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A Variable Refrigerant Flow Heat Pump Computer Model in EnergyPlus

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ABSTRACT

This paper provides an overview of the variable refrigerant flow heat pump computer model included with the Department of Energy's EnergyPlus™ whole-building energy simulation software. The mathematical model for a variable refrigerant flow heat pump operating in cooling or heating mode, and a detailed model for the variable refrigerant flow direct-expansion (DX) cooling coil are described in detail.

INTRODUCTION

Through work sponsored by the United States Department of Energy (DOE) under Award Numbers DE-EE0003848, the Florida Solar Energy Center/University of Central Florida has implemented a computer model for a variable refrigerant flow (VRF) heat pump (HP) in the DOE's EnergyPlus™ whole-building energy simulation software (2011). DOE's EnergyPlus software (Crawley et al. 1999) builds on the strengths of the Building Loads and System Thermodynamics (BLAST Support Office 1992) and DOE-2 (Winklemann et al. 1993) computer simulation programs.

The VRF HP computer model allows multiple indoor terminal units to be connected to a single outdoor unit via refrigerant lines. Typically, only one variable-speed compressor modulates outdoor unit capacity to meet a varying load and this computer model employs a single variable-speed compressor. Although manufacturers allow connection of multiple outdoor units to accommodate larger capacity ranges, this characteristic is not modeled. When operating in cooling mode, the indoor terminal units and outdoor unit (compressor[s]) are controlled to maintain a low-side refrigerant pressure or temperature. In heating mode, the outdoor unit is controlled to a high-side refrigerant pressure or temperature.

In contrast, this computer model relies on empirical equations to define performance. Further operational details are not available since the control algorithms are proprietary.

This VRF HP computer model provides either cooling or heating and does not simulate heat recovery mode (i.e., simultaneous cooling and heating) since the operating performance in heat recovery mode is not well understood. A VRF heat recovery model is scheduled to be added to EnergyPlus in the near future. Modeling VRF systems is not new to the world of building energy simulation programs. A DOE-2 VRF function was previously created with guidance from VRF manufacturers and a subsequent application for adoption of this system type under Title 24-2005 was provided to the California Energy Commission (CEC) (Application 2008), however, the application process has stalled. The CEC application described the DOE-2 VRF model algorithm in detail. The DOE-2 VRF function uses empirical models based on observation and is similar in many respects to the model described here. The model described here represents the computer model implemented in EnergyPlus, not necessarily a model that accurately represents performance of VRF HP systems in actual installations. However, all attempts were made to ensure this first-generation model embodied the fundamental performance characteristics based on the limited knowledge of VRF HP systems in general.

Given the operational complexities of VRF HP systems, the model described here may have shortcomings that require further evaluation. For example, how does performance change when modeling multiple compressors? Is accuracy compromised given the method employed to limit zone coil capacity when insufficient system capacity exists? Does the VRF coil model developed for this VRF HP model accurately

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define coil performance? Is system part-load performance accurately modeled? These and other questions can only be answered with additional research. As part of this work effort sponsored through DOE, lab and field testing may provide additional insight into VRF HP system performance, and, ultimately, lead to a revised and improved model.

VARIABLE REFRIGERANT FLOW HEAT PUMP COMPUTER MODEL

The VRF HP computer model is a performance-based empirical model that describes several operating characteristics. The system capacity and power vary based on: (1) indoor and outdoor conditions, (2) the part-load performance of the heat pump's variable-speed compressor, (3) the combination ratio (CR), which is defined as the ratio of the total indoor terminal unit rated capacity and the total outdoor unit rated capacity, and (4) the losses associated with the refrigerant distribution piping. These performance characteristics are typically found in manufacturers' literature. Performance characteristics that generally apply to direct-expansion (DX) equipment and are identical to the EnergyPlus single-speed DX cooling coil model are not discussed in this paper (e.g., defrost and evaporatively-cooled condensers).

