Evaluation of Energy Saving Potential of Variable Refrigerant Flow (VRF) Systems Compared with Variable Air Volume (VAV) Systems in the U.S. Climate Locations

Dongsu Kim[1], Sam J. Cox[1], Heejin Cho[1], and Piljae Im[2,*]

[1]Mechanical Engineering Department, Mississippi State University, MS 39762, USA
[2]Building Technologies Research and Integration Center, Oak Ridge National Laboratory, TN 37831, USA

ABSTRACT

Variable refrigerant flow (VRF) systems are known for their high energy performance and thus are able to reduce cooling and heating energy both in residential and commercial buildings. The energy savings potential of this system has been demonstrated in several studies by comparing the system performance with conventional Heating Ventilation and Air Conditioning (HVAC) systems such as rooftop variable air volume systems and central chiller and boiler systems. This paper evaluates the performance of VRF and rooftop VAV (RTU-VAV) with reheat systems in a simulation environment using widely accepted whole building energy modeling software, EnergyPlus. A medium office reference building model, developed by the U.S. Department of Energy (DOE), is used to assess the performance of VRF and RTU-VAV systems. Each system is placed in 16 different locations, representing all U.S. climate zones, to evaluate the performance variations. Both models are compliant with the minimum energy code requirements prescribed in American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) standard 90.1-2010 - Energy standard for buildings except low-rise residential buildings. Finally, a comparison study between the simulation results of VRF and RTU-VAV models is made to demonstrate energy savings potential of VRF systems in the 16 climate locations. The simulation results show that the VRF system would save 25% to 55% HVAC energy use compared to the RTU-VAV system in the 16 climate locations.

KEYWORDS

Variable refrigerant flow, Rooftop unit, Variable air volume, Building simulation, U.S. climate zones

---

[1] This manuscript has been authored by UT-Battelle, LLC under Contract No. DE-AC05-00OR22725 with the U.S. Department of Energy. The United States Government retains and the publisher, by accepting the article for publication, acknowledges that the United States Government retains a non-exclusive, paid-up, irrevocable, world-wide license to publish or reproduce the published form of this manuscript, or allow others to do so, for United States Government purposes. The Department of Energy will provide public access to these results of federally sponsored research in accordance with the DOE Public Access Plan (http://energy.gov/downloads/doe-public-access-plan).

* Corresponding author email: imp1@ornl.gov

INTRODUCTION

Residential and commercial buildings represent approximately 35% of the energy consumption in the U.S. (EIA 2015), with the HVAC systems consuming around one third to one half of the total building energy consumption (Goetzler et al. 2014). Variable air volume (VAV) systems have been widely used in commercial buildings since being introduced to the U.S. market in the 1970s (Aynur et al. 2009b). VAV systems have several advantages over other HVAC systems, namely less fan capacity, greater flexibility with respect to varying loads, and better indoor environment (Murphy 2011). Current studies have also showed the energy savings potential and thermal comfort of VAV systems by comparing other systems and development of control strategies in VAV systems (Aynur et al. 2009b). Although there have been various efforts to improve the performance of VAV systems over the years, VAV systems still tend to have issues with control systems, such as dampers, supply air temperature, and ventilation of multiple zones in commercial buildings, because of their complex structures and configurations (Okochi & Yao 2016).

Variable refrigerant flow (VRF) systems have been popular in many Asian and European countries with numerous benefits, including: ease of installation, design flexibility, maintenance, and energy efficiency (Amarnath & Blatt 2008). As a relatively new HVAC technology in the U.S., VRF systems have been gaining more attention in comparison to conventional HVAC systems, e.g. constant air volume and VAV systems (Aynur et al. 2009a). A comparative study showed that VRF systems could consume about 11% to 22% less energy use than fan-coil units and VAV systems (Zhou et al. 2007), about 35% less energy use than central chiller/boiler systems (Goetzler 2007), and around 20% energy savings over rooftop (RTU) during cooling season (Im & Munk 2015). However, several concerns for the adoption of VRF system in the U.S. still exist such as lack of awareness for their energy savings potential, higher initial cost, and lack of optimized and integrated VRF control system (Aynur 2010).

