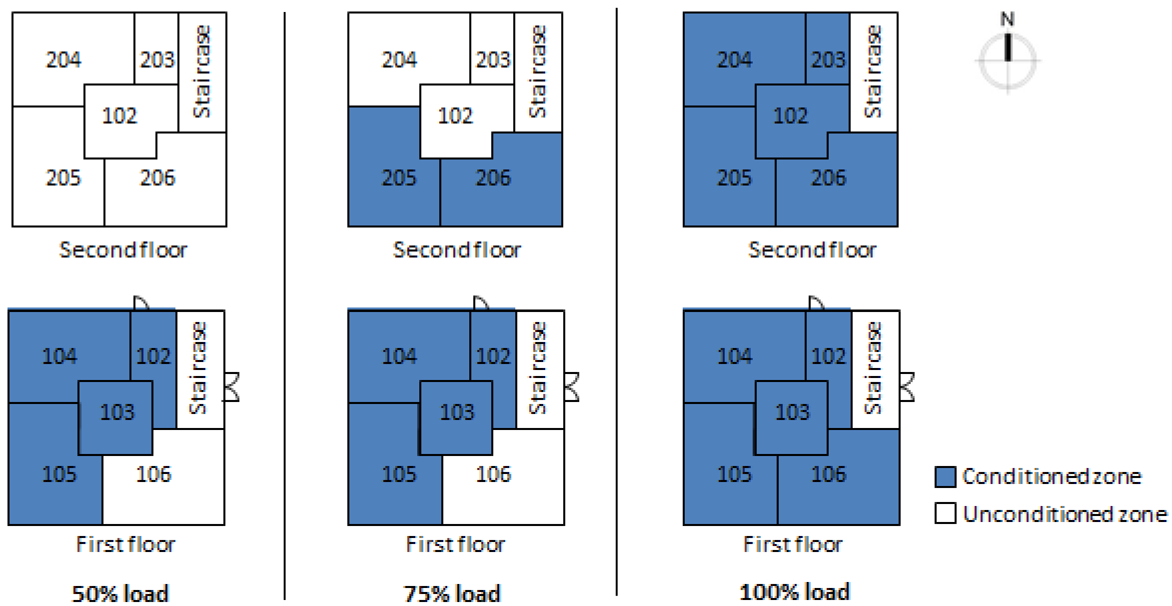

Figure 3: Zone conditions to emulate (a) 50% load, (b) 75% load and (c) 100% load

2.4 Data Monitoring

The performance of the RTU and VRF systems were evaluated using refrigerant and air-side measurements at 30-second intervals. Figure 2 shows the air-side monitoring points, which include the room temperature and relative humidity, supply, return, and mixed-air temperature and relative humidity. Airflow rate was measured at the supply and return side and fresh air supply for the RTU system, and at each indoor unit and DOAS supply side for the VRF system. Power measurements were obtained separately for the RTU unit, supply fan, and DOAS fan for the baseline RTU system. Power consumption for the VRF outdoor unit, each VRF indoor unit, and DOAS system were measured as well.

3. RESULTS

The analysis is based on the measured data from July 11, 2015 through September 12, 2015. During this period, the RTU and VRF systems were operated under 50%, 75% and 100% load conditions alternately, as described above. The performance of RTU and VRF systems were compared in terms of (a) energy use, (b) the ability to maintain room temperature, and (c) system efficiency. The energy use and thermal performance comparison were performed using measured hourly data for occupied hours only, excluding the startup hours (i.e., 8 am to 6 pm). The COP analysis was performed using both hourly and 1-minute data.

3.1 Thermal Condition Analysis

‡ The operation scenario was named 50% and 75% loads based on the combined rated capacity of indoor units.

Figure 4 shows the measured hourly room temperature statistics (minimum, first quartile, median, third quartile, and maximum) for 8 am to 6 pm in all rooms of the building during RTU and VRF system operation at 100%, 75% and 50% load conditions. The outdoor air temperature statistics are also plotted. The thermostat cooling set point (i.e., 24ºC or 75.2ºF) is marked as a blue line across the plots. Temperature statistics in unconditioned rooms are marked as gray bars. The following observations can be made:

- Both RTU and VRF systems maintained room temperature very well, especially in the first floor rooms. The VRF system maintained room temperature in a slightly tighter range compared to the RTU system.
- The second floor rooms were generally slightly overheated.
- 5 to 6 rooms were overcooled during the RTU 100% operation, and two rooms were overcooled during VRF 100% operation.