COOLING OPERATION

Modeling cooling performance begins by defining the model inputs as shown in Figure 1. Each zone terminal unit is then simulated to calculate the operating coil capacity. Specific calculations for terminal-unit cooling coils are described later in this paper. Piping losses are assumed constant throughout the simulation. The total cooling load is calculated as the sum of the zone cooling coil loads divided by the fractional cooling mode piping losses (Equation 1). Each zone coil can provide up to the maximum available coil capacity. A similar calculation is performed for zones that have a heating load.

$$\dot{Q}_{coil, total, cool} = \frac{\sum_{i=1}^n \dot{Q}_{coil(i), cool}}{PL_{correction, cool}} \quad (1)$$

If the total coil load is nonzero, the system operating mode is determined to be either cooling or heating; otherwise the outdoor unit and coils are off. The supply air fan may be programmed to operate when the system is off. The system operating mode is determined based on a user selection to monitor either a master thermostat, the largest total coil cooling or heating load, the number of coils requiring heating or

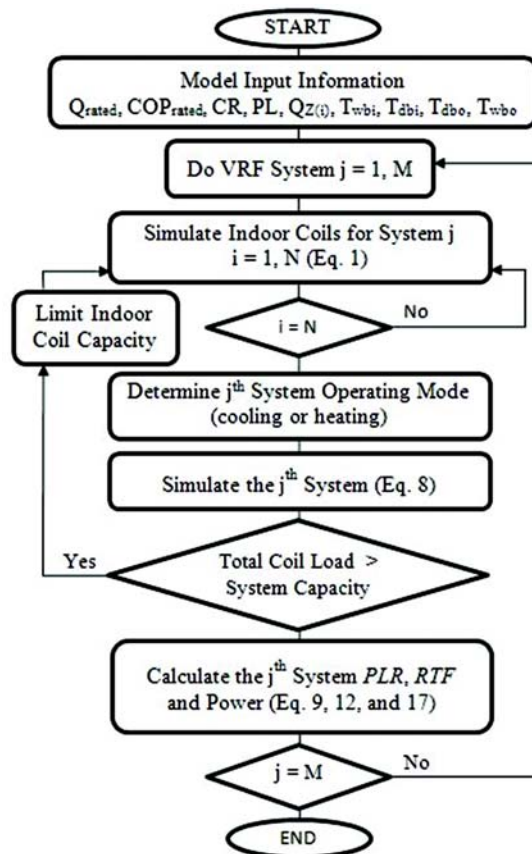


Figure 1 VRF HP computer-model flow chart.

cooling, or a schedule that determines operating mode. When the system operates, the total coil load, including the associated refrigerant piping losses, is assumed to be equivalent to the load imposed on the outdoor unit (i.e., the compressor[s]).

A VRF HP's available capacity and subsequent power consumption can vary significantly over a wide range of outdoor temperatures. For example, Figure 2 presents one manufacturer's cooling performance data and shows a distinct difference in performance at low and high outdoor temperatures. As shown in the figure, the VRF HP cooling-capacity ratio and cooling-power ratio (i.e., actual operating capacity/power divided by the reference capacity/power, where reference refers to the rated capacity at rated conditions) vary with indoor wet-bulb temperature and outdoor dry-bulb temperature. As outdoor temperature increases, cooling capacity decreases and cooling power increases which is a direct result of increasing head pressure on the compression system. Additionally, as indoor wet-bulb temperature increases, both capacity and power increase since an increase in the coil entering air wet-bulb temperature corresponds to a larger enthalpy difference across the cooling coil. When variable-speed compressors are employed, the available cooling capacity at lower outdoor temperatures can be held constant through controls (e.g., outdoor unit fan and compressor speed controls).

The VRF computer model directly simulates the change in capacity and power through empirical performance curves. The empirical performance curves representing capacity and power are a function of both the indoor coil and outdoor unit entering air temperature. The variation in performance with outdoor entering air temperature would be difficult to model using a single performance curve since performance over the range of outdoor conditions obviously has two distinct performance regions. It is not anticipated that more than two distinct performance regions will be encountered; therefore, the VRF computer model allows up to two performance curves and automatically determines which curve should be used based on a boundary curve separating the two performance regions.

