Multiple-Zone Variable Refrigerant Flow System Modeling and Equipment Performance Mapping

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ABSTRACT

We developed a variable refrigerant flow (VRF), vapor compression, system model, which has five indoor units, one outdoor unit and one water heater. The VRF system can run simultaneous space conditioning (cooling or heating) and water heating. The indoor units and outdoor unit use fin-&-tube coil heat exchangers, and the water heater uses a tube-in-tube heat exchanger. The fin-&-tube coil heat exchangers are modeled using a segment-by-segment approach, and the tube-in-tube water heater is modeled using a phase-by-phase approach. The compressor used is a variable-speed rotary design. We calibrated our model against a manufacturer's product literature. Based on the vapor compression system model, we investigated the methodology for generating VRF equipment performance maps, which can be used for energy simulations in TRNSYS and EnergyPlus, etc. In the study, the major independent variables for mapping are identified, and the deviations between the simplified performance map and the actual equipment system simulation are quantified.

INTRODUCTION

Variable refrigerant flow (VRF) space cooling and space heating units have been used widely in Asia for commercial and residential applications. A variable refrigerant flow system typically uses a variable-speed compressor or multiple compressors in the outdoor unit for capacity modulation. It has multiple indoor terminal units and requires no duct for circulating air, so it can also be called a ductless or multi-split system. The VRF system design is modular and flexible; adding multiple outdoor units can support indoor units for an entire commercial building. Goetzler (2007) stated that VRF systems are more energy efficient than a typical air-source, central ducted, heat pump system, due to elimination of the duct losses and its superior part-load performance. VRF systems provide better comfort since they match each zonal load individually.

Steady-state vapor compression system modeling has been quite an established area. There have been well validated and robust system models. These advanced vapor compression system modeling tools can be linked with complex component models, e.g. phase-by-phase heat exchanger models or segment-by-segment heat exchanger models. But most of these system simulation models are limited to single-stage vapor compression configurations, with a single condenser and evaporator pair. VRF system modeling can be challenging, since there are many variables to handle; for example, compressor speed and different indoor air dry and wet bulb temperatures in individual zones, etc. Since the indoor units are in parallel configuration, and they impact each other by connecting to the same compressor and outdoor heat exchanger, a simultaneous solver like Newton-Raphson is necessary for system solving. Integrating complex heat exchanger models to the Newton-Raphson system solver leads to more difficulties, since they will add many more equations, which are usually not amenable for simultaneous solving. Due to these complexities, open publications and research results for VRF system modeling using

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complex component models are still limited. In this study, we have enhanced our existing modeling capability to handle VRF multi-split systems, using advanced heat exchanger models, so as to simulate VRF space cooling, space heating, dedicated water heating and simultaneous space cooling and water heating modes.

Vapor compression system simulation models can be used to generate full- and part-load performance curves or tables for energy simulation software like EnergyPlus (2011) and TRNSYS (Klein 2010). The latest EnergyPlus release (version 6.0) already includes functionalities of space cooling and space heating of VRF systems, based on combinations of two-variable performance curves.

For modeling zone cooling capacity, EnergyPlus uses off-design performance curves to consider terminal unit capacity changing with ambient and indoor conditions as shown in Equation 1 and 2,

$$CAPFT_{coil,cooling} = a + b\left(T_{wb,i}\right) + c\left(T_{wb,i}\right)^2 + d\left(T_c\right) + e\left(T_c\right)^2 + f\left(T_{wb,i}\right)\left(T_c\right) \tag{1}$$

$$\dot{Q}_{coil(i),cooling,total} = \dot{Q}_{coil(i),cooling,rated}\left(CAPFT_{coil,cooling}\right) \tag{2}$$

Where $CAPFT_{coil,cooling}$ = Zone Coil Cooling Capacity Correction Factor (function of temperature); $\dot{Q}_{coil(i),cooling,total}$ = Zone terminal unit total (sensible + latent) cooling capacity [W]; $\dot{Q}_{coil(i),cooling,rated}$ = rated total (sensible + latent) cooling capacity in zone i (W); $T_{wb,i}$ = wet-bulb temperature of the air entering the cooling coil in zone i (°C); T_c = temperature of the air entering an air-cooled condenser (°C); and $a-f$ = bi-quadratic equation coefficients for Cooling Capacity Correction Factor.