Therefore, this study aims to evaluate the performance of VRF and RTU-VAV systems in a simulation environment using a whole building energy modeling software, EnergyPlus, under different weather conditions and to evaluate the energy savings potential of VRF systems over RTU-VAV systems in 16 U.S. climate locations that are compliant with the minimum energy code requirements prescribed in ASHRAE standard 90.1-2010.

SIMULATION MODEL

EnergyPlus 8.4 currently provides VRF system and single packaged RTUs with VAV reheat modeling capability (DOE 2015a). EnergyPlus prototype medium office models shown in Figure 1, developed by Pacific Northwest National Laboratory (PNNL) (Thornton et al. 2011), are used to assess the performance of the VRF and the RTU-VAV systems. For this study, the original prototype medium office models developed in version 8.0 of EnergyPlus were converted into version 8.4 to implement supplementary heaters with VRF systems. Each building model, in 16 different climate zones, complies with the minimum energy code requirements prescribed in the ASHRAE Standard 90.1-2010 (ASHRAE 2010).

Table 1 summarizes the construction details of the prototype medium office. The prototype medium office has a rectangular floor plan and total floor area of 4,982 m^2.

Each floor has 5 thermal zones, including 4 perimeter zones and 1 core zone, and 33% window-to-wall fraction. All construction details and internal heat gains are based on ASHRAE Standard 90.1 requirements and provided in Table 1. In this study, the energy consumption for equipment and lighting power were not compared as these are not changed, and only the HVAC energy uses were compared between two systems.

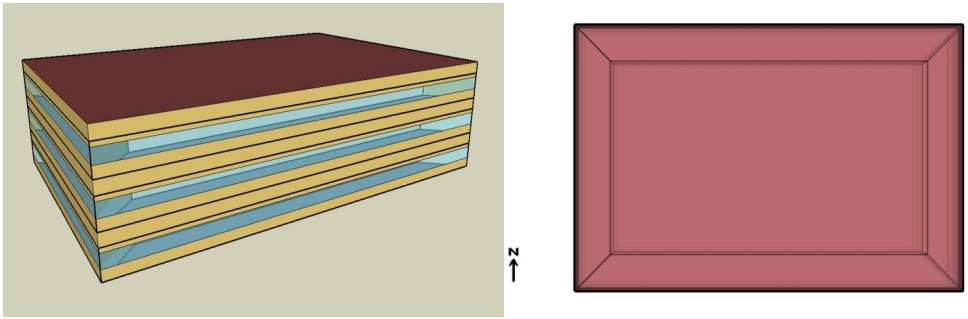

Figure 1. *3D view (left) and a floor plan (right) of a simulated medium Office*

Table 1*. Construction details of the medium office building*

Building component	*Construction detail*	*Reference*
Total floor area	4982.2 m^2 (53,600 ft^2)	-
Number of floors	3	-
Floor to floor height	3.96 m (13 ft.)	-
Floor to ceiling height	2.74 m (1.22m ceiling plenum)	-
Glazing sill height	1 m (3.35 ft.)	-
Thermal zoning	Core 60%, perimeter 40% (perimeter zone depth: 4.57 m (15 ft.)	-
Infiltration	0.00056896 $m^3/s\text{-}m^2$ (0.111999776 $ft^3/min\text{-}ft^2$)	PNNL-18898: Infiltration Modeling Guidelines for Commercial Building Energy Analysis.
Wall construction	Steel-Frame Walls (2X4 16IN OC) 0.085 m. Stucco+0.016 m. gypsum board + wall Insulation+0.016 m.	2003 CBECS Data and PNNL's CBECS Study 2007. ASHRAE 90.1-2010
Floor construction	Construction requirements	ASHRAE 90.1-2010
Roof construction	Roof membrane + roof insulation + metal decking	Construction requirements of ASHRAE 90.1-2010
Window fraction (window-to-wall ratio)	33%	2003 CBECS Data and PNNL's CBECS Study 2007
Windows	Construction requirements	ASHRAE 90.1-2010
Lighting power density	9.687519375 (W/m^2)	ASHRAE 90.1-STD 2010
Daylighting controls	Control requirements	ASHRAE 90.1-STD 2010
Equipment power density	7937.2 for core zones / 1673.2 and 1059.2 for perimeter zones (W)	ASHRAE 90.1-2004
Occupancy	1.8579 (m^2/person)	ASHRAE 90.1-2004