Equation 2 represents an assumption that VRF cooling performance can be estimated using a load-weighted average coil entering air wet-bulb temperature. This average temperature is used as one of the two independent variables in the capacity and energy input performance equations (Equations 5 and 14).

$$T_{wb,avg} = \sum_1^n (T_{wb,i}) \left(\frac{\dot{Q}_{coil(i), cool}}{\dot{Q}_{coil, total, cool}} \right) \quad (2)$$

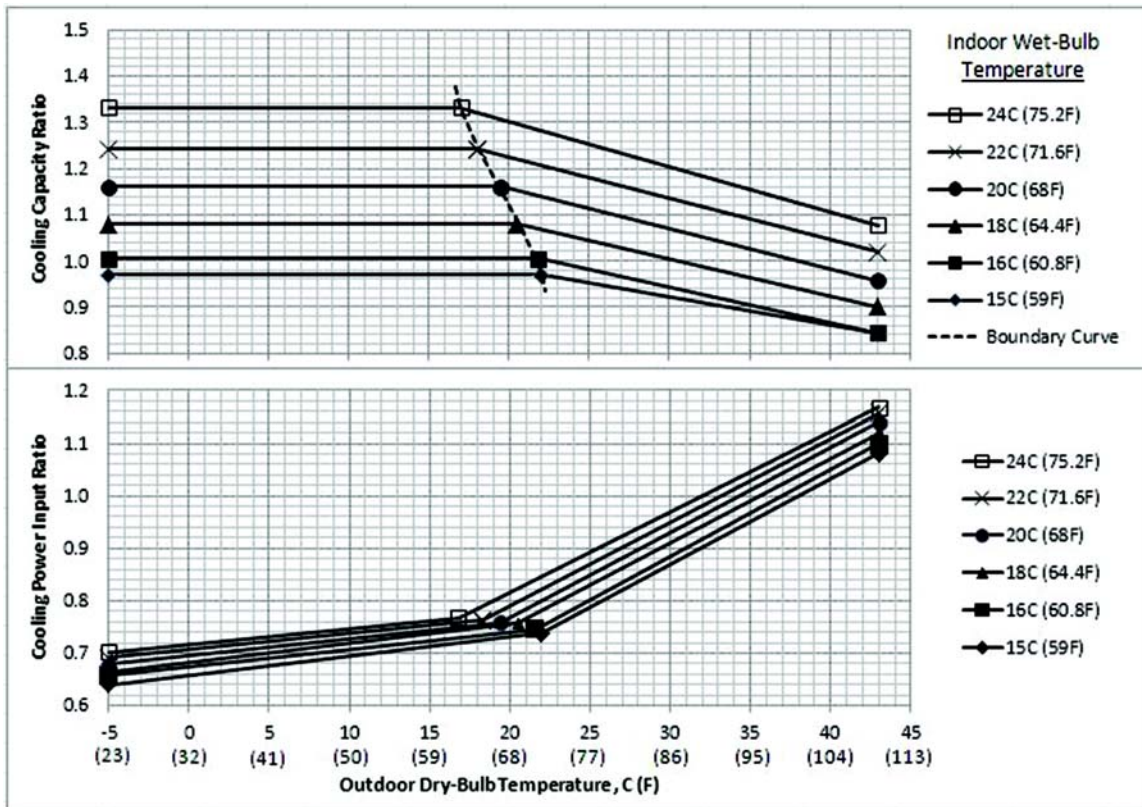


Figure 2 Example variable refrigerant flow cooling performance (Mitsubishi 2009)

$$T_{boundary} = a + b(T_{wb,avg}) + c(T_{wb,avg})^2 + d(T_{wb,avg})^3 \quad (3)$$