For equations (1) and (2), the basic assumption is that individual terminal cooling capacity only depends on the terminal wet bulb temperature and outdoor temperature, and has no interactions with other terminals. The related control strategy could use electronic expansion valves to control a fixed superheat degree for each terminal unit. Variable speed compressor operation is used to maintain a relatively constant evaporating pressure, and so the indoor air flow would see a constant heat sink source.

In the EnergyPlus modeling approach, the heat pump unit energy input ratio (EIRFT) would depend on two inputs – weighted average wet-bulb temperature of the air entering all operating cooling coils and ambient temperature.

$$EIRFT_{cooling} = a + b(T_{wb,avg}) + c(T_{wb,avg})^2 + d(T_c) + e(T_c)^2 + f(T_{wb,avg})(T_c) \tag{3}$$

Where $T_{wb,avg} = \sum_{1}^{i}\left(T_{wb,i}\right)\left(\dot{Q}_{zone(i)} \big/ \dot{Q}_{zone,total}\right)$, the weighted average wet-bulb temperature of the air entering all operating cooling coils (°C), and $EIRFT_{cooling}$ = the cooling energy input ratio correction factor (function of temperature)

The format of energy performance curves in heating mode is similar to the cooling mode, simply replacing the zone, wet bulb temperature and the weighted average, indoor, wet-bulb temperature with the zone, dry bulb temperature and the weighted average, indoor, dry-bulb temperature.

We want to extend the curve-fitting methodology of EnergyPlus to performance look-up tables, using the same combination of independent variables. To verify the methodology, we need extensive VRF equipment performance data in a wide range of operating conditions, e.g. varied air and water flow rates, differed indoor dry bulb and wet bulb temperatures to individual terminals, multiple unit combination, and outdoor air temperature. Manufacturer may provide data for building energy design and equipment selection, however, the available data is limited, compared to what can be simulated in a system simulation model. Thus, we use our detailed vapor compression, VRF system simulation model to match a real product's performance at rated conditions, and then extend performance simulations to all the required ranges. The system configuration we simulated is a five-zone, VRF heat pump, with water heating capability, as shown in Figure 1. The indoor units and outdoor unit use fin-&-tube coil heat exchangers, and the water heater uses a tube-in-tube heat exchanger. The fin-&-tube coil heat exchangers are modeled using a segment-by-segment approach and the tube-in-tube water heater is modeled using a phase-by-phase approach. The compressor used is a variable-speed rotary design, which is modeled using a ten-coefficient compressor map based on ANSI/ARI Standard 540-2004, and we used linear interpolation between the speed levels. The simulation data was used as the basis to verify the EnergyPlus performance mapping methodology.

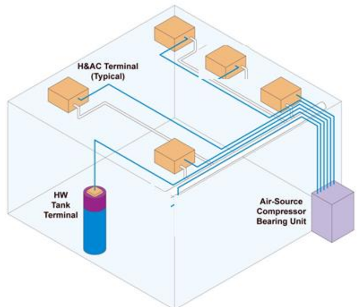

Figure 1 *Five Zone Variable Refrigerant Flow System*

MODEL CALIBRATIONS

Baseline System

We identified manufacturer's performance data for a VRF product having five indoor units and one outdoor unit, using a variable speed, rotary compressor; it is the closest match to our targeted configuration. However, the product literature doesn't include necessary information for modeling the heat exchangers and compressor. Also, the particular VRF heat pump unit chosen doesn't have water heating capability.

Our approach was to calibrate our VRF, vapor compression, system model using the product data at rating conditions so as to represent real product performance, and then add a tube-in-tube heat exchanger to the system for water heating performance simulations. Furthermore, we ran the calibrated system simulations over an extensive operating range to verify the VRF performance mapping methods used in EnergyPlus. Some basic information for the chosen baseline product can be seen in Table 1. All five indoor units have the same rated capacity. The unit combination table can be seen in Table 2, which describes capacity and input power variations with varying the number of indoor units.