The prototype building model includes a RTU system with VAV electric reheat. This system is an air conditioning system that varies the supply air volume flow rate through an air handing unit (AHU) using a damper located in a VAV terminal box in order to offset heating and cooling load and meet the set-point temperature, (DOE 2015a). Original input values were used for the RTU-VAV system as described in Table 2 (Thornton et al. 2011). The domestic supply hot water system was removed

for the RTU-VAV system. Although economizer is one of requirements for ASHRAE 90.1-2010 and it is originally used for the prototype medium office model, economizers were not included for both RTU and VRF models in this study to have fair comparisons. In the current version of EnegyPlus (version 8.4) an economizer option is not available for the VRF model. Energy management system (EMS) inputs for optimum set-point temperature were also modified to maintain the same set-point temperature for both the RTU-VAV and the VRF systems. Although nighttime set-back temperatures were used for most of cities, some cities in hot and warm climate zones did not use the set-back temperature during the cooling season, depending on outside weather conditions (Thornton et al. 2011).

There are two general types of the VRF heat pump (VRF-HP) and VRF heat recovery (VRF-HR) systems. A VRF-HP system only provide heating or cooling at a given time, while a VRF-HR is capable of operating simultaneously in heating and/or cooling mode (Goetzler 2007). In this study, the VRF-HP system was used, and the VRF-HP system employs a variable speed compressor and electronic expansion valve to independently vary the flow rate to each terminal unit under part-load conditions. For VRF-HP simulation models, the original prototype medium offices, which initially include the VAV system, were modified by replacing the VAV system with a VRF-HP system (DOE 2015b). The VRF indoor units were operated by the zone thermostat for cooling or heating conditions to meet the sensible cooling or heating loads. The multiple VRF indoor units were directly connected to the air-cooled VRF outdoor unit. Three outdoor air-cooled units were created to cover the three floor building, and each zone individually had one indoor unit. Although the VRF system in EnergyPlus can simultaneously operate cooling and heating mode, only one indoor location on each floor was chosen as the master thermostat, which determined the VRF system operating mode based on the thermostat location zone load (DOE 2015a).

Table 2 summarizes the HVAC input details of both the RTU-VAV and the VRF systems. The coefficients of performance curves in the EnergyPlus VRF model (Raustad 2013) were used, and the coefficient of performance (COP) values for the rated cooling and heating were set to 3.39 and 3.55 for the VRF system, respectively. The VRF rated cooling COP value of 3.39 were selected to match the RTU-VAV rated cooling COP, which is the code minimum value from ASHRAE 90.1-2010. The VRF rated heating COP of 3.55 was directly obtained by a previous study (Wang 2014), which is also the EnergyPlus default value. All cooling and heating capacities were auto-sized to design day conditions corresponding with ASHRAE climate zones in both the RTU-VAV and the VRF systems.

For zone outdoor air (OA) requirements based on number of occupants and zone floor area, ventilation optimization was implemented using a control algorithm that is available on “Controller: Mechanical Ventilation (CMV)” object in EnergyPlus for the RTU-VAV models. This input fields in the CMV object included zone minimum OA requirement as determined by ASHRAE Standard 62.1 (Thornton et al. 2011). Although the RTU-VAV models always used OA ventilation optimization controls to

maintain OA requirements during simulation period, the VRF models employ a simple way to bring OA through individual zones using "Outdoor Air Mixer (OAM)" object in EnergyPlus during VRF systems operation (DOE 2015a). The RTU-VAV models complied with ASHRAE Standard 62.1 values provided in *Table 2*. However, the OA ventilation values of VRF models were modified to reflect the actual air flow rate that the RTU-VAV models.