The boundary curve in Equation 3 calculates the outdoor air dry-bulb boundary temperature as a function of average coil entering air wet-bulb temperature where the change in performance occurs (see Figure 2). If the current outdoor dry-bulb temperature entering the outdoor unit coil (T_c) is less than or equal to the outdoor dry-bulb boundary temperature ($T_{boundary}$), the lower-temperature region performance curve is used, otherwise, the higher-temperature region curve is used. Coefficients a – d are determined based on a regression analysis of the manufacturers' data according to Equation 3 where the low- and high-temperature performance regions intersect. All subsequent equations presented in this paper also employ coefficients starting with the letter a ; this does not imply these coefficients are the same for different equations. The method used to specify VRF HP performance through regression analysis is well documented in literature and has been documented specifically for the VRF HP computer model (Raustad 2012).

The VRF HP outdoor unit capacity is modeled using a normalized capacity as a function of temperature (CAPFT) correction fraction (Equation 4). This same model is used for other DX equipment in EnergyPlus. For each specific indoor wet-bulb and outdoor dry-bulb temperature (j in Equation 4), the available cooling capacity is normalized to the reference cooling capacity, creating a capacity correction fraction as a function of temperatures. These capacity correction fractions, along with the operating conditions, define the cooling performance in both the low and high outdoor dry-bulb temperature regions. The coefficients a through f in Equation 5 are solved through regression analysis of these data.

$$CAPFT_{hp,cool,j} = \frac{\dot{Q}_{hp,cool,j}}{\dot{Q}_{hp,cool,ref}} \quad (4)$$

$$CAPFT_{hp,cool} = a + b(T_{wb,avg}) + c(T_{wb,avg})^2 + d(T_c) + e(T_c)^2 + f(T_{wb,avg})(T_c) \quad (5)$$

The total available cooling capacity provided by the VRF HP outdoor unit is also a function of the capacity of the indoor terminal units connected via the refrigerant piping (Equation 6). When the total indoor terminal unit reference cooling capacity is greater than the outdoor unit reference cooling capacity, the outdoor unit will not be able to meet the entire cooling demand. In this case, a combination ratio (CR) correction fraction is used to adjust the outdoor unit's available capacity. This model attribute allows indoor terminal units to be added or removed from the model without changing the outdoor unit.

Manufacturers typically provide CR performance information, and coefficients a – d in Equation 7 are solved through

regression analysis. The total available heat-pump cooling capacity can then be calculated.

$$CR_{cool} = \frac{\sum^n \dot{Q}_{coil(i),cool,ref}}{\dot{Q}_{hp,cool,ref}} \quad (6)$$

$$CR_{cool,correction} = a + b(CR_{cool}) + c(CR_{cool})^2 + d(CR_{cool})^3, CR_{cool} \geq 1 \quad (7)$$

$$\dot{Q}_{hp,total,cool} = \dot{Q}_{hp,cool,ref}(CAPFT_{hp,cool})(CR_{cool,correction}) \quad (8)$$

If the zone total cooling load (Equation 1) is greater than the available VRF HP cooling capacity (Equation 8), the computer model limits the available capacity of the zone coils with the highest loads such that the total coil demand is equal to the available capacity provided by the outdoor unit. The model assumes that terminal units with the lowest loads could increase the refrigerant flow rate to meet the same load without adversely affecting overall system performance. The model also assumes that the coils with the greatest loads impact the outdoor unit performance to a greater degree and, therefore, coil capacity is limited using a top-down approach. In addition to the limiting case described here, where the available VRF HP cooling capacity is unable to meet the zone total cooling load, there is also the possibility where the zone coil may not be able to meet the zone load. This situation is handled directly by the coil model, where the coil's capacity is limited by its maximum available output (see Equation 25).