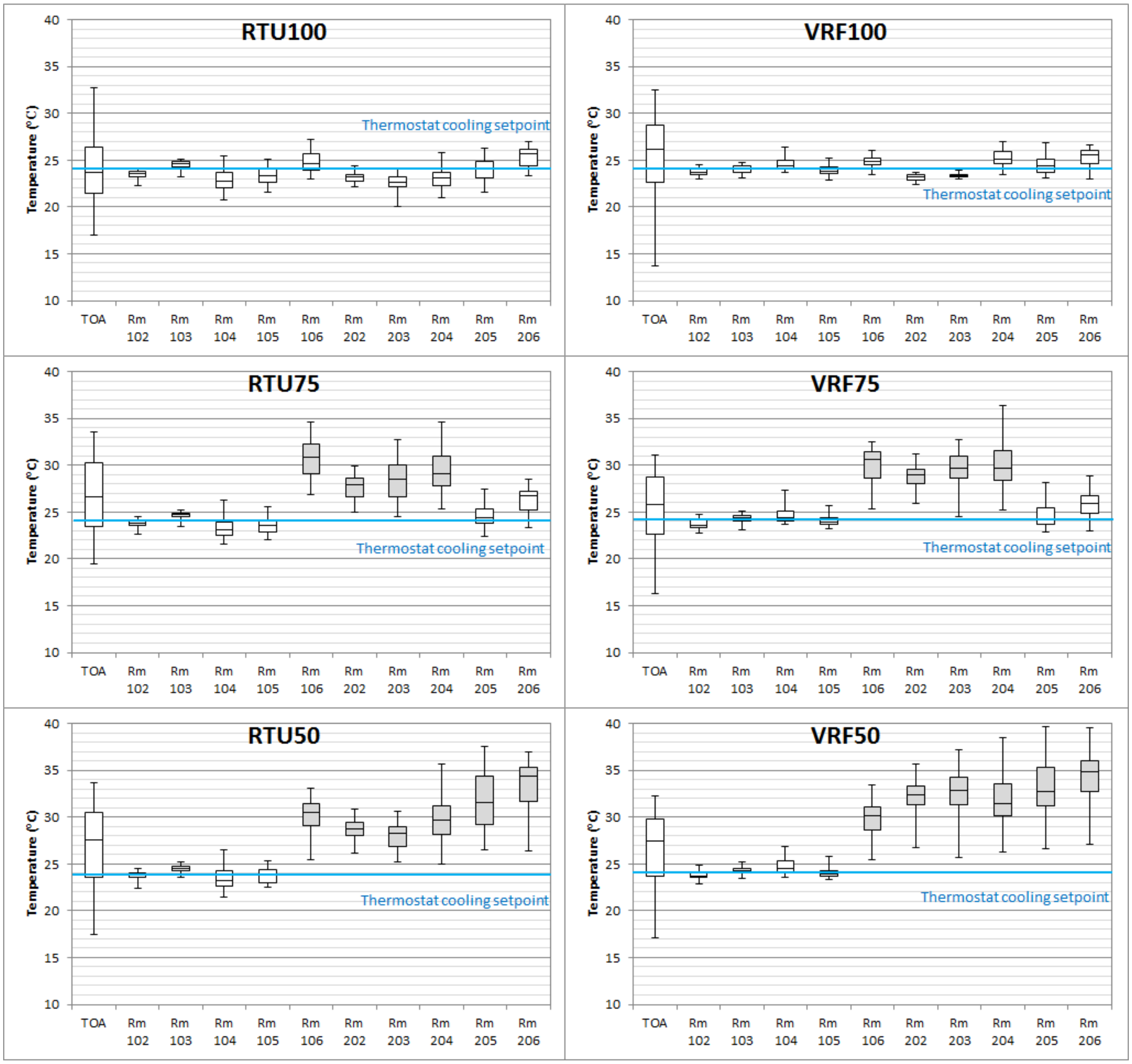

Figure 4: Room temperature during RTU and VRF system operation at 100% (top), 75% (middle) and 50% (bottom) load conditions