In addition, supplementary electric heaters were added with the VRF simulation models since the VRF system provided by the EnergyPlus simulation has a limit on the heating capacity when the outdoor air (OA) temperature can be lower than -20°C. The supplementary electric heaters can provide additional heating throughout the space by only natural convection, which has an immediate impact on the zone air heat balance. The heating and cooling set-points for the VRF models shown in Table 2 were the same as that of the RTU-VAV models.

***Table 2**. EnergyPlus HVAC model for RTU-VAV versus VRF HP models*

HVAC System component	*RTU-VAV system*	*VRF HP system*
Heating	Gas furnace inside the packaged air conditioning unit	VRF DX heating coil
Cooling	Unitary DX inside the packaged air conditioning unit	VRF DX cooling coil
Distribution and terminal units	VAV terminal box with damper and electric reheating coil (minimum supply air at 30% of the design peak supply air)	Variable refrigerant flow (VRF) DX cooling and heating coils (air-to-air heat pump)
Total Cooling capacity	Auto-sized to design day	Auto-sized to design day
Total Heating capacity	Auto-sized to design day	Auto-sized to design day
Cooling COP	3.39	3.39
Heating COP	0.8 (Heating gas burner efficiency)	3.55
Thermostat set-point (occupied hours)	75°F Cooling / 70°F Heating	75°F Cooling / 70°F Heating
Thermostat set-back (unoccupied hours)	80°F Cooling / 60°F Heating	80°F Cooling / 60°F Heating
Supply Fan type	Fan: Variable volume	Fan: On/Off
Indoor supply fan efficiency (%)	60% to 62% depending on the fan motor size	0.7
Maximum Outdoor Ventilation Air through OA Mixer	0.000598 m3/s-m^2 (0.118 cfm/ft^2)	0.000598 m3/s-m^2 (0.118 cfm/ft^2)
Supplementary heater type	-	Zonal baseboard convective elec. Heater (natural convection unit)
Supplementary heater heating capacity	-	Auto-sized to design day
Supplementary heater efficiency	-	0.97

IDENTIFY CLIMATE LOCATIONS

Building models with each system are placed in 16 different cities representing all U.S. climate zones to evaluate the performance variations in different weather conditions. The 16 climate zones constructed by the International Energy

Conservation Code (IECC) and ASHRAE were used for this study to study the energy saving potential of the VRF system from the RTU-VAV system. Table 3 lists the 16 representative locations identified in ASHREA Standard 90.1-2010 in the U.S. climate zones.

