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Field Performance Measurements of VRF System with Subcooling Heat Exchanger

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Modeling of Variable Refrigerant Flow System for the Cooling Season

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ABSTRACT

In this study, the cooling performance of the multi-split variable refrigerant flow (VRF) air conditioning system operated in the academic building environment was simulated with EnergyPlus software, which has a new module for VRF heat pump systems. Simulation results were validated with the field test results during the cooling season. The comparison result shows that 87.5% of all simulated daily power consumption data agree with the experimental data within ±15% deviation. The root-mean-square deviations of daily, weekly and monthly electricity power consumptions for the total simulation period between the simulated and measured values are 5.63 kWh, 11.12 kWh and 37.58 kWh, respectively. The averages of the absolute values of the daily, weekly and monthly relative error for the total simulation period are 7.97%, 2.40% and 2.22%, respectively.

1. INTRODUCTION

Variable refrigerant flow (VRF) air conditioning systems have been widely used for residential and commercial buildings. VRF systems consist of an outdoor condensing unit and a network of indoor units connected through refrigerant piping within the conditioned space. Cooling loads and system cooling capacity vary with many parameters such as environmental conditions, the number of indoor units operating in cooling or heating mode, airflow rates, internal loads in each zone, and indoor conditions. A VRF heat pump system's indoor units control the capacity by using an electronic expansion valve. At the system level, the outdoor unit conducts load management through the inverter-driven variable-speed compressor, or alternative combinations of outdoor unit fan motors. The target evaporating temperature and degree of superheating are monitored and used for controlling the compressor speed, the fan motor speed, and the opening of the EEV to ensure a stable target temperature control, regardless of varying loads and environmental conditions.

Since the VRF systems, which were introduced about 20 years ago in Asia, have become popular in many countries, this technology has been widely studied, both experimentally and numerically. Zhou *et al*. (2007) compared the energy consumption of the VRF system with variable air volume (VAV) system and fan-coil plus fresh air (FPFA) system numerically. Their simulation results showed that the energy-saving potentials of the VRF system were expected to be 22.2% and 11.7% as compared with the VAV system and the FPFA system, respectively. Zhou *et al*. (2008) tested the performance of VRF system and used their data for validating their customized VRF simulation module. The mean relative error of VRF system's daily power consumption was 28.31%. Kang *et al*. (2009) experimentally investigated a heat recovery type VRF system, which can provide heating and cooling simultaneously to different zones of a building. The optimized system COP was 146.5% and 133.7% higher compared to the cooling-only and heating-only mode of the non-heat recovery type VRF system, respectively. Li *et al*. (2009) developed a module for a water-cooled VRF system in the software of EnergyPlus. They numerically compared the power consumption of water-cooled VRF system with those of the air-cooled VRF system and the FPFA system. Although the monthly power consumption of FPFA system was higher than the two systems, the difference in power consumption between the water-cooled VRF and the air-cooled one was small. Liu and Hong (2010) investigated the energy efficiency of the heat recovery type VRF system and the ground source heat pump (GSHP) system using EnergyPro and eQUEST, respectively. For the locations representing hot and cold climates, the GSHP system saves up to 24.1% of cooling and heating energy consumption as compared to the heat recovery type VRF system. Aynur *et al*. (2006) experimentally investigated the performance of a VRF air conditioning system in a field test during the cooling season with different control modes, master control and individual control. They found that the performance of VRF system in the individual control mode was 8.6% higher than that of the VRF master control mode. The main disadvantage of VRF system is that it cannot provide any ventilation by itself. The effect of the ventilation on the VRF system's performance including the indoor thermal comfort, energy consumption and system efficiency was investigated experimentally and numerically by Aynur *et al*.(2008a and 2008b).

In this study, the cooling performance of the multi-split VRF air conditioning system operated in the academic building environment was simulated with the VRF module developed for EnergyPlus version 7.0 software (DOE, 2011), which is verified and adopted by the EnergyPlus. Then the simulated performance of a multi-split VRF air conditioning system was compared with test results for validation.