3.2 Power Consumption Analysis

Figure 5 shows the power consumption during occupied hours when the RTU and VRF systems were operated under different loads. In general, VRF power consumption for all three operations were less than the RTU power use in similar conditions. For RTU energy plots, there are many instances that the energy use of 50% load and 75% load are close each other. Comparing this data across the VRF and RTU system by load shows that difference in the energy use between RTU and VRF system was higher at smaller load (i.e., 50% and 75%). This is due to the better isolation of zone air with the VRF system operation versus the mixing of return air from conditioned zones with the warmer air from the unconditioned zones in the plenum during the RTU system operation.

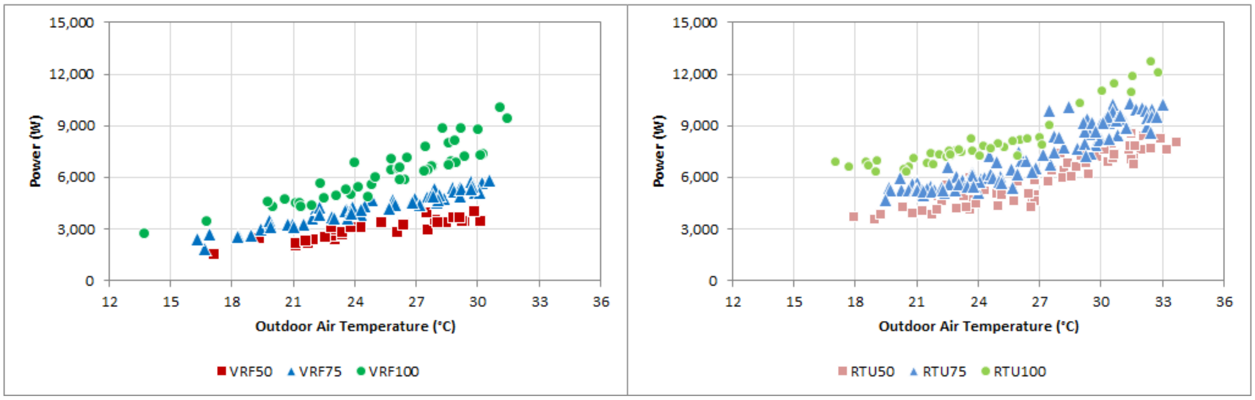

Figure 5: Measured power consumption by VRF system (left) and RTU system (right)

Figure 6a shows a comparison of predicted average daily cooling season energy use between the VRF and RTU systems at different loads. Only weekdays from July through September 2015 and occupied hours (including startup) are accounted for. The predicted energy use is calculated using the weather normalized curve-fit for the measured data shown in Figure 5. The VRF system saved 29%, 36% and 46% cooling energy use at 100%, 75% and 50% loads, respectively.

Figure 6b shows the energy savings from VRF system at different load conditions for a range of outdoor air temperature observed during the monitoring period. As expected, the savings from VRF system were higher for partially conditioned building. At outdoor temperature above 28ºC, the savings from VRF system remained slightly above 20% and were higher at less warm temperature. Savings from VRF system at 50% load was consistently higher than that at 75% load. At these part loads, savings were slightly above 30% and 40% at outdoor air temperature between 70F and 80F. The savings rise gradually beyond this temperature range.