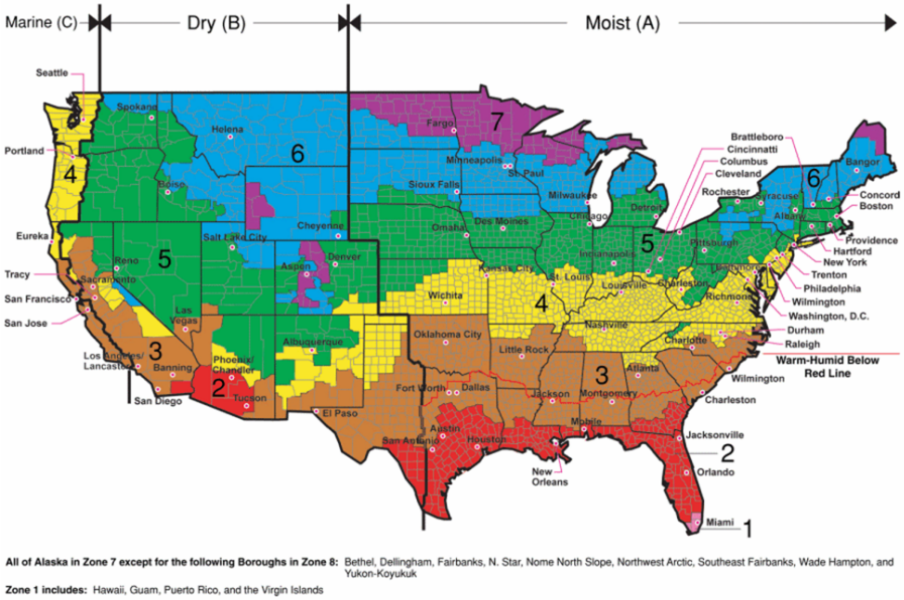

Figure 2. *ASHRAE climate zones in the United States. (ASHRAE 2013)*

Table 3. *16 cities representing the climate zones*

Climate zone	*16 Representative cities*
1A	Miami, Florida (very hot, humid)
1B	Houston, Texas (hot, humid)
2A	Phoenix, Arizona (hot, dry)
2B	Atlanta, Georgia (warm, humid)
3A	Los Angeles, California (warm, dry)
3B	Las Vegas, Nevada (warm, dry)
3C	San Francisco, California (warm, marine)
4A	Baltimore, Maryland (mixed, humid)
4B	Albuquerque, New Mexico (mixed, dry)
4C	Seattle, Washington (mixed, marine)
5A	Chicago, Illinois (cool, humid)
5B	Boulder, Colorado (cool, dry)
6A	Minneapolis, Minnesota (cold, humid)
6B	Helena, Montana (cold, dry)
7A	Duluth, Minnesota (very cold)
8A	Fairbanks, Alaska (subarctic)

RESULTS AND DISCUSSION

A comparison study between the simulation results of the VRF and the RTU-VAV models is made to evaluate energy saving potential of VRF system in 16 U.S. climate locations. TMY3 weather files of selected climate locations were used (DOE n.d.). The comparison is mainly based on total annual HVAC site energy usage, which includes total annual heating, cooling, and HVAC fans energy uses.

The results of HVAC site energy use for each system are shown in Figure 2 and Figure 3, while Table 4 lists the detailed HVAC site energy usage of the simulation results for the VRF and the RTU-VAV models. Figure 2 shows annual HVAC site energy usage for the RTU-VAV models. The results indicate that HVAC site energy usage varies within the 16 U.S. climate conditions and it is found that cold climate locations generally tend to consume more HVAC site energy use than hot and warm climate locations due to the heating energy consumption. It shows that the highest HVAC site energy use (558.9 MWh/year or 112.2 kWh/m^2-year) of RTU-VAV models is for Fairbanks, AK. The lowest HVAC site energy use (112.1MWh/year or 22.5kWh/m^2-year) of RTU-VAV models is for Los Angeles, CA. The simulation results of the VRF energy comparison for each climate location are presented in Figure 3. It turns out that the HVAC site energy use of the VRF models for each climate location shows similar energy usage patterns to the RTU-VAV system, and several cold climate locations, such as Minneapolis, MN, Duluth, MN, and Fairbanks, AK, show relatively higher supplementary heating energy uses. The highest HVAC site energy use (385.2MWh/year or 77.3kWh/m^2-year) of the VRF models is for Fairbanks, AK. The lowest HVAC site energy use (50.7 MWh/year or 10.2 kWh/m^2-year) of VRF models is for San Francisco, CA.