Table 1. Specification of Baseline Unit

Specification	Indoor Unit	Outdoor Unit
Cooling Capacity [Btu/h]	7000.0	6300/30000/36000 (Min-Rated-Max)
Heating Capacity [Btu/h]	8000.0	7560/34500/41400 (Min-Rated-Max)
Rated Air Flow Rate [CFM]	177*	2119*
Sensible Heat Transfer Ratio (SHR)	70%	N/A
Fan Power [W]	30.0	130.0
Dimensions (W*H*D) [in]	33.1*10.6*6.0	34.3*31.8*12.6

*The indoor and outdoor air flow rates are assumed to be fixed at all compressor speeds. Unit Conversions: [Btu/h] = 0.293 [W]. [CFM] = 4.72e-4 [m^3/s]. [in] = 0.0254 [m]. [R] = 1/1.8 [C]. [°F] = 32 + 1.8 * [°C].

Table 2. Unit Combination Table

Number of Indoor Units	Cooling Capacity[Btu/h]	Cooling Power [W]	Cooling EER	Heating Capacity [Btu/h]	Heating Power [W]	Heating EER
1	7,000	740	9.5	8,000	900	8.9
2	14,000	1,232	11.4	16,800	1,562	10.8

3	21,000	1,862	11.3	25,200	1,998	12.6
4	28,000	2,435	11.5	33,100	2,619	12.6
5	30,000	2,386	12.6	34,500	2,250	15.3

The cooling capacities are rated at indoor DB/WB conditions of 80.6/66.2 °F and outdoor conditions of 95/75.2 °F DB/WB; the heating capacities are rated at indoor DB/WB conditions of 68/59 °F, and outdoor conditions of 44.6/42.8 °F DB/WB. We can see in Table 2 that there is not much difference in total capacity between four indoor units and five units (2.33 tons vs. 2.5 tons, respectively). This indicates that a 2.33 ton nominal cooling capacity with four operating indoor units might be a desirable design point to size the compressor. The cooling capacity (or heating capacity) per unit from one indoor unit to four indoor units are almost the same (7000 Btu/h for cooling, and 8000 Btu/h for heating), however, the capacity per unit with five operating indoor units is much less than the other combinations (6000 Btu/h for cooling, and 6900 Btu/h for heating). That implies that the control strategy of running five indoor units is most likely different from the other combinations, which may involve different suction and discharge saturation temperature settings.

Model Adjustments

From the indoor and outdoor unit dimensions, we can estimate the fin-&-tube coil heat exchanger sizes. We assume the indoor and outdoor heat exchanger dimensions in Table 3.

Table 3. Assumed Heat Exchanger Dimensions

Dimensions	Indoor Fin-Tube Coil	Outdoor Fin-Tube Coil
Rows	2	2
Number of Circuits	2	7
Flow Arrangement in Cooling Mode	Counter Cross Flow	Counter Cross Flow
Tube Number per Row	10	28
Tube Number per Circuit	10	8
Tube Length [in]	30.0	32.0
Fin Density [fins/ft] (determined by matching rated capacity)	160	190
Tube Pitch Vertical to Air Flow Direction [in]	1.0	1.0
Tube Pitch in Air Flow Direction	0.875	0.875
Frontal Air Flow Area [ft^2]	2.08	6.22
Tube Diameter [in]	0.276 OD/0.248 ID	0.276 OD/0.248 ID
Subcooling Degree [R]	10 (in heating mode)	10 (in cooling mode)
Superheat Degree [R]	10 (in cooling mode)	2 (in heating mode)

Regarding the compressor shell heat loss ratio, we assume 10% heat loss in cooling mode and 20% in heating mode. For line pressure drops, we assume 2 psiD [13.8 kPa] in suction line for cooling mode, and 2 psiD in discharge line for heating mode. The above numbers are rough guesses based on engineering rules of thumb. Based on these assumptions, we calibrated our heat exchanger models by changing the fin densities to match the rated capacity and SHR, which brought the heat exchanger model representations in line with the exact product sensible and total capacities. In addition, we assume there is a liquid receiver inside the outdoor unit, so as to maintain a constant subcooling degree for both heating and cooling modes, and minimum superheat degree for heating mode.

We used a 2.5-ton variable-speed rotary compressor map, which has four sets of mass flow rate and power curves, at speeds of 1800 RPM, 3600 RPM, 5400 RPM and 7200 RPM, respectively. However, this compressor map needs calibration, since it is not the same brand used in the baseline system.