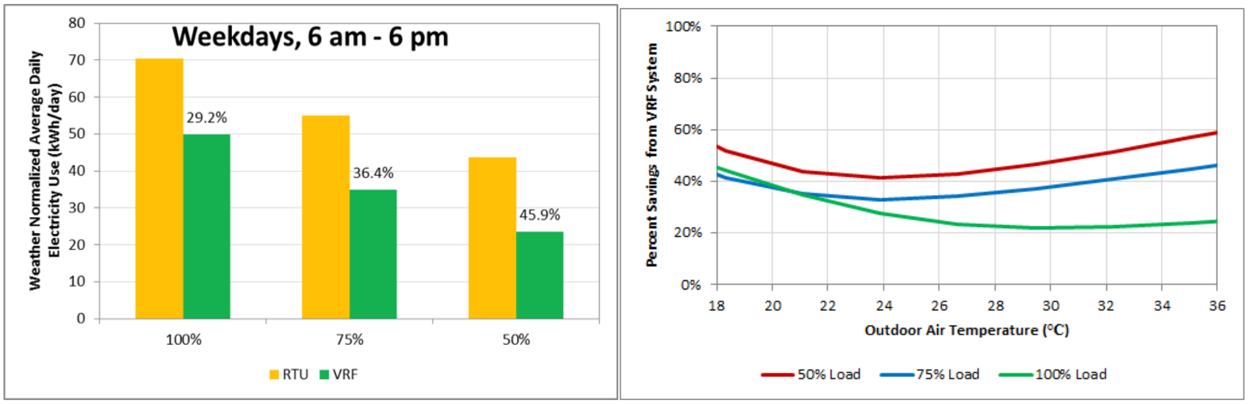

Figure 6: Estimated cooling season energy use analysis: (a) total energy use, and (b) energy savings from VRF system

3.3 COP (Coefficient of Performance) Analysis

Figure 7 shows the COP of VRF and RTU systems with respect to outdoor air temperature. As discussed above, the operating efficiency of the VRF system is more sensitive to outdoor temperature driven loads (i.e., lower COP at cooler temperatures). At lower outdoor air temperature, the impact of zone-control driven variability in loads is observed. At higher temperature, this impact diminishes. The efficiency of the RTU system shows similar trend (lower COP at 50% load, even lower at cooler temperature) but the impact is less pronounced. Over the data monitoring period, it shows that the average cooling COP was 4.2, 3.9, and 3.7 for the VRF system and 3.1, 3.0, 2.5 for the RTU system under the 100%, 75% and 50% load conditions, respectively.