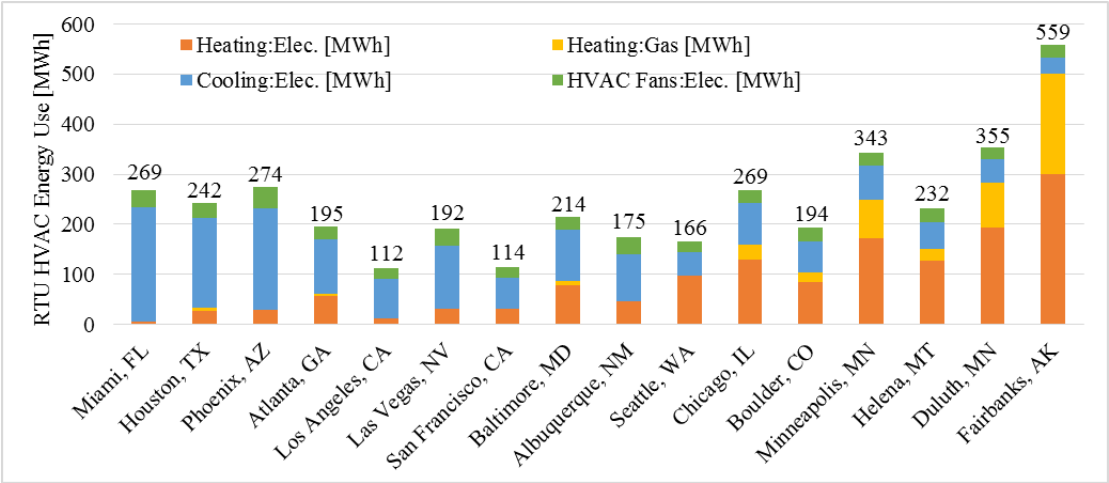

***Figure 3**. Annual HVAC site energy use for RTU-VAV models*

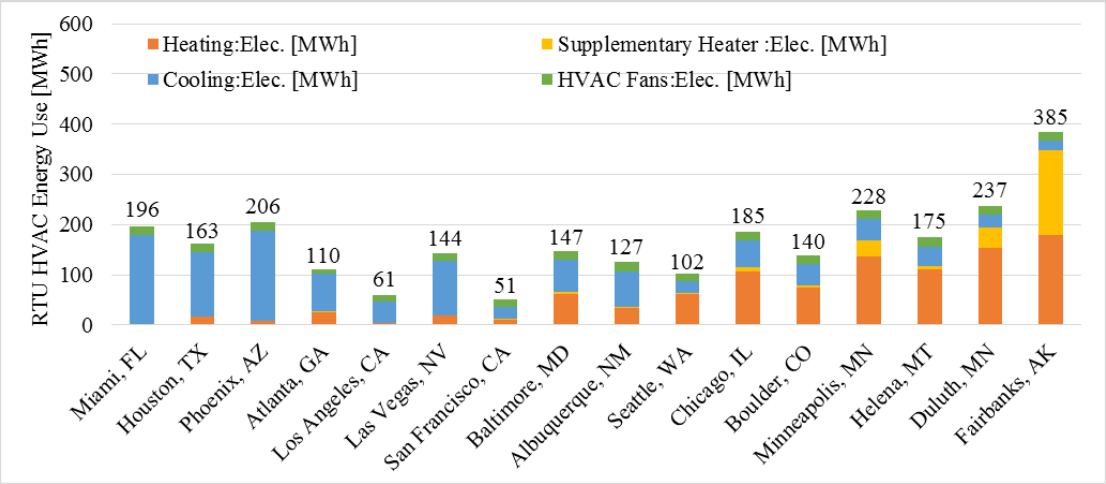

***Figure 4**. Annual HVAC site energy use for VRF HP models*

***Table 4.** Annual HVAC site energy use for RTU-VAV and VRF HP models*

	RTU Heating : Elec. [MWh]	*RTU Heating : Gas [MWh]*	*RTU Cooling : Elec. [MWh]*	*RTU HVAC Fans: Elec. [MWh]*	*VRF Heating : Elec. [MWh]*	*Supplementary Heater: Elec. [MWh]*	*VRF Cooling : Elec. [MWh]*	*VRF HVAC Fans: Elec. [MWh]*
Miami, FL	5.0	0.0	230.2	33.9	0.7	0.1	179.0	16.0
Houston, TX	27.4	6.2	179.6	28.4	16.4	1.0	128.8	16.5
Phoenix, AZ	28.8	0.0	202.4	42.6	8.0	0.5	178.5	19.0
Atlanta, GA	56.0	5.4	108.9	24.9	26.3	1.4	74.1	8.7
Los Angeles, CA	11.6	0.0	79.6	20.9	4.1	0.4	42.2	13.8
Las Vegas, NV	31.7	0.5	124.4	35.5	18.8	0.6	108.7	15.3
San Francisco, CA	31.1	0.0	62.2	20.3	11.6	0.9	23.7	14.5
Baltimore, MD	78.0	9.6	101.3	25.3	62.5	2.8	65.3	16.0
Albuquerque, NM	45.4	0.7	94.9	33.6	35.1	0.9	71.7	19.2
Seattle, WA	96.6	1.5	47.2	20.5	62.5	2.2	22.0	15.5
Chicago, IL	128.6	31.2	83.5	25.4	107.8	8.2	52.7	16.5
Boulder, CO	85.6	18.4	61.7	28.0	75.1	4.5	42.8	17.2
Minneapolis, MN	172.5	76.7	69.2	24.6	136.9	32.3	42.9	16.0
Helena, MT	126.8	23.4	54.0	27.5	111.7	6.7	37.4	18.8
Duluth, MN	193.6	88.7	48.3	24.1	153.3	41.1	26.0	16.6
Fairbanks, AK	300.1	200.3	33.6	24.9	178.7	170.1	19.5	16.9

