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Evaluation of energy savings potential of variable refrigerant flow (VRF) from variable air volume (VAV) in the U.S. climate locations

Dongsu Kim [a], Sam J. Cox [a], Heejin Cho [a,*], Piljae Im [b]

[a] Mechanical Engineering, Mississippi State University, Starkville, MS 39759, USA

[b] Building Technologies Research and Integration Center, Oak Ridge National Laboratory, TN 37831, USA

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ABSTRACT

Variable refrigerant flow (VRF) systems are known for their high energy performance and thus can improve energy efficiency 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 HVAC systems such as rooftop variable air volume systems (RTU-VAV) and central chiller and boiler systems. This paper evaluates the performance of VRF and RTU-VAV systems in a simulation environment using widely-accepted whole building energy modeling software, EnergyPlus. A medium office prototype 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 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. The simulation results show that the VRF systems would save around 15–42% and 18–33% for HVAC site and source energy uses compared to the RTU-VAV systems. In addition, calculated results for annual HVAC cost savings point out that hot and mild climates show higher percentage cost savings for the VRF systems than cold climates mainly due to the differences in electricity and gas use for heating sources.

1. Introduction

1.1. Background and purpose

Residential and commercial buildings represent approximately 35% of the energy consumption in the US (EIA, 2015), with heating, ventilation and air conditioning (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 for current HVAC systems and have become one of the primary systems in commercial buildings since being introduced to the US market in the 1970s (Aynur et al., 2009a). 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). Yao et al. (2007) evaluated the energy savings potential of the variable air volume (VAV) system by comparing with constant air volume (CAV) and fan coil unit (FCU) systems in a simulation environment for six different climate zones in China. Their results showed that VAV systems included about 17%–38% energy savings potential when compared to the CAV system and about 5%–10% energy savings when compared to the FCU system throughout diverse locations in China. Aktacir et al. (2006) investigated a life-cycle cost (LCC) of CAV and VAV systems using detailed load profiles as well as initial cost and operating costs. The study showed that the LCC of the VAV system was higher than that of CAV system in the first year mainly due to higher initial cost of VAV system. However, after a certain period of time, the VAV system became economically attractive as the lower operational cost would gradually offset the higher initial cost. Wang et al. (2015) demonstrated the energy savings of the VAV system integrating membrane-based zonal energy recovery ventilators for commercial buildings using EnergyPlus. With additional energy

* Corresponding author.
E-mail address: cho@me.msstate.edu (H. Cho).

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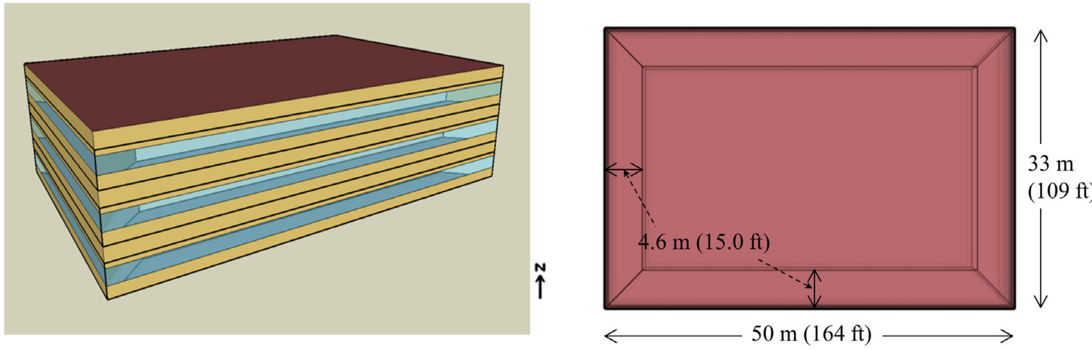

Fig. 1. 3D view (left) and a floor plan (right) of a simulated medium office.

recovery ventilation systems and damper controls, the annual HVAC energy savings potential could achieve around 18%–49% when compared to the baseline model, which did not use air side economizer controls and energy recovery ventilators. Although there have been various efforts to improve the performance of VAV systems over the years, VAV systems still tend to have issues with controlling dampers, supply air temperature, and ventilation of multiple zones in commercial buildings because of their complex structures and configurations (Okochi and 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 and Blatt, 2008). As a relatively new HVAC technology in the US, VRF systems have been gaining more attention in comparison to conventional HVAC systems, e.g. constant air volume and VAV systems (Aynur et al., 2009b). Zhou et al. (2007) analyzed a comparative study with VAV and FCU systems and investigated the energy savings potential of VRF systems using EnergyPlus. Their simulation results showed that VRF systems could consume about 11%–22% less energy use when compared to FCU and VAV systems, respectively. Goetzler (2007) also mentioned the high efficiency VRF models tended to show about 30%–40% less energy use than central chiller/boiler systems. Im and Munk (2015) evaluated the energy performance of a multi-split VRF system compared to the typical RTU-VAV systems, which were both installed in the Oak Ridge National Laboratory's Flexible Research Platform (FRP) building. In their experimental analysis, energy savings of the VRF system were estimated to be around 20% over the RTU-VAV system during cooling operation. Yu et al. (2016) performed the comparative study between VRF and VAV systems in terms of the cooling energy savings under two different weather conditions. They presented that VRF systems could achieve about 40%–53% energy savings when compared to VAV systems, depending on operating modes and indoor temperature setpoint. However, several concerns for the adoption of VRF system in the US still exist such as lack of awareness for energy efficiency advantages, higher initial cost, and lack of optimized and integrated VRF control strategies (Goetzler, 2007; Yu et al., 2016; Aynur, 2010).

Therefore, as an effort to evaluate the energy savings potential of VRF systems, this study evaluates 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 evaluate the energy and cost savings potential of VRF systems over RTU-VAV systems in 16 US climate locations.

2. Simulation model

2.1. Simulation software

DOE's flagship building energy modeling software, EnergyPlus version 8.4, was used for the energy simulation modeling in this study. EnergyPlus has three basic components: a simulation manager, a heat and mass balance simulation module, and a building systems simulation module, inherited from the most popular features and capabilities of BLAST and DOE-2 (Crawley et al., 2001). Zone heating and cooling loads can be calculated by the heat balance method, which is recommend by American Society of Heating, Refrigerating, and Air Conditioning Engineers (ASHRAE) (ASHRAE, 2009), and then passed to the building HVAC system modules at the same time step to calculate heating and cooling system, plant, and electrical system response. EnergyPlus has been extensively validated through analytical, comparative, and empirical tests through ASHRAE 14-2002 guidelines (ANSI/ASHRAE, 2002), and the HVAC simulation results of EnergyPlus have shown good agreement with the following well-known simulation tools: DOE-2.1E, TRANSYS, and ESP-r (Witte et al., 2006). In addition, the detailed contents in terms of testing and validating the current EnergyPlus version components have been presented in previous studies (DOE, 2016).

2.2. Building description

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

The prototype medium office has a rectangular floor plan and total floor area of 4982 m^2. Each floor has 5 thermal zones, including 4 perimeter zones and 1 core zone with 33% window-to-wall fraction. All construction details and internal heat gains are based on ASHRAE Standard 90.1-2010 requirements. This study did not compare the energy consumption for equipment and lighting power as these are not changed. Only the HVAC energy consumptions are compared between two systems.

2.3. RTU-VAV system

The prototype building model has the RTU system with the VAV electric reheat. This system is the air conditioning system that varies the supply air volume flow rate through the air handing unit (AHU) using dampers located in the VAV terminal box in order to curtail heating and cooling loads and meet the set-point temperature (DOE, 2015a). Original input values are mostly used for the RTU-VAV system as described in Table 1 (Thornton et al., 2011; Goel et al., 2014).

Although most of original input values are directly obtained from the prototype medium office models, some of input values are slightly modified in this study, including Energy Manage System

Table 1
EnergyPlus HVAC model for RTU-VAV versus VRF systems.

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	3.23
Heating COP	0.8 (heating gas burner efficiency)	3.20
Thermostat set-point (occupied hours)	24 °C for cooling/21.1 °C for heating	24 °C for cooling/21.1 °C for heating
Thermostat set-back (unoccupied hours)	26.6 °C for cooling/15.5 °C for heating	26.6 °C for cooling/15.5 °C for heating
Supply fan type	Fan: variable volume	Fan: on/off
Indoor supply fan efficiency (%)	60%–62% depending on the fan motor size	0.7
Outdoor ventilation air	0.000431773 $m^3/s - m^2$ (0.0085 cfm/ft^2)	0.000431773 $m^3/s - m^2$ (0.0085 cfm/ft^2)
Supply air set-point manager	Outdoor air reset set-point (differential dry bulb in economizer control type): 15.6 °C if outdoor air is lower than 10 °C and 12.8 °C if outdoor air is higher than 21.1 °C Constant supply air set-point (differential enthalpy in economizer control type): 12.8 °C for cooling and heating operation	N/A
Supplementary heater type	N/A	Zonal baseboard convective elec. heater (natural convection unit)
Supplementary heater heating capacity	N/A	Auto-sized to design day
Supplementary heater efficiency	N/A	0.97

(EMS) controls for the thermostat set-point temperature optimum controls and the domestic hot water system. EMS inputs for the optimum set-point temperature are modified to maintain the same set-point temperature for both the RTU-VAV and the VRF systems, provided in Table 1, and the domestic supply hot water system was removed for the RTU-VAV system since only the HVAC energy consumptions are compared between the RTU-VAV and the VRF systems. Although night-time set-back temperatures are used for most of the cities, some models in hot and warm climate zones do not use the set-back temperature during the cooling season, depending on outside weather conditions (Thornton et al., 2011).

As an economizer system is one of requirements for ASHRAE 90.1-2010 Standard with medium commercial buildings, the RTU-VAV prototype included the economizer (required by ASHRAE 90.1-2010) with the supply air reset temperature controls based on outdoor temperature. The economizer was operated with two different control types. The differential enthalpy economizers are used in 1A, 2A, 3A, and 4A climates' prototype models while the differential dry-bulb economizers are used for all other climate locations (DOE, 2015b).

2.4. VRF system

VRF systems distribute refrigerant to each terminal unit usually placed in each thermal zone. VRF systems use a variable speed compressor and electronic expansion valve to independently vary the flow rate to each terminal unit to meet the thermal load. There are two general types of VRF systems: heat pump (HP) and heat recovery (HR) systems. A VRF HP system only provide either heating or cooling modes in the indoor units during the same operation, while a VRF HR system is capable of operating simultaneously in heating and/or cooling modes. In this study, the VRF HP system is used, and the VRF HP models in EnergyPlus have been developed and improved from version 7.2 through 8.4 for the simulation accuracy and the ability to consider the dynamics of more operational parameters (Raustad, 2013; Hong et al., 2016). EnergyPlus 8.4 provides two different VRF models to simulate the energy performance of the VRF HP systems: the System Curve based Model (VRF-SysCurve), updated at EnergyPlus 7.2, and the Physics based Model (VRF-FluidTCtrl), which was updated at EnergyPlus version 8.4 (DOE, 2015a). In this study, the VRF-SysCurve model is used, based on empirical performance curves found in manufacturers' literature (Raustad, 2013).

For the VRF HP simulation models, the original prototype medium offices are modified by replacing the RTU-VAV system with the VRF HP system. The VRF HP units are operated by the master zone thermostat to determine the operational mode between cooling or heating (DOE, 2015b). The VRF HP system tends not to provide adequate thermal comfort in some thermal zones when the system cannot provide simultaneous cooling and heating, especially during heating season (Im et al., 2015). To overcome this limitation, separate outdoor units for core and perimeter zones are modeled to be able to meet simultaneous cooling and heating requirements. In total, six outdoor units are created to cover all core and perimeter zones in the three-floor building, and each zone has one indoor unit.

Table 1 summarizes the HVAC input details of both the RTU-VAV and the VRF HP systems. The default performance curves in the EnergyPlus VRF HP model (Raustad, 2013) are used, and the coefficient of performance (COP) values for the rated cooling and heating are set to 3.23 and 3.2, respectively, for the VRF HP system. Although the rated cooling and heating COPs of the VRF HP system can vary depending on the VRF HP system capacity, the values used in this study are determined based on the VRF HP minimum efficiency requirements prescribed in ASRHAE Standard 90.1-2010 (ASHRAE, 2011). All cooling and heating capacities are auto-sized to design day conditions corresponding with ASHRAE climate zones in both the RTU-VAV and the VRF HP systems. For zone outdoor air (OA) requirements based on number of occupants and zone floor area, ventilation optimization are implemented using the 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 include zone minimum OA requirement as determined by ASHRAE Standard 62.1 (Thornton et al., 2011). Although the RTU-VAV models always use OA ventilation optimization controls to maintain OA requirements with the economizer operation during simulation period, the VRF models employ a simple way to bring OA through individual zones using the "Outdoor Air Mixer (OAM)" object in EnergyPlus during the VRF HP system's operation.

In addition, supplementary electric heaters are added with the VRF HP simulation models since the VRF system provided by EnergyPlus 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

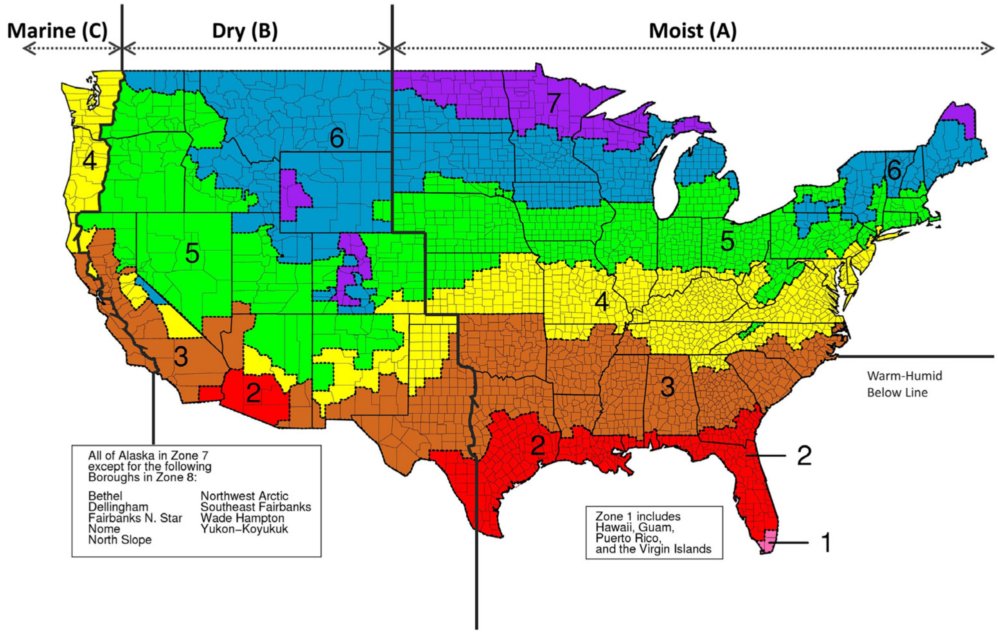

Fig. 2. ASHRAE climate zones in the United State (Briggs et al., 2003).

Table 2
16 cities representing the climate zones (Briggs et al., 2003).

Climate zone	16 representative cities	Condition
1A	Miami, Florida (FL)	Very hot, humid
2A	Houston, Texas (TX)	Hot, humid
2B	Phoenix, Arizona (AZ)	Hot, dry
3A	Atlanta, Georgia (GA)	Warm, humid
3B-Coast	Los Angeles, California (CA)	Warm, dry
3B	Las Vegas, Nevada (NV)	Warm, dry
3C	San Francisco, California (CA)	Warm, marine
4A	Baltimore, Maryland (MD)	Mixed, humid
4B	Albuquerque, New Mexico (NM)	Mixed, dry
4C	Seattle, Washington (WA)	Mixed, marine
5A	Chicago, Illinois (IL)	Cool, humid
5B	Boulder, Colorado (CO)	Cool, dry
6A	Minneapolis, Minnesota (MN)	Cold, humid
6B	Helena, Montana (MT)	Cold, dry
7	Duluth, Minnesota (MN)	Very cold
8	Fairbanks, Alaska (AK)	Subarctic

space by only natural convection, which has an immediate impact on the zone air heat balance (DOE, 2015a). The heating and cooling set-points for the VRF HP models shown in Table 1 are the same as that of the RTU-VAV models.

2.5. Validation studies of EnergyPlus VRF and RTU-VAV models

The validation process of the VRF HP and the RTU-VAV simulation models in EnergyPlus have been previously validated with measured data in various studies (Aynur et al., 2009b; Zhou et al., 2007; Hong et al., 2016; Zhou et al., 2008; Kwon et al., 2012; Li et al., 2009).

Aynur et al. (2009b) experimentally evaluated validation of EnergyPlus of the variable air volume (VAV) air conditioning system in the existing office. The packaged VAV system was used with the actual values of the construction information and internal heat gains such as occupants, lighting, and office equipment. The manufacturer's data for the existing RTU and the VAV boxes was used in the simulation package inputs in EnergyPlus. Their comparison results showed that the simulation results from the RTU-VAV model fit reasonably well with the measured data, including 71.1% of all simulated power consumption data was within ±15% range from the measured data. This is mainly due to the difference of solar weather data between the weather data location they used and the experimental location. It was also found that most of the indoor temperature and the relative humidity were within ±1.5 °C and ±18% range, respectively, from the simulated data.

Sharma and Raustad (2013) used an empirical model to simulate the VRF HP system using the existing VRF module in EnergyPlus 7.2 and compared electric consumption for the lab-measured and simulated data. Their results showed that about 72% of all the simulated total energy falls within ±25% of the measured data of total energy and included around 21% for coefficient of variation of the root mean square error (CV-RMSE), which is a reasonable variability between measured and simulated data. Zhou and Wang (2006) developed the VRF simulation module in EnergyPlus and then validated it using experimental data (Zhou et al., 2008). They showed the simulated results of the VRF air conditioning system from EnergyPlus agreed well with the measured data with a mean relative error around 25% and 28%, respectively, for the total cooling energy and power consumption. The average errors also included 6% for COP and 18% for part load ratio (PLR). They also pointed out that both the air conditioning system with proper controls and the building element information need to be considered carefully in order to achieve accurate performance data of the VRF system in EnergyPlus. Kwon et al. (2012) evaluated the validation of the VRF HP model in the EnergyPlus version 7.2. They highlighted the root mean square deviation of daily, weekly, and monthly electricity use between the simulated and measured data are 5.6 kWh, 11.1 kWh, and 37.6 kWh, respectively. It is also found that the average absolute error in electricity use is 7.8%, 2.4%, and 2.2% for the daily, weekly, and monthly values, respectively, for the entire simulation period.

3. Climate locations

The building models with each system are placed in 16 different cities representing all US 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 are used for this study to evaluate the energy savings potential of the VRF system from the RTU-VAV system. Table 2 lists the 16 representative locations identified in ASHRAE Standard 90.1-2010 in the US climate zones (see Fig. 2).

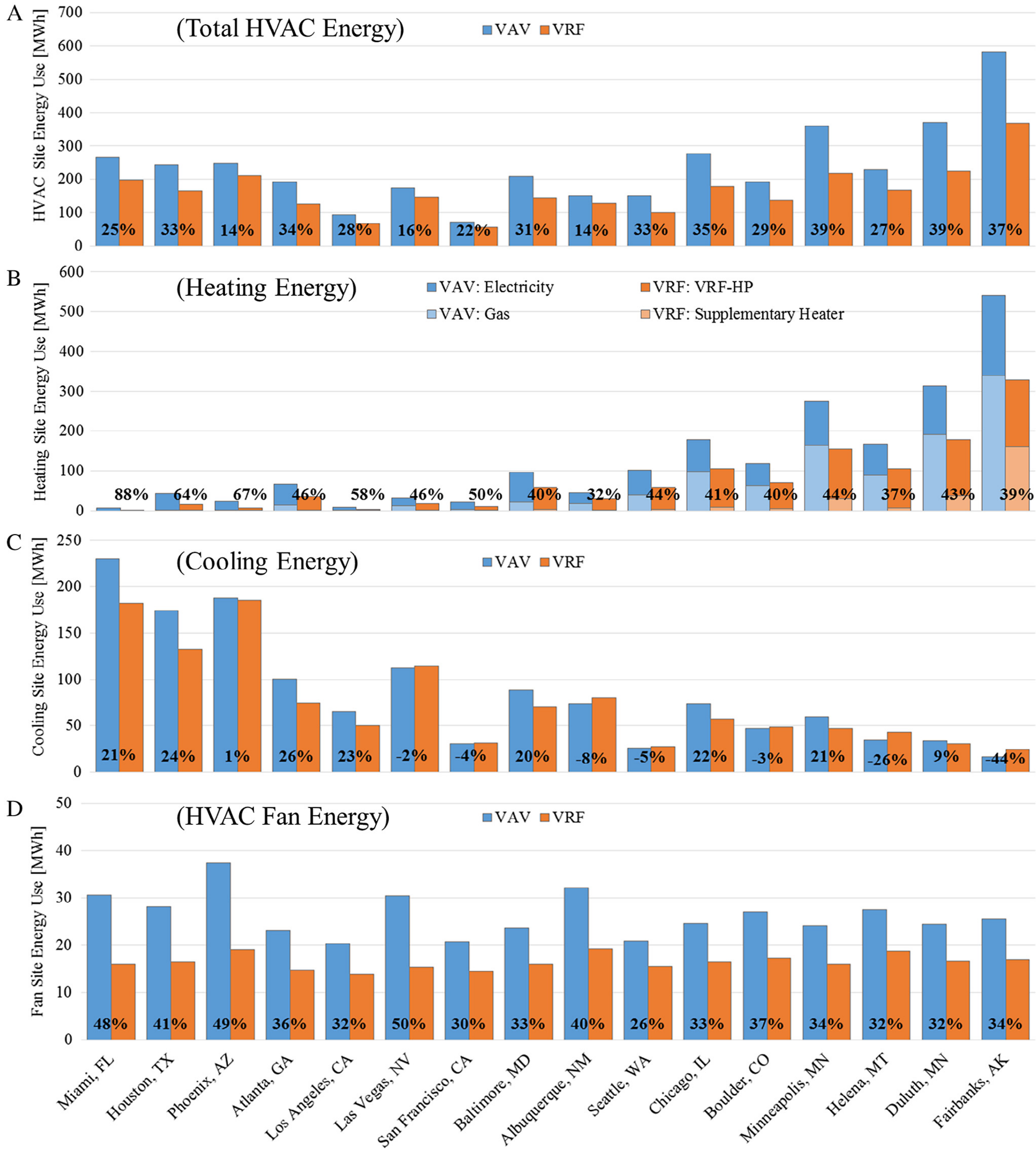

Fig. 3. Annual HVAC site energy savings for the VRF and RTU-VAV models; (A) HVAC total site energy, (B) heating site energy, (C) cooling site energy, and (D) HVAC fan site energy savings potential of the VRF systems.

4. Results and discussion

4.1. Comparison of annual HVAC site energy use

A comparison study between the simulation results of the VRF HP and the RTU-VAV models is made to evaluate energy and cost savings potential of the VRF HP systems in 16 US climate locations. The comparison is based on all annual HVAC energy consumption used within the entire prototype office building, which includes total annual heating, cooling, and HVAC fan energy usage. The results of HVAC site energy use for each system are shown in Fig. 3.

Percentage saving of the HVAC energy for 16 climates when compared to the RTU-VAV models is calculated as follows:

$$E_{savings} = \left(1 - \frac{E_{\mathrm{VRF}}}{E_{\mathrm{VAV}}}\right) \times 100. \quad (1)$$

Fig. 3 shows annual HVAC site energy usage and savings for the VRF HP and the RTU-VAV models. As shown in this figure, the simulation analysis results in the VRF HP models use less annual

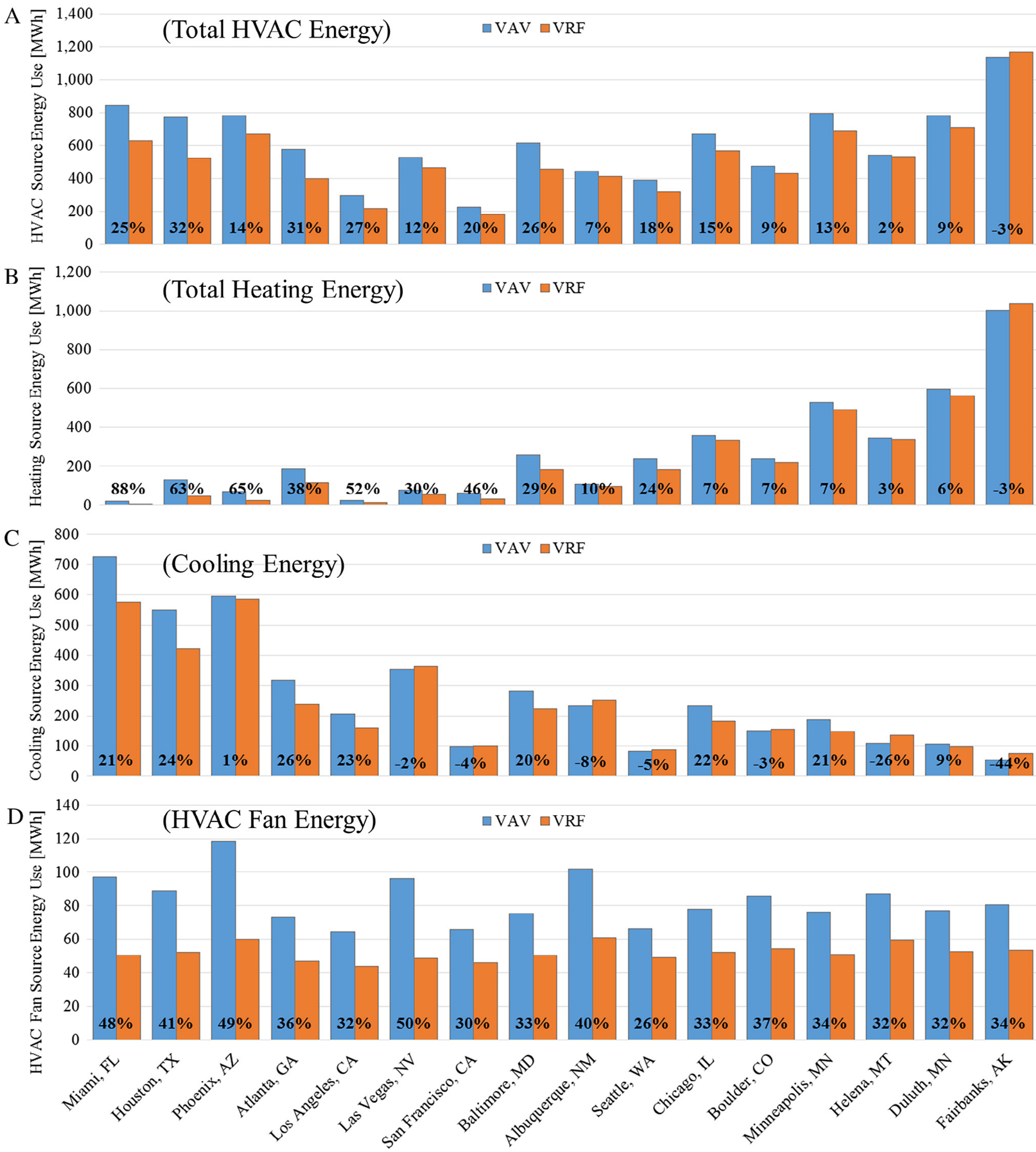

Fig. 4. Annual HVAC source energy savings for the VRF and RTU-VAV models; (A) HVAC total source energy, (B) heating source energy, (C) cooling source energy, and (D) HVAC fan source energy savings potential of the VRF systems.

HVAC site energy than the RTU-VAV models in all climate zones, and cold climates tend to use more HVAC site energy than mild climates mostly due to the heating energy consumption. The total annual HVAC site energy savings are about 14%–39% for the 16 climate zones when compared with the RTU-VAV models.

Annual heating site energy savings of the VRF HP models from the RTU-VAV models are presented in Fig. 3(B). Annual heating site energy usage for the RTU-VAV models includes heating electricity consumption of the VAV reheat coils and gas consumption of main air handing unit (AHU) heating coils. For the VRF HP models, annual heating site energy includes the VRF heating coils and the supplementary heaters. The results indicate the VRF HP models consume about 32%–88% less heating energy use for 16 climate locations when compared to the RTU-VAV models. It also shows that the highest percentage savings for heating site energy use is around 88% (5.4 MWh/year) for Miami, and the lowest percentage savings is around 32% (14.5 MWh/year) for Albuquerque.

Fig. 3(C) shows annual cooling site energy differences between the VRF HP and the RTU-VAV models. Cooling energy savings tend to be less than heating energy savings. In some climates, the VRF HP models spend more cooling energy compared to that of the RTU-VAV models, mainly due to use of economizers and OA reset con-

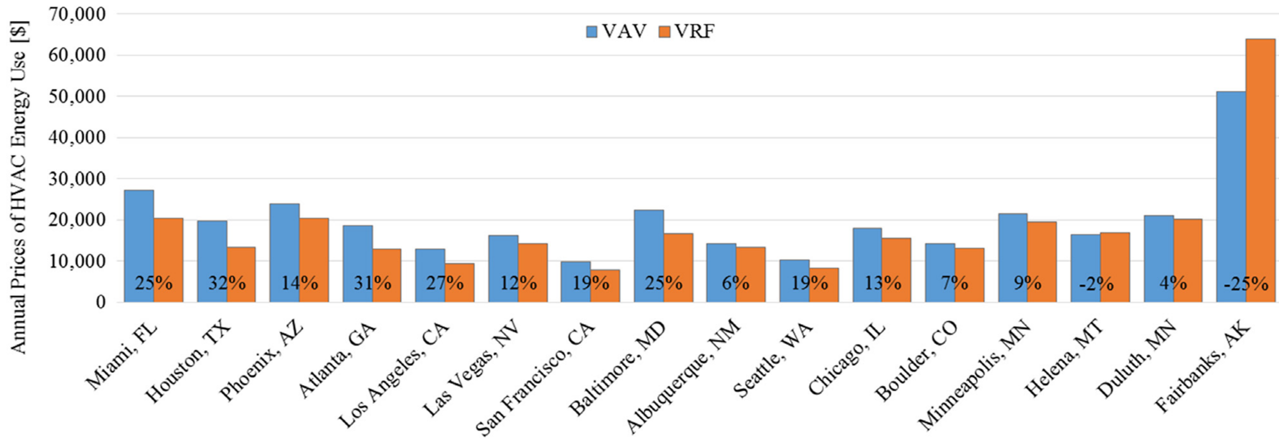

Fig. 5. Calculated annual HVAC energy cost savings potential of the VRF models in 16 climate locations.

trols for the RTU-VAV models. It shows that the highest percentage savings for the VRF HP models compared with the RTU-VAV cooling site energy is around 26% (25.8 MWh/year) for Atlanta, GA. The lowest percentage saving is around 1% (2.6 MWh/year) for Phoenix, AZ. As the modeled RTU-VAV is equipped with economizers and has a supply air OA reset control, the simultaneous cooling and reheating for some climate zones can be significantly reduced with more enhanced controls with addition of economizers for the RTU-VAV models. This process results in less cooling energy usage for the RTU-VAV models and less saving potential of the VRF HP models in the corresponding climate locations.

Energy savings in terms of the annual HVAC fan site energy consumption are also the major energy savings potential for the VRF HP systems compared to the RTU-VAV systems as shown in Fig. 3(D). The VRF HP models used an on–off fan, which operates based on heating and cooling loads, while a variable air volume fan is used for the RTU-VAV models commonly used for VAV systems. For fan operating schedules, the VRF HP models are always turned on to satisfy general OA requirements provided through the OA mixer object. In contrast, the RTU-VAV models are operated based on the minimum OA requirements with economizers, which are not available for the current VRF system in EnergyPlus 8.4. When compared to the RTU-VAV models as a percentage of total HVAC fan energy use, the simulation analysis as shown in Fig. 3(D) shows that the VRF HP models use around 26%–50% less HVAC fan site energy than the RTU-VAV models throughout the chosen climate locations. The HVAC fan site energy savings potential of the VRF HP models include 50% (15.1 MWh/year) for Las Vegas, NV, which is the highest saving location, and 26% (5.4 MWh/year) Seattle, WA, which is the lowest saving location. These results are expected mainly due to the fact that the RTU-VAV fans served by the AHU tend to have high enough static pressure to push out all the air to each VAV terminal box while the VRF indoor unit fans have lower static pressure to supply conditioned air to each zone since the VRF refrigerant delivers heat and cool to indoor units.

4.2. Comparison of annual HVAC source energy use

The "source" energy is calculated from the "site" energy. Source energy is the energy usage at the utility generating facility needed to provide the electricity used at the site, and the embedded energy of fuel delivered to the site, such as the natural gas (Deru and Torcellini, 2007). Since the power mix of the electric grid varies with demand loads, site to source energy conversion factors can be changed based on state locations, energy types, and time. The default conversion factors that EnergyPlus 8.4 provides in the output report, therefore, were used in this study and kept the same values across all the climates locations. The conversion factors from site energy to source energy were set to 3.167 (source kWh per site kWh) for electricity and 1.084 (source kWh per site kWh) for natural gas (DOE, 2015b).

Fig. 4 shows annual HVAC source energy savings potential of the VRF HP models in 16 US climates. The calculated analysis turns out that the VRF HP models consume about 2%–32% less HVAC source energy from the RTU-VAV models. Comparing heating source energy usage as a percentage of total HVAC source energy consumption, the highest percentage savings for heating source energy is around 88% (17.0 MWh/year) for Miami, FL, and the lowest percentage savings is around 3% (10.4 MWh/year) for Helena, MT, respectively. For cooling source energy savings potential of the VRF HP models as seen in Fig. 4(C), the percent savings are the same for site and source energy because all cooling energy usage is from electricity. The calculated results include around 26% (81.6 MWh/year) for Atlanta, GA, which is the highest percentage savings when compared with the RTU-VAV models. In addition, the lowest percentage saving is around 1% (8.1 MWh/year) for Phoenix, AZ, while some climates show that the VRF HP models consume more cooling source energy than the RTU-VAV models due primarily to the precooling of economizers and supply air OA reset controls. As expected, HVAC fan source energy savings potential of the VRF HP models also point out the same pattern compared to site energy usage, while the HVAC fan source energy uses are significantly higher than the site energy usage. The calculated results indicate that the highest percentage savings for heating source energy use is around 50% (47.9 MWh/year) for Las Vegas, NV, and the lowest percentage savings is around 26% (17.1 MWh/year) for Seattle, WA.

4.3. Comparison of annual HVAC cost savings potential in climate locations

The energy cost savings potential of the VRF HP systems compared with the RTU-VAV systems is estimated within the 16 climate locations using the prototype medium office model. The average electricity and natural gas prices for the states in 2015, provided in Table 3, are found from the US Energy Information Administration (EIA) and used in the cost savings analysis for each representative climate location.

The energy costs and the VRF HP systems savings compared with the RTU-VAV systems in the 16 climate locations are summarized in Fig. 5.

Fig. 5 represents the calculated cost savings of the VRF HP models. The percentage HVAC cost savings are around 4%–32%. The highest percentage cost savings for annual HVAC energy

Table 3
Average electricity and natural gas prices in 16 representative cities.

	Average price of electricity ($ per kWh)	Average price of natural gas ($ per Mcf [1000 ft^3])
Miami, FL	0.102	10.7
Houston, TX	0.081	8.3
Phoenix, AZ	0.097	10.5
Atlanta, GA	0.102	8.5
Los Angeles, CA	0.140	8.0
Las Vegas, NV	0.098	8.7
San Francisco, CA	0.140	8.0
Baltimore, MD	0.116	10.0
Albuquerque, NM	0.104	7.9
Seattle, WA	0.082	9.1
Chicago, IL	0.087	7.3
Boulder, CO	0.097	8.2
Minneapolis, MN	0.0897	7.3
Helena, MT	0.1006	7.82
Duluth, MN	0.0897	7.3
Fairbanks, AK	0.1733	7.8

use is around 32% (6383.7 $/year) for Houston, TX, and the lowest percentage savings is around 4% (796.1 $/year) for Duluth, MN. However, several cold climate locations, such as Fairbanks, AK and Helena, MT, show the VRF HP systems are about 2%–25% more costly for HVAC energy usage than the RTU-VAV systems.

5. Conclusion and implications

The performance of the rooftop unit variable air volume (RTU-VAV) and the variable refrigerant flow heat pump (VRF HP) were evaluated and compared using whole building energy modeling software, EnergyPlus 8.4, in 16 US 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. Sixteen 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-VAV system.

The simulation results showed that cold climate locations generally tended to show more HVAC site energy uses than hot and mild climate locations due to the heating energy consumption for both the RTU-VAV and the VRF HP systems. It was also found that the VRF HP models included around 14%–39% annual HVAC site energy savings potential over the RTU-VAV models in 16 US climate locations. After conversion to source energy use, annual HVAC source energy savings were estimated to be about 2%–32% for the VRF HP models. Comparing annual HVAC cost savings as a percentage savings, the VRF HP models mostly showed higher cost savings potential than the RTU-VAV models within hot and mild climates, while the RTU-VAV models used less HVAC energy costs in several cold climate zones, mainly due to the differences in electricity and gas consumption.

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