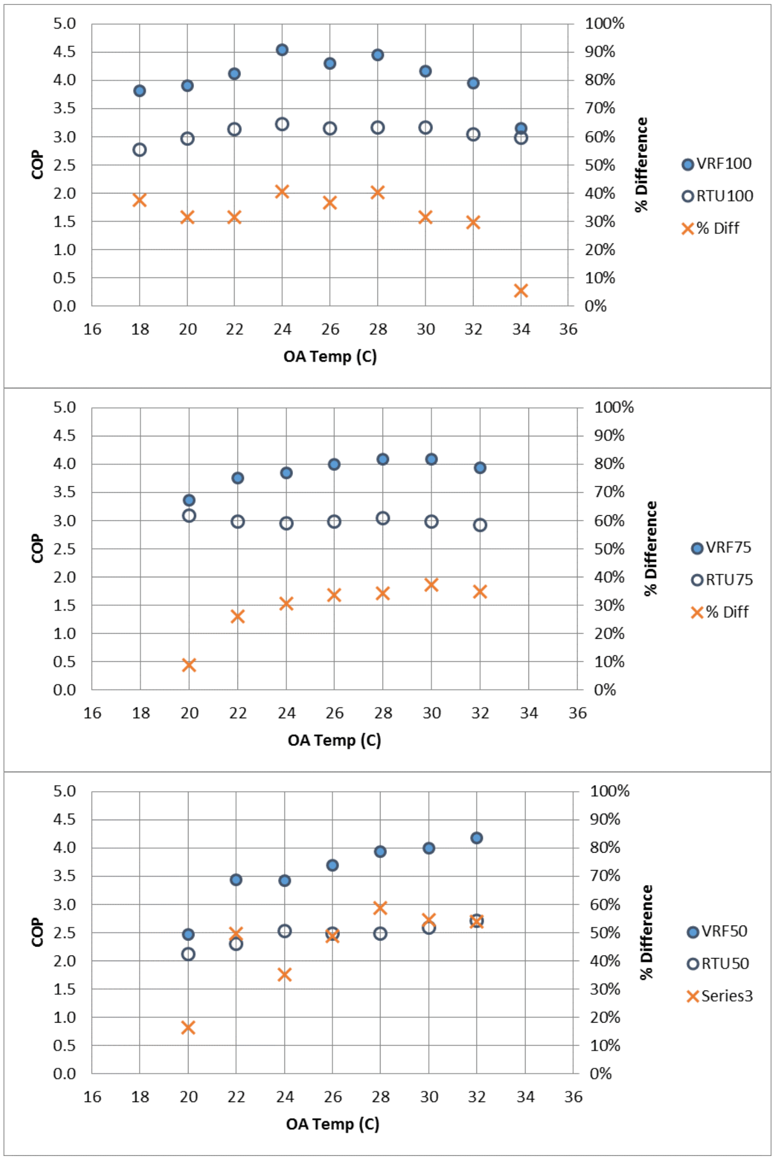

Figure 7: COP comparison between RTU and VRF Systems (top: 100% operation; middle: 75% operation; bottom: 50% operation)

3.4 Quasi-Steady-State COP (Coefficient of Performance) Analysis

Analyzing high resolution, 1 minute interval data allows further insight into the equipment operation by enabling the filtering of near steady-state data and for the separation of the data by operating stage or capacity. Based on analysis of how the equipment COP varies with the runtime during a cycle, 6 minutes was chosen as the cutoff for the minimum amount of runtime in a particular state that would be included in a quasi-steady-state data set.

Using the quasi-steady-state data set, the COP of the RTU was plotted against outdoor temperature in Figure 8 The outdoor air temperature was binned in 2.8°C (5°F) increments, e.g. 25°C (77°F) bin includes temperatures greater than or equal to 23.9°C (75°F) and less than 26.7°C (80°F). The data was also divided by unit stage with low stage representing single compressor operation and high stage representing dual compressor operation. As expected, the efficiency of the system decreases with increasing ambient temperature. The expected trend of higher efficiency during low stage operation is not consistently seen in the different operating schedules. This can be explained by looking at both the outdoor and indoor airflow. The outdoor airflow, as indicated by fan power, did not change between high and low stage. It is likely that the outdoor fan is running at a higher than-optimal speed in low stage resulting in excess fan power and reduced efficiency. The indoor blower on the RTU varies its speed in order to maintain a constant static pressure in the duct system, so the number VAV boxes that are open determines the airflow. This results in the RTU100 schedule having the highest indoor airflow and the RTU50 schedule having the lowest. The indoor airflow is independent of the compressor staging though, resulting in a wide range in the ratio of indoor airflow to cooling capacity from ~5.7-14.2 m^3/min (~200-500 scfm) per ton of cooling. Generally, the optimal indoor fan speed for efficiency falls somewhere in the middle of this range, with higher airflows resulting in excessive fan power and lower airflows resulting in lower suction saturation temperatures and therefore higher compressor power. The RTU75 low stage operation and RTU100 high stage operation fell closest to the 9.9 m^3/min (350 scfm) per ton of cooling airflow and exhibit the highest efficiencies of their respective stages. The best efficiency is achieved at an optimal balance between capacity, airflow, and fan power.