The energy savings potential for the VRF models compared to the RTU models in 16 U.S. climate locations in terms of the HVAC system components are presented in Figure 5. The simulation analysis shows that the VRF systems use around 25% to 55% less HVAC site energy than the RTU-VAV systems throughout the selected climate locations. The HVAC site energy savings include 55.3% (62.8MWh/year) for San Francisco, CA, which is the highest saving location, and 24.6% (57.0MWh/year) for Helena, MT, which is the lowest saving location within the 16 U.S. climate locations.

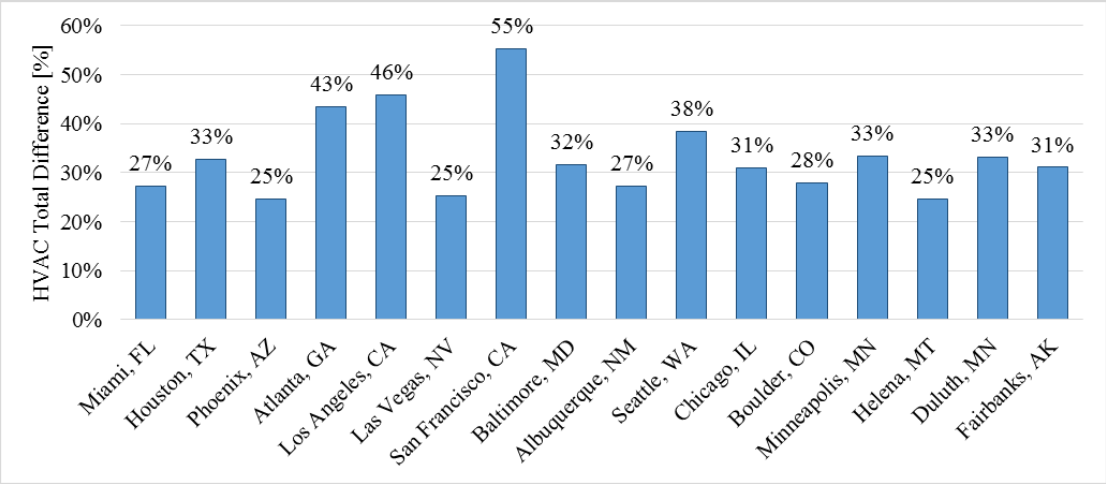

***Figure 5**. Annual HVAC site energy total difference (vs. RTU-VAV system)*

The mild climates, such as San Francisco and Los Angeles, have the largest percentage savings compared to other climate locations as shown in Figure 5 because the RTU-VAV systems use more electric reheat coil energy in those two locations to maintain the room temperature set-point and prevent overcooling the zones during the cooling season. As the modeled RTU is not equipped with economizers and has relatively simple control, the simultaneous cooling and reheating for mild climate zones can be significantly reduced with more enhanced control with addition of economizer, though, which results in less energy savings in the corresponding climate locations.

CONCLUSION AND IMPLICATIONS

The performance of rooftop unit variable air volume (RTU-VAV) and variable refrigerant flow (VRF) were evaluated and compared using whole building energy modeling software, EnergyPlus 8.4, in 16 U.S. climate locations. In this study, the EnergyPlus prototype medium office models were used for the RTU-VAV system and modified to model the VRF system by replacing the VAV system with the VRF system. 16 different weather conditions corresponding with ASHRAE climate zones were used for this study to demonstrate the energy savings potential of the VRF system from the RTU-VSV system.