COOLING ELECTRIC POWER CALCULATIONS

Using the previously calculated cooling requirement, the part-load ratio (PLR), runtime fraction (RTF), and VRF HP energy input can be determined. Equation 9 shows that PLR is calculated as the sum of the individual coil total capacities (sensible plus latent) divided by the available VRF HP cooling capacity (Equation 8). The VRF HP compressor continually operates (i.e., modulates) as long as the PLR is above the model input minimum limit (PLR_{min}). If the operating PLR is less than the specified minimum PLR, the VRF compressor will cycle on and off (Equation 10). When the VRF HP compressor cycles, a part-load correlation is used to account for cycling losses (Equation 11). Cycling losses impact energy use and are calculated based on the system RTF. The RTF defines the fractional amount of time the compressor must operate to overcome cycling losses (Equation 12). Similar to part-load ratio, which refers to the fractional load, the runtime fraction refers to the fractional time the compressor must operate to meet the fractional load. There are two restrictions imposed on the calculation of RTF. The cycling ratio fraction (CRatFrac) must be greater than or equal to 0.7 and CRatFrac

must be greater than or equal to the cycling ratio (CRat). These restrictions assure that RTF will not exceed 1. Since it takes a finite amount of time for a compression system to start up and reach steady-state output, RTF is always greater than or equal to the cycling ratio, and RTF is equal to 1 when $PLR \geq PLR_{min}$. Manufacturers do not typically provide the information required to develop Equation 11 through regression analysis. Coefficients *a* through *d* are more generally derived from laboratory testing. These coefficients could be derived from losses typical of single-speed DX equipment. Single-speed DX cooling computer models refer to this degradation as the part-load fraction correlation (DOE 2011).

$$PLR = \frac{\sum_{i=1}^n \dot{Q}_{coil(i),cool}}{\dot{Q}_{HP,total,cool}} \quad (9)$$

$$CRat = \frac{PLR}{PLR_{min}}, CRat \leq 1 \quad (10)$$

$$CRatFrac = a + b(CRat) + c(CRat)^2 + d(CRat)^3 \quad (11)$$

$$RTF_{hp} = \frac{CRat}{CRatFrac}, 0.7 \leq CRatFrac \quad (12)$$

and $CRatFrac \geq CRat$

The VRF HP outdoor unit energy input is modeled using a normalized energy input ratio as a function of temperature (EIRFT) correction fraction (Equation 13). Alternately, the normalized EIRFT can be calculated by dividing the power input ratio by the capacity input ratio (see Figure 2). A part-load term accounts for changes in the VRF compressor speed above the minimum compressor part-load ratio (Equation 15). When the zone coil's operate at part-load, the outdoor unit also operates at a lower part-load ratio (i.e., operates at a lower compressor speed). This in turn reduces energy use according to the part-load energy input ratio correlation (Equation 16). As previously discussed for capacity, the program allows up to two full-load energy input ratio (EIR) performance curves to be entered and automatically determines which curve should be used based on a boundary curve and the current operating conditions. The VRF HP's cooling energy input (Equation 17) is based on four distinct multipliers. The rated power, shown as the reference cooling capacity divided by the reference coefficient of performance (COP), is adjusted for changes in the operating capacity (CAPFT). This quotient is multiplied by the normalized EIRFT correction fraction. The first two terms combined yield the full-load power at the specific operating conditions. The impact of part-load performance and operating RTF are then included as the third and fourth terms, respectively. The VRF HP model uses two part-load power performance curves (EIRFPLR) since the slope of these curves change significantly at $PLR = 1$. Either the low or high PLR curve is used based on the operating PLR. Coefficients for

Equation's 14 and 16 are found through regression analysis of manufactures' performance data (using Equation's 13 and 15).

$$EIRFT_{hp,cool,j} = \frac{\frac{P_{hp,cool,j}}{\dot{Q}_{hp,cool,j}}}{\frac{P_{hp,cool,ref}}{\dot{Q}_{hp,cool,ref}}} \quad (13)$$

$$EIRFT_{hp,cool} = a + b(T_{wb,avg}) + c(T_{wb,avg})^2 + d(T_c) + e(T_c)^2 + f(T_{wb,avg})(T_c) \quad (14)$$

$$EIRFPLR_{hp,cool,PLR} = \frac{P_{hp,cool,PLR}}{P_{hp,cool,ref}}, PLR \geq PLR_{min} \quad (15)$$

$$EIRFPLR_{hp,cool} = a + b(PLR) + c(PLR)^2 + d(PLR)^3, PLR \geq PLR_{min} \quad (16)$$

$$P_{hp,cool} = \left(\frac{\dot{Q}_{cool,total,ref}(CAPFT_{hp,cool})}{COP_{cool,ref}} \right) \cdot (EIRFT_{hp,cool})(EIRFPLR_{hp,cool})(RTF_{hp}) \quad (17)$$