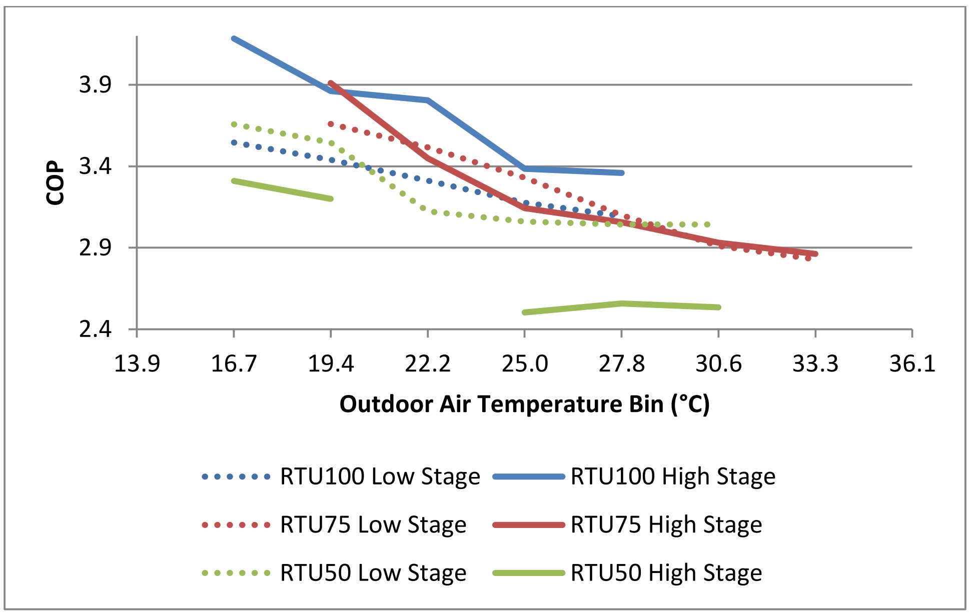

Figure 8: Comparison of RTU efficiency during different schedules

The VRF system is a much more complicated system having multiple variables impacting the performance including compressor speed, number of indoor units actively cooling (effective evaporator size), and outdoor temperature. Based on the earlier COP analysis, it is evident that the VRF system performance is not following the typical trend with outdoor ambient temperature of increasing efficiency with decreasing outdoor ambient temperature. Figure 9 shows a plot of the system COP as a function of outdoor air temperature and the ratio of the rated capacity of the actively cooling indoor units to the total rated indoor unit capacity. The data from the different schedules are shown with different style markers, while the color of the marker and the label indicate the average COP for the data. Efficiency is generally good when more than 30% of the indoor unit capacity is actively cooling. However, there is a significant drop in efficiency below this level likely corresponding to an imbalance in system capacity and load requiring the use of the hot gas bypass. This is reflected in generally higher COPs for the VRF100 schedule when all units are available for cooling and lower performance for the schedules with several indoor units turned off (i.e., VRF50 & VRF75).

To examine the impact of compressor speed on efficiency, the data was filtered to only include times when the rated cooling capacity of the active indoor units was at least 80% of the total rated indoor capacity. This limits the data to times when nearly all of the indoor units were providing cooling. The measured cooling capacity was then divided by the total rated cooling capacity and this data was placed into 0.1 increment bins. With all else held constant, an increase in compressor speed will increase the cooling capacity, so the ratio of the measured cooling capacity to the rated cooling capacity is used as a proxy to represent compressor speed. The efficiency of this binned data was plotted against the binned outdoor air temperature data as seen in Figure 10. This data shows that, in general, with a fixed evaporator size the VRF system is more efficient when operating at lower compressor speeds, indicated by lower capacity in this case.

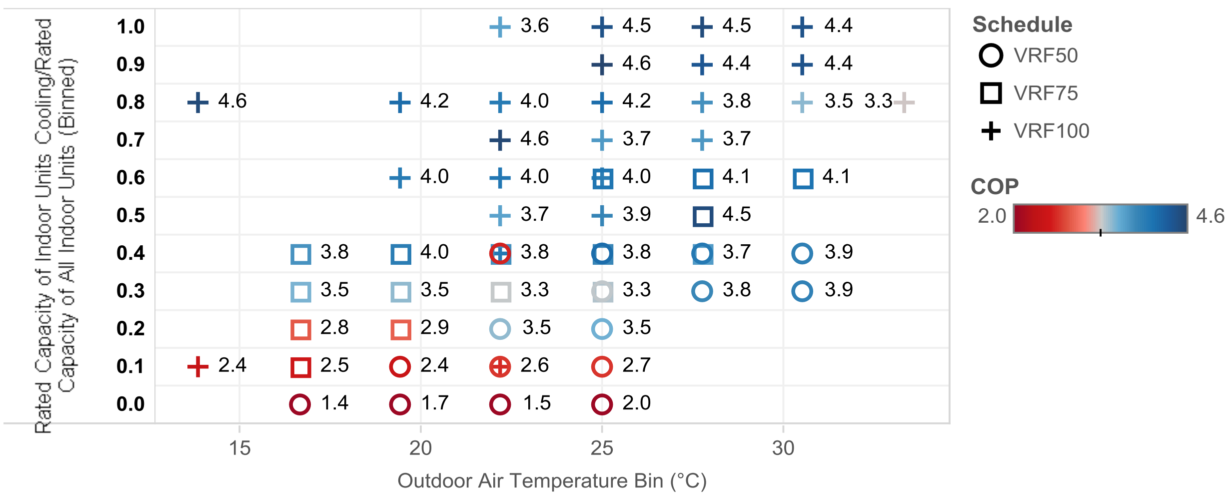

Figure 9: Impact of the rated capacity of active indoor units on efficiency for the VRF system