2. MODEL DEVELOPMENT

2.1 Building Description

Figure 1 illustrates the target building and its zones for modeling, located in College Park, Maryland. The Legacy OpenStudio Plug-in for Google SketchUp (DOE, 2011) was used to create the building geometry in EnergyPlus input file. Table 1 shows the characteristics of the internal loads in each thermal zone. A ceiling height is 2.48 m. The hallway and three unconditioned zones were included for the simulation. Since there were two indoor units in Room B and Room C, each room was sub-divided into two thermal zones with imaginary air wall, which had a very low thermal conductivity, as shown in Figure 1. Construction material properties are listed in Table 2. Interior wall 2 was used between each room and hallway.

2.2 VRF System Description

Newly developed EnergyPlus Objects (DOE, 2011) (ZoneHVAC: Terminal: VariableRefrigerantFlow and AirConditioner: VariableRefrigerantFlow) were used to model the multi-split VRF system. Figure 2 shows the node connections of VRF terminal units with draw-through fan placement. EnergyPlus Object (DOE, 2011) (Fan:ConstantVolume) was used to model the indoor fans since they were operated continuously with constant air volume flow rate during the cooling operation. An operating mode, either cooling or heating, was decided from the following options: LoadPriority, ZonePriority, ThermostatOffsetPriority and MasterThermostatPriority.

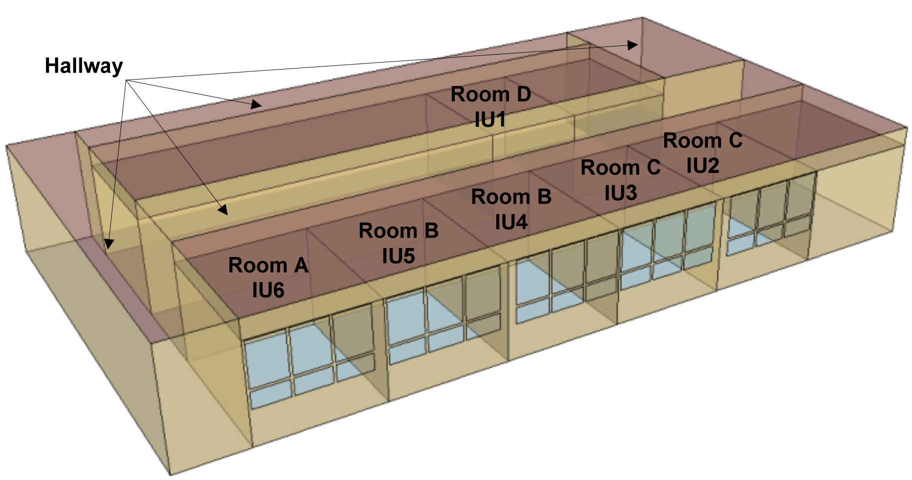


Figure 1: Illustration of 3D image of the target office suite

Table 1: Number of internal load sources

Room	Room A	Room B1	Room B2	Room C1	Room C2	Room D
Space Area (m^2)	11.36	11.07	11.07	11.19	11.19	9.34
Number of Occupants	1	3	4	3	4	1
Office Equipment (W/m^2)	73.50	56.46	54.20	55.85	53.62	89.94
Lighting (W/m^2)	13.20	13.55	13.55	13.40	13.40	16.06

Table 2: Building material characteristics

Section	Exterior Wall	Interior Wall 1	Interior Wall 2	Ceiling	Windows	Roof
Constructions	Bricks, Air and Plaster	Gymsum Plaster and Air Space	Bricks	Acoustic Tile	Two Single-Pane Glazing with Air Gap	Insulation, Brick and Polyurethane
Total Thickness (m)	0.177	0.124	0.152	0.025	0.014	0.184