The equations previously presented for capacity (Equation's 4 and 5) and energy input ratio (Equation's 13–17) are nearly identical to a previously developed chiller model (Hydeman 2002). These empirical models are also similar to those describing other DX equipment models in EnergyPlus.

HEATING OPERATION

Heating operation is nearly identical to cooling operation although there is a subtle difference in the heating performance model. VRF HP heating performance is typically a function of indoor dry-bulb temperature and outdoor wet-bulb temperature. Some manufacturers may not provide performance data based on outdoor wet-bulb temperature and will instead provide this data based on outdoor dry-bulb temperature. In this case, all performance aspects may be modeled using outdoor dry-bulb temperature. For the CAPFT and EIRFT performance calculations when performance is specified as a function of outdoor wet-bulb temperature, the independent variables used for heating performance curves are indoor dry-bulb temperature and outdoor wet-bulb temperature. When performance is specified as a function of outdoor dry-bulb temperature, the independent variables used for heating performance curves are indoor dry-bulb temperature and outdoor dry-bulb temperature. Be consistent when selecting the outdoor temperature type.

VRF HP heating operation is modeled using a methodology similar to that described for cooling operation. Equations 1 through 7 are used identically to define heating performance.

Simply substitute the subscript “heat” for “cool” and substitute “db” for “wb”, as necessary, to identify the outdoor temperature property. Also remember to correctly choose the outdoor temperature type for the boundary curve dependent variable in Equation 3. Calculations for defrost operation are not presented in this paper and are described elsewhere (DOE 2011), however, the impact of defrost is shown in Equation 18 describing the available heating capacity. The only difference between Equations 8 and 18 is the use of a defrost correction fraction, which accounts for the change in heating capacity during defrost.

$$\dot{Q}_{hp,total,heat} = \dot{Q}_{heat,total,ref}(CAPFT_{hp,heat}) \cdot (CR_{heat,correction})(HeatCapFrac_{defrost}) \quad (18)$$

Heating electric energy input calculations are also nearly identical to those described for cooling. Equations 9 through 16 are used, again substituting the subscript “heat” for “cool” and “db” for “wb”, as necessary, in each equation to calculate heating power. The only difference between Equation 17 and 19 is the use of a defrost correction fraction, which accounts for the change in power during defrost.

$$P_{hp,heat} = \left(\frac{\dot{Q}_{heat,total,ref}(CAPFT_{hp,heat})}{COP_{heat,ref}} \right) \times (EIRFT_{hp,heat})(EIRFPLR_{hp,heat})(RTF_{hp}) \times (HeatPowFrac_{Defrost}) \quad (19)$$

As with the cooling operation, if the total zone heating load is greater than the total available heating capacity, the computer model limits the available heating capacity of the terminal units with the highest loads such that the total zone heating capacity equals the available capacity provided by the outdoor unit.

VARIABLE REFRIGERANT FLOW COOLING COIL MODEL

The original EnergyPlus DX cooling coil model is a single-speed compressor model that originated from DOE-2. This model was subsequently improved (Henderson 2000) and differences in these models are well documented (Kruis 2010). The VRF DX cooling coil model builds on the original EnergyPlus cooling coil model by modulating the coil capacity required to meet a specific zone load. The reference coil capacity is modified by a temperature-dependent term, which accounts for operating conditions different from the rating point.