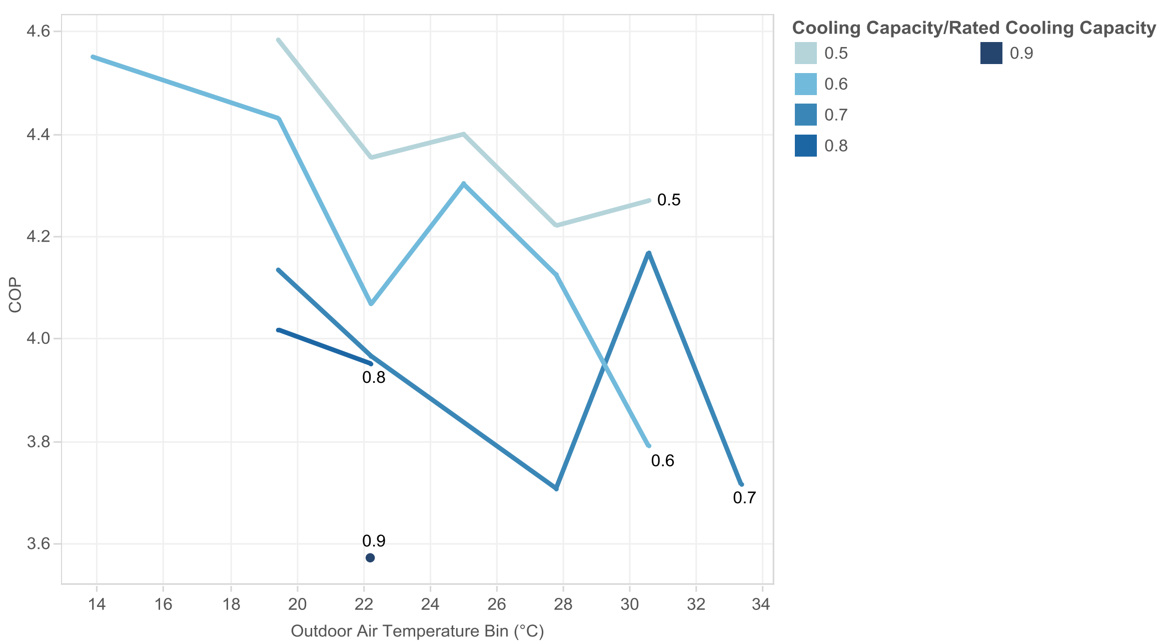

Figure 10: Efficiency of VRF system when most indoor units are providing cooling.

4. CONCLUSION

This paper presented the full and part-load performance of a VRF system compared with a baseline conventional RTU system in a two-story, 300 sqm 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 are maintained alternately by conditioning either the entire building or selected zones, and emulating the occupancy, accordingly. During the study period, each system is 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 are monitored. The cooling season performance and energy use of both systems was monitored during the summer of 2015. The system performance was evaluated in terms of weather-normalized HVAC energy consumption, the ability to maintain desired indoor temperature in the conditioned zones, and seasonal average COP.

- Hourly zone temperature analysis shows that both RTU and VRF systems maintained room temperature very well, especially in the first floor rooms. The second floor rooms were generally slightly overheated. The VRF system maintained room temperature in a slightly tighter range compared to the RTU system.
- The energy savings for the VRF system compared with the RTU system for the cooling season are estimated to be 29%, 36%, and 46% under the 100%, 75%, and 50% load conditions, respectively.
- The COP analysis based on 1 hour data shows that the average cooling COP was 4.2, 3.9, and 3.7 for the VRF system and 3.1, 3.0, 2.5 for the RTU system under the 100%, 75% and 50% load conditions.
- The quasi-steady-state data analysis indicates that the VRF system operates most efficiently when all indoor units are actively cooling and the compressor is operating at reduced speed

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