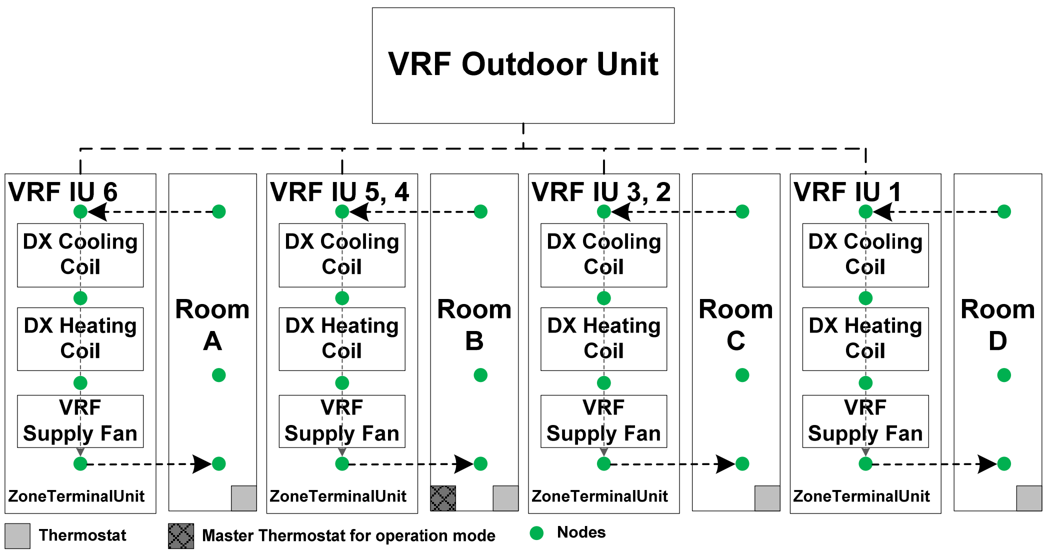


Figure 2: Nodes connection of VRF terminal unit and location of thermostats

The default option, MasterThermostatPriority, is the control based on the master thermostat by which all terminal units are operated. Since the multi-split VRF system could not provide any ventilation, outdoor air flow rate to the zone terminal unit during cooling operation was set to be zero in simulation. The model uses performance information at reference conditions along with the correlations for the cooling capacity and power consumption. The VRF system performance correlations were generated by curve fitting the manufacturer's performance data.

The operating capacity of the heat pump ($CAPFT_{OU,cooling}$) is calculated using the Equation (1), which is a function of a load-weighted average indoor web-bulb temperature ($T_{wb,avg}$) and outdoor dry-bulb temperature (T_c).

$$CAPFT_{OU,cooling} = a + b(T_{wb,avg}) + c(T_{wb,avg})^2 + d(T_c) + e(T_c)^2 + f(T_{wb,avg})(T_c) \quad (1)$$

In order to correct the performance for off-design cases, the cooling combination ratio correction factor ($CR_{cooling,correction}$) is calculated as a function of the rated cooling combination ratio ($CR_{cooling,rated}$), which is defined as the total terminal unit's rated cooling capacity divided by the heat pump system's rated cooling capacity as in Equation (2).

$$CR_{cooling,correction} = a + b(CR_{cooling,rated}) + c(CR_{cooling,rated})^2 + d(CR_{cooling,rated})^3 \quad (2)$$

$$\text{where } CR_{cooling,rated} = \frac{\sum_1^6 \dot{Q}_{coil(i),cooling,rated}}{\dot{Q}_{coil,cooling,rated}}$$

The heat pump's total cooling capacity ($\dot{Q}_{OU,cooling,total}$) is then calculated from the rated cooling capacity ($\dot{Q}_{OU,cooling,rated}$) using Equations (1) and (2) as shown in Equation (3).

$$\dot{Q}_{OU,cooling,total} = \dot{Q}_{OU,cooling,rated}\left(CAPFT_{OU,cooling}\right)\left(CR_{cooling,correction}\right) \quad (3)$$

The cooling energy input ratio modifier (*EIRFT*) is a function of temperature as shown in Equation (4). The cooling energy input ratio modifier (*EIRFPLR*) is a function of Part Load Ratio (*PLR*) as shown in Equation (5).

$$EIRFT_{OU,cooling} = a + b(T_{wb,avg}) + c(T_{wb,avg})^2 + d(T_c) + e(T_c)^2 + f(T_{wb,avg})(T_c) \quad (4)$$

$$EIRFPLR_{cooling} = a + b(PLR) + c(PLR)^2 + d(PLR)^3 \quad (5)$$

The energy efficiency of the VRF system is highly affected by the capability of adjusting heating and cooling outputs to meet the dynamic building heating and cooling loads. The part load ratio (*PLR*) is defined as the ratio of total heat pump condenser cooling load and heat pump total available cooling capacity. The cycling ratio (*CyclingRatio*) is defined as the ratio of the *PLR* and the minimum *PLR* obtained from the experiment as shown in Equation (6).