We mainly used the manufacturer's data in cooling mode for adjusting the system model to match the product performance. Then, we used the same calibrated component models for heating mode predictions. Good matches of heating mode performances should justify the validity of our calibration practice.

<u>Space Cooling Mode (AC):</u>

1. At an indoor air flow rate of 177 cfm, an indoor DB/WB condition of 80.6/66.2F, and an evaporator, refrigerant liquid, entering temperature of 104 °F (i.e. condensing saturation temperature at 114 °F), we varied the evaporator fin density

and evaporating saturation temperature (compressor suction) to match the rated capacity of 7000 Btu/h and SHR of 70.0%. This gave the required fin density for the indoor heat exchanger and a suction saturation temperature of 50.5 °F.

2. Based on the calibrated, evaporator model, we set up four, identical, indoor units connected with the compressor (rated speed mode), and we varied the compressor, mass flow rate, calibration multiplier to match the required cooling capacity for the four units at the compressor speed of 5400 RPM (90 HZ), which is a typical speed for rating variable speed compressors. The controlled suction saturation temperature was fixed at 50.5 °F. Controlling suction pressure leads to relatively constant indoor capacity and SHR. This compressor mass flow rate calibration multiplier, which was used for all other speed levels, is essentially calibrating for the difference between displacement volume of the baseline VRF unit compressor and that of the compressor for which we had map data..
3. Based on the calibrated models above, we varied the outdoor, heat exchanger fin density to match a condensing saturation temperature at 114 °F, which is an assumed temperature for the rating point condition.
4. In cooling mode with four indoor units, based on the calibrated compressor, evaporator and condenser models, we varied the compressor power consumption calibration multiplier to match the rated EER at 5400 RPM.
5. The above four steps were iterated as needed to meet all the rating point targets.
6. We next ran three indoor units, then two indoor units, having the same indoor and outdoor conditions and flow rates as four indoor units. We varied the compressor speed to match the suction saturation temperature setting of 50.5 °F, and calibrated the power consumption multiplier at 3600 RPM, and 1800 RPM to match the EER at each unit combination.
7. For the combination of five indoor units, we kept the same indoor air flow rate for each unit and raised the evaporating temperature to match the required total capacity. The resultant suction saturation temperature was 54.5 °F and the required compressor speed was 5450 RPM. In this case, the sensible heat transfer ratio (SHR) is not controlled at 70%; instead, the resultant SHR is 81.5%.

<u>Space Heating Mode (AH):</u>

1. Using the same calibrated, component models and dimensions from cooling mode, we ran the combination of four indoor units, and varied the indoor air flow rate and condenser saturation temperature to match the required capacity and an assumed indoor supply air temperature of 94.5 °F. From this, we obtained a condensing temperature setting at 103 °F and an indoor air flow rate of 280 cfm. We then varied the compressor power multiplier at 7200 RPM to match the heating EER at this combination. The 4-unit indoor air flow rate is used for all other heating mode unit combinations, and the condensing temperature setting of 103 °F is used for indoor unit combinations of four and below.
2. We ran the combination of five indoor units, adjusting the condensing temperature to match the required heating capacity, and the calculated condensing temperature was 96 °F. In this case, the supply air temperature is not controlled at 94.5 °F; but rather, the resultant air supply temperature with the fixed indoor airflow assumption drops to 91.8 °F.

Figures 2 and 3 below present the comparisons between measured (product literature) and the predicted (model) efficiencies, based on the calibrated system model. Figure 2 shows good matches in cooling EERs, since these points were actually used for calibration. There is a noticeable deviation at one indoor unit, likely because the required compressor speed is below 1800 RPM (the lowest available compressor speed from the map) necessitating extrapolation of the map data to the lower speed. The heating EER comparisons in Figure 3 demonstrate good validity of the calibration method, as the good matches result from the same setup as in cooling mode. However, at the highest or lowest unit number, the agreement between the predicted and the measured EER values is poorer. At the lowest unit numbers, the lower bound of the compressor speed for map coefficients is exceeded. While at the combination of five indoor units in heating mode, it seems probable that there still may be some unaccounted operational factors.