The simulation results showed that cold climate locations generally tend to show more HVAC energy use than hot and warm climate locations due to higher heating energy consumption for the RTU-VAV and the VRF systems. It was also found that the VRF systems included around 25% to 55% annual energy savings over the RTU-VAV systems in 16 U.S. climate locations for a medium office building.

ACKNOWLEDGEMENTS

This effort was supported by Samsung Electronics and U.S. Department of Energy.

REFERENCES

Amarnath, A. & Blatt, M., 2008. Variable Refrigerant Flow : An Emerging Air Conditioner and Heat Pump Technology Evolution of the Technology. *ACEEE Summer Study on Energy Efficiency in Buildings*, pp.1–13.

ASHRAE. 2013. *ANSI/ASHRAE/IES Standard 90.1-2013. Energy Standard for Buildings Except Low-Rise Residential Buildings,* Atlanta, GA.

Aynur, T.N., 2010. Variable refrigerant flow systems: A review. *Energy and Buildings*, 42(7), pp.1106–1112.

Aynur, T.N., Hwang, Y. & Radermacher, R., 2009a. Simulation comparison of VAV and VRF air conditioning systems in an existing building for the cooling season. *Energy and Buildings*, 41(11), pp.1143–1150.

Aynur, T.N., Hwang, Y. & Radermacher, R., 2009b. Simulation of a VAV air conditioning system in an existing building for the cooling mode. *Energy and Buildings*, 41(9), pp.922–929.

DOE, 2015a. *EnergyPlus Engineering Reference: The Reference to EnergyPlus Calculations, Version 8.4*, Washington, DC.

DOE, 2015b. *EnergyPlus Input Output Reference: The Encyclopedic Reference To EnergyPlus Input and Output, Version 8.4*, Washington, DC.

DOE, EnergyPlus Weather Files: All Regions - North and Central America WMO Region 4 - USA. Available at: https://energyplus.net/weather-region/north_and_central_america_wmo_region_4/USA [Accessed May 20, 2016].

EIA, 2015. Energy Consumption by Sector. *Energy*. Available at: http://www.eia.gov/totalenergy/data/monthly/pdf/sec2_3.pdf [Accessed January 1, 2016].

Goetzler, W., 2007. Variable Refrigerant Flow Systems. *ASHRAE Journal*, 49(4), pp.24–31.

Goetzler, W., Guernsey, M. & Young, J., 2014. Research and Development Roadmap for Emerging HVAC Technologies. , (October), p.121.

Im, P. & Munk, J.D., 2015. Evaluation of Variable Refrigerant Flow (VRF) System Performance Using an Occupancy Simulated Research Building : Introduction and Summer Data Analysis Compared with a Baseline RTU System. *ASHRAE Annual Conference*, pp.1–9.

Murphy, J., 2011. High-performance VAV systems. *ASHRAE Journal*, 53(10), pp.18–28.

Okochi, G.S. & Yao, Y., 2016. A review of recent developments and technological advancements of variable-air-volume (VAV) air-conditioning systems. *Renewable and Sustainable Energy Reviews*, 59, pp.784–817.

Raustad, R., 2013. A variable refrigerant flow heat pump computer model in energyplus. *ASHRAE Transactions*, 119(PART 1), pp.299–308.

Thornton, B. et al., 2011. Achieving the 30% Goal : Energy and Cost Savings Analysis of ASHRAE Standard 90.1-2010. *PNNL-20405*.

Wang, S., 2014. Energy modeling of ground source heat pump vs. variable refrigerant flow systems in representative US climate zones. *Energy and Buildings*, 72,

pp.222–228.

Zhou, Y.P. et al., 2007. Energy simulation in the variable refrigerant flow air-conditioning system under cooling conditions. *Energy and Buildings*, 39(2), pp.212–220.