$$CyclingRatio = PLR/PLR_{\min} \quad (6)$$

Equation (7) shows the Cycling Ratio Fraction correlation (*CyclingRatioFrac*) as a function of cycling ratio. The cycling ratio was determined by the on time ratio of the compressor during the field test.

$$CyclingRatioFrac = a + b(CyclingRatio) + c(CyclingRatio)^2 + d(CyclingRatio)^3 \quad (7)$$

Equations (6) and (7) are used to obtain the heat pump runtime fraction (*HPRTF*) used in Equation (8).

$$HPRTF = \frac{CyclingRatio}{CyclingRatioFrac} \quad (8)$$

Finally, the electric power consumption of VRF system is calculated using Equation (9).

$$PowerConsumption = \left(\frac{\dot{Q}_{OU,cooling,rated}\; CAPFT_{OU,cooling}}{COP_{cooling,reference}}\right)\left(EIRFT_{OU,cooling}\right)\left(EIRFPLR_{cooling}\right)\left(HPRTF\right) \quad (9)$$

Detailed information on calculating the total cooling capacity of the indoor units can be found in DOE's EnergyPlus documentation (2011). The rated cooling capacities of indoor units and outdoor unit are summarized in Table 3.

Table 3: Rated performance of VRF heat pump system

Unit	Outdoor Unit	Indoor Units		
		IU 1 and IU 6	IU 2 and IU 3	IU 4 and IU 5
Rated cooling capacity (kW)	22.4	2.2	3.6	5.6
Rated high air flow rate (m^3/min)	180	5.6	7.0	12.0

2.3 Weather Data and Schedules

Papa *et al.* (2007) concluded that outdoor temperature is the most important factor of air conditioning energy consumption. The changes of outside temperature account for 60 ~ 70% of building energy consumption (Yezioro *et al*., 2008). The actual outdoor dry-bulb temperature and relative humidity were measured on site and used as the weather data. Other actual 2011 weather conditions including solar radiation, solar illuminances, wind speed, wind direction, and atmospheric pressure, was obtained from the integration of a newly released NOAA historical data set for Arlington, Virginia (around 22.5 km southeast of College park, Maryland) (Weather Analytics, 2011). Information on the operation schedules (occupancy, lighting and equipment) and thermostat set-point were specified for more accurate simulation results. Although the operation schedules during the test period had a regular pattern, the thermostat set point temperature was varied occasionally during the experimental period. Since the set point temperature was recorded, it was reflected in the simulation for the accurate validation.

3. MODEL VALIDATION

VRF heat pump system charged with R410A was installed in educational offices and its field performance was tested from June to August in 2011. The actual outdoor conditions measured were used in the weather file, and daily, weekly, and monthly simulation results and experimental data were compared.

3.1 Experimental Set-up

Figure 3 shows the office suite with four rooms which was used for the field performance tests. Based on the building load estimation, two wall-mounted type indoor units were installed in Room B and Room C. A detailed schematic drawing of the system, measurement instrument, evaluation methodology and the accuracies of the sensors can be found in the previous work by Laeun *et al*. (2012). The error analysis of the experimental results was performed according to the propagation of the uncertainty (Kline and McClintock, 1953). Maximum uncertainty values of the cooling capacity of the VRF system, the cooling performance factor, and the power consumption are ±0.15 kW, ±0.76 and ±0.78 kW, respectively.