Table 4 and 5 illustrates the compressor speed and saturation temperature variations with number of operating indoor units in cooling mode and heating mode, respectively. With fewer operating indoor units, the system refrigerant flow rate is reduced, and the heat transfer load in the outdoor exchanger is lessened. Consequently, we can see the discharge saturation temperature decreases in cooling mode, and the suction temperature increases in heating mode, as the number of indoor units drops. However, variations in the saturation temperatures diminish at a low number of indoor units, since the outdoor heat exchanger almost reaches thermal pinching in those cases.

Table 4. Compressor Speeds and Discharge Saturation Temperatures vs.Number of Active Indoor Units in Space Cooling Mode

Space Cooling at Rated Conditions	Compressor Speed [RPM]	Discharge Saturation Temperature [°F] -$T_{s,dis}$
One Indoor Unit	1366 ($T_{sat, suction}$ = 50.5 °F)	105.1
Two Indoor Units	2699 ($T_{sat, suction}$ = 50.5 °F)	105.3
Three Indoor Units	4057 ($T_{sat, suction}$ = 50.5 °F)	109.2
Four Indoor Units	5400 ($T_{sat, suction}$ = 50.5 °F)	114.0
Five Indoor Units	5450 ($T_{sat, suction}$ = 54.5 °F)	114.3

Table 5. Compressor Speeds and Suction Saturation Temperatures Changing vs.Number of Active Indoor Units in Space Heating Mode

Space Heating at Rated Conditions	Compressor Speed [RPM]	Suction Saturation Temperature [°F] - $T_{s,suc}$
One Indoor Unit	1696 ($T_{sat, discharge}$ = 103 °F)	38.0
Two Indoor Units	3283 ($T_{sat, discharge}$ = 103 °F)	37.0
Three Indoor Units	4729 ($T_{sat, discharge}$ = 103 °F)	35.2
Four Indoor Units	6138 ($T_{sat, discharge}$ = 103 °F)	33.3
Five Indoor Units	6881 ($T_{sat, discharge}$ = 96 °F)	32.1

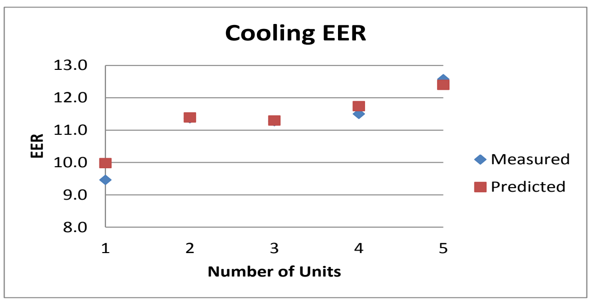

Figure 2 Comparisons between Measured and Predicted EERs in Cooling Mode

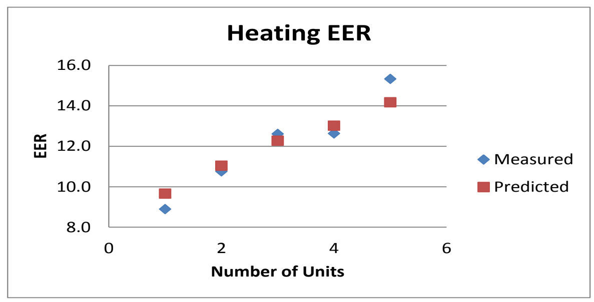

Figure 3 Comparisons between Measured and Predicted EERs in Heating Mode

WATER HEATING SIMULATIONS

We extended the calibrated system model to include water heating capability by adding a water-to-refrigerant tube-in-tube heat exchanger. The tube-in-tube heat exchanger uses fluted tubes, with refrigerant flowing outside and water flow inside, in counter flow arrangement. The fluted tubes are 11.42 feet (3.4 m) long, and have double flutes. The pump power is 40 W and water flow rate is controlled at 3 GPM. We fixed the water temperature entering the water heater as 110 °F, and used the same indoor and outdoor air flow rates as obtained in space cooling mode. For simultaneous space cooling and water heating mode, the indoor units are evaporators and the condenser is for water heating; for this mode, we varied the compressor speed to match the same suction saturation temperatures as in the cooling mode, i.e. 54.5 °F for five indoor units, and 50.5 °F for other combinations of indoor units. For dedicated water heating mode, the outdoor heat exchanger is the evaporator, and the water heater is the condenser. Table 6 shows the results of water heating for both modes. Regarding the dedicated water heating mode, we ran two cases: one has the outdoor condition at 44.6/42.8F DB/WB and the compressor speed is fixed at 5400 RPM; the other has the outdoor condition at 65/54.3 °F DB/WB and the compressor speed is fixed at 3000 RPM. For both the cases, the subcooling degree at the exit of the water heater is 10 °R.

Table 6. Water Heating Performance in Simultaneous Space Cooling and Water Heating Mode and Dedicated Water Heating Mode

Simultaneous space cooling and water heating	Speed [RPM]	$T_{s,dis}$ [°F]	Water Heating Capacity [Btu/h]	Space Cooling EER	Combined EER (AC+WH)
One Indoor Unit	1599	124.0	9731.8	7.9	18.6
Two Indoor Units	3227	129.7	19695.7	7.7	18.3
Three Indoor Units	4840	135.6	29412.3	7.8	18.5

Four Indoor Units	6667	141.5	39637.5	7.5	17.9
Five Indoor Units	6675	142.7	41582.8	7.7	18.3
Dedicated Water Heating	5400 fixed	135 °F $T_{s,dis}$/35 °F $T_{s,suc}$	27527.6	N/A	9.4
Dedicated Water Heating	3000 fixed	131 °F $T_{s,dis}$/55 °F $T_{s,suc}$	20184.9	N/A	12.4

We can see from Table 6 that the simultaneous space cooling and water heating mode demands higher compressor speed than the space cooling mode for the same number of active indoor units, due to its higher condensing temperature. The water heating capacity decreases with the number of active indoor units, since there is less heat obtained from the source and smaller refrigerant mass flow rate. For real VRF water heating situations, operation with only one active indoor unit may not be desirable since this may result in too low a compressor speed. Instead, we could use the dedicated mode here or, alternatively, could let the suction temperature rise to 54.5 °F or greater, so as to have a higher compressor speed.

VERIFICATION OF PERFORMANCE MAPPING METHODS

In order to verify the performance mapping method as shown in Equations 1 to 3, we conducted an exhaustive validation. Our validation involved running the above VRF system model for space cooling mode, space heating mode, and simultaneous space cooling and water heating mode. We choose the indoor unit number of four to represent the full mode.

Space Cooling Mode (AC):

We fixed the indoor, dry bulb temperature at 80 °F, and fixed the suction, saturation temperature at 50.5 °F (leading to 70% SHR), while allowing automatic adjustment of the compressor speed. We varied the indoor WB of each indoor terminal by five levels, from 60 °F (RH = 0.3) to 72.5 °F (RH = 0.7), and varied outdoor air temperature by five levels from 82 °F to 115 °F. We randomized the combinations of the outdoor DB and four, indoor, terminal WBs by a 5×5×5×5×5 = 3125 data point matrix.

Simultaneous space cooling and water heating mode (ACWH):

The parametric runs have the same indoor conditions and suction saturation temperature setting, except that we varied the water entering temperature to the water heater at five levels, from 82 °F to 115 °F. For the performance curve-fitting, we used similar forms as the space cooling mode, just replacing the temperature of the air entering an air-cooled condenser with the water temperature entering the water heater. It shall be noted that in ACWH mode, we only verified the curve-fitting accuracy for predicting terminal, cooling capacity and EIR of total power consumption to total cooling capacity. The water heating capacity and water heating EIR predictions should have similar accuracy, since the water heating capacity is just the sum of indoor capacities and the compressor power, considering compressor shell loss ratio.

Space Heating Mode (AH):

We varied the indoor dry bulb temperature at five levels, from 64 °F to 80 °F; and fixed the discharge saturation temperature at 103 °F, by automatically adjusting the compressor speed. We varied the outdoor air temperature by five levels, from 17 °F to 57 °F. We randomized the combinations of the outdoor DB and four indoor terminal DBs by a 5×5×5×5×5 = 3125 data point matrix.

Based on the comprehensive data set, we did curve-fits in the forms of Equations 1 to 3, and compared the values predicted by the curve-fits to the simulation data by the vapor compression system simulation model. Accuracy of the curve-fits would demonstrate validity of the performance mapping method. As shown in Table 7, the deviations between simple curve-fits and the system simulations are presented as relative errors in the form of (Curve-fit Predicted – Model Simulated Value)/Model Simulated Value. We show the maximum and minimum deviations, as well as the standard deviation (Stdev) to indicate the variation in all the relative deviations. It has to be noted that equations 1 to 3 have limitations in correlating dry heat transfer conditions for space cooling mode, since the indoor wet bulb temperature is not relevant for dry heat transfer. Thus, we separate the comparisons into two categories: one is for all the listed conditions, and the other only includes the conditions having wet heat transfer, i.e. SHR < 100%.

Table 7. Curve-Fit Accuracy for AC, ACWH and AH Modes

Performance Predictions	AC			ACWH			AH		
	Max	Min	Stdev	Max	Min	Stdev	Max	Min	Stdev

Terminal Cooling Capacity (All)	4.4%	-3.5%	2.2%	3.1%	-3.0%	1.5%	2.8%	-3.3%	1.0%
Terminal Cooling Capacity (Wet)	1.2%	-1.1%	0.3%	1.9%	-0.8%	0.3%	N/A	N/A	N/A
EIR (All)	5.1%	-5.3%	1.3%	14.8%	-4.9	1.9%	3.0%	-1.4%	0.5%
EIR (Wet)	2.7%	-2.7%	0.9%	2.1%	-3.3%	0.9%	N/A	N/A	N/A

DISCUSSION

As shown in Table 7, the curve-fitting accuracy for wet conditions is much better than the cases including dry heat transfer. Thus, the curve-fitting form in equations 1 to 3 should be used with care when the indoor humidity is relatively low. Although EnergyPlus has not used the curve-fitting method for simultaneous space cooling and water heating for VRF systems, the results in Table 7 prove that the same methods can be reused with confidence by simply replacing the temperature of the air entering an air-cooled condenser with the water temperature entering the water heater. Table 7 demonstrates that the curve-fitting accuracy for space heating mode is good for all the listed conditions.

It should be noted that we only generated data to verify the performance mapping methods at the combination of four indoor units. In Tables 4 and 5, we can see when the system capacity and indoor unit number requirements are larger than full mode (100%), the control strategies for setting the saturation temperatures are different, and the cooling or heating capacity per indoor unit is varied as well. Thus, it is necessary to introduce another independent variable for correlating the impact of varying the number or sizes of the indoor units. EnergyPlus uses the Unit Combination Ratio for this purpose; the combination ratio is defined as the total rated indoor cooling capacity (or heating capacity) of the active units divided by the rated heat pump cooling capacity of the outdoor unit (100% rated capacity), which also approximately indicates how many indoor units are on/off, as a ratio to total rated indoor unit number, if all the indoor units are identical. EnergyPlus uses the Combination Ratio Correction Factor Curve to correct the total unit cooling capacity (or heating capacity) only when the combination ratio is greater than or equal to 1. However, we believe that the combination ratio should be used in all the off-design performance calculations (even when the combination ratio is below 1), since the active unit number has a significant impact on the required compressor speed, and the compressor speed results in varied volumetric and isentropic compression efficiencies. As illustrated in Figures 2 and 3, the space cooling and heating EERs change with the number of indoor units, even when the number is below four (100% capacity). In order to provide a more accurate map in the full range, in the future we will investigate how to correlate the effect of indoor unit combination to the VRF energy performance mapping.

CONCLUSION

We developed a VRF, steady-state, system simulation model, which is linked with detailed segment-by-segment and phase-by-phase heat exchanger models. It has been calibrated against a real five-zone VRF product, and demonstrated good validity.

We used our VRF system model to generate a large amount of simulation data for verifying the performance mapping methods used in EnergyPlus. We proved that the curve-fitting methods are credible for space cooling, space heating, and simultaneous space cooling and water heating modes. However, special care needs to be taken if there is no dehumidification for indoor units in cooling mode. In addition, our current study is limited to one combination of indoor units. We believe that the indoor unit combination value shall be an important factor impacting VRF system performance mapping, and plan to investigate this effect in the future. And the simulation work is specific to the selected system configuration, and future effort can be used to model VRF systems with different configurations.

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