The capacity of a DX cooling coil is primarily a function of entering air wet-bulb temperature. The outdoor conditions can also affect coil performance, but the impact outside conditions have on coil performance is more predominant in single-

speed compression systems. Since a variable-speed compressor can change speed to compensate for variations in outdoor weather, the VRF coil model is assumed to be primarily affected by indoor wet-bulb temperature. For this reason, the cooling coil's capacity as a function of temperature term (CAPFT) may be calculated using a linear, quadratic, or cubic equation form using only indoor wet-bulb temperature as the independent variable (Equation 20). For the EnergyPlus VRF coil model, this information is typically derived from manufacturers' data for outdoor unit capacity as a function of indoor coil entering air wet-bulb temperature. If additional information is available to allow the coil performance to be a function of both indoor wet-bulb and outdoor unit coil entering air temperature, a biquadratic form of the equation may be used (Equation 21).

$$CAPFT_{coil,cool} = a + b(T_{wb}) + c(T_{wb})^2 + d(T_{wb})^3 \quad (20)$$

$$\text{or} \quad CAPFT_{coil,cool} = a + b(T_{wb}) + c(T_{wb})^2 + d(T_c) + e(T_c)^2 + f(T_{wb})(T_c) \quad (21)$$

The model also accounts for off-design airflow through the coil. Although the current VRF DX coil model has a variable capacity, it utilizes a constant-speed fan component. The capacity as a function of flow fraction (CAPFF) term allows a cooling coil with some reference airflow rate to be operated at alternate flow rates. This does not imply use of a variable-speed fan, only the use of an alternate airflow rate that is different from the reference airflow rate. Although the terminal unit airflow rate would rarely deviate from the manufacturer's specified airflow rate, the term is available for those specific simulation characteristics. Given a range of flow fractions (Equation 22) and corresponding normalized capacity values, equation coefficients may be calculated (Equation 23). The total available cooling capacity is then calculated as shown in Equation 24.

$$ff = \text{flow fraction} = \left(\frac{\dot{m}}{\dot{m}_{ref}} \right) \quad (22)$$

$$CAPFF_{coil,cool} = a + b(ff) + c(ff)^2 + d(ff)^3 \quad (23)$$

$$\dot{Q}_{coil(i),cool} = \dot{Q}_{coil(i),ref}(CAPFT_{coil,cool})(CAPFF_{coil,cool}) \quad (24)$$

And finally, Equation 25 shows the total delivered capacity to the zone is calculated as the sum of the terminal unit components. The fan heat and outdoor air load are added to the modulated coil capacity and this sum, if sufficient cooling capacity is available, will equal the zone load. As previously described, if the total zone terminal unit cooling load is greater

than the total available outdoor unit cooling capacity, the computer model limits the available cooling capacity of the terminal units with the highest loads. After each coil is simulated, if the total coil capacity is greater than the available outdoor unit capacity, a maximum coil capacity (MaxCap) is calculated and used to limit coils with the largest loads such that the total zone terminal unit cooling capacity is equal to the available capacity provided by the VRF HP outdoor unit.

$$\dot{Q}_{zone(i),cool} = (\text{MIN}(\text{MaxCap}, \dot{Q}_{coil(i),cool}))(\text{PLR}) + \dot{Q}_{fan(i)} + \dot{Q}_{oa(i)} \quad (25)$$

At the time this model was developed, it was assumed that there was no lower limit on terminal unit PLR. In the future, a minimum operating PLR (or load) may be included to allow simulations where the terminal unit cycles off at low loads.

In addition to calculating the total cooling capacity provided by the DX cooling coil, it is important to properly determine the breakdown of total cooling capacity into its sensible and latent components. The model computes the sensible and latent split using the apparatus dew-point/bypass factor approach method. This method is analogous to the NTU-effectiveness calculations used for sensible-only heat exchangers (Henderson et al. 2000). The reference total capacity and reference sensible heat ratio (SHR) are first used to determine the reference slope of the air process line through the cooling coil (Equation 26).

$$\text{SlopeReference} = \left(\frac{w_{in} - w_{out}}{T_{db,in} - T_{db,out}} \right)_{