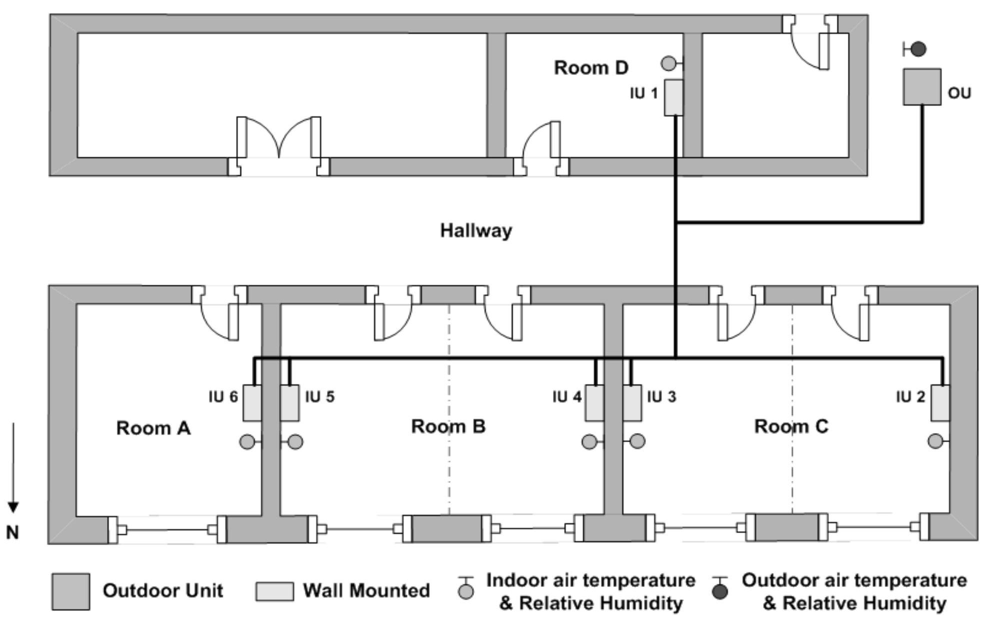
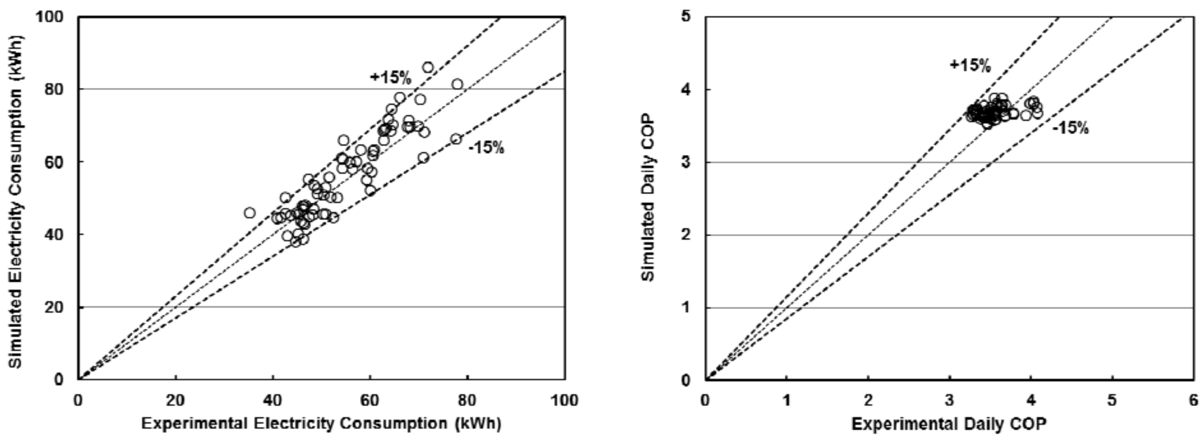
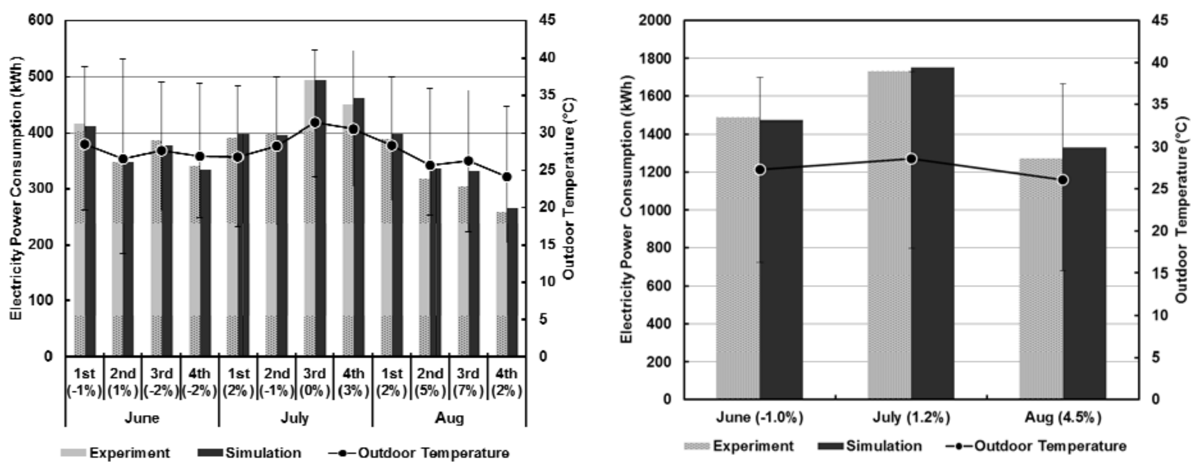


Figure 3: Layout of the office suite



(a) Daily electricity power consumption (b) Daily COP

Figure 4: Comparison between simulated results and experiment data



(a) Weekly electricity power consumption (b) Monthly electricity power consumption

Figure 5: Comparison between simulated results and experimental data

3.3 Weekly and Monthly Electricity Power Consumption

Since the long-term power consumption is more interesting than the short-term power consumption in the early design stage (Yezioro *et al*. 2008), the weekly and monthly electricity power consumptions were compared as shown in Figure 5. Band bar in Figure 5 (a) and (b) represents the temperature range for each week and month. Maximum errors in weekly and monthly electricity power consumptions are about 7.9% and 4.5%, respectively. The root-mean-square deviations of weekly and monthly electricity power consumptions for the total simulation period between the simulated and measured values are 11.12 kWh and 37.58 kWh, respectively. The averages of the absolute values of the weekly and monthly relative errors for the total simulation period are 2.40% and 2.22%, respectively. Comparison of the simulated and measured electricity power consumptions in a monthly basis shows significantly smaller deviations than those in the daily basis in terms of the mean of the absolute values. The reasons for small differences in monthly result are due to the factor that the underpredictions in some days compensate for overpredictions in other days resulting in an improved monthly comparison. Because of this reason, computed average results are generally in better agreement for longer time periods (Li *et al.*, 2010). Therefore, the root-mean-square deviations should be compared in parallel.

4. CONCLUSIONS

In this paper, VRF air-conditioning system in an academic office suite was numerically and experimentally investigated for the cooling season. The simulation study was performed by the newly developed building energy simulation program, EnergyPlus. Compared to the result obtained by the customized VRF heat pump simulation module (Zhou *et al*., 2008), the result of the current study shows a better agreement. This is because the cooling combination ratio correction factor and cooling cycling ratio fraction correlation in response to the cycling ratio were used in the current modeling work in order to correct the performance for off-design cases. The comparison of the simulated data and measured data was performed at three levels of time scale: daily, weekly and monthly. The root-mean-square deviation of daily, weekly and monthly electricity power consumptions for the total simulation period between the simulated and measured values are 5.63 kWh, 11.12 kWh and 37.58 kWh, respectively. The averages of the absolute values of the daily, weekly and monthly relative errors in electricity power consumption for the total simulation period are 7.97%, 2.40% and 2.22%, respectively. A better agreement between simulation and experimental results on a weekly and monthly basis was due to the reduced influences of uncertain factors.

NOMENCLATURE

CAPFT	Cooling Capacity Ratio Modifier Function of Temperature
CR	Cooling Combination
CyclingRatio	Cycling Ratio
CyclingRatioFrac	Cooling Part-Load Fraction Correlation Function of Cycling Ratio
EIRFT	Energy Input Ratio Modifier Function of Temperature
EIRFPLR	Energy Input Ration Modifier Function of Part Load Ratio
HPRTF	Heat Pump Run Time Fraction

Subscripts

OU	Outdoor Unit
min	minimum
c	outdoor dry-bulb temperature
wb ,avg	load-weighted average wet-bulb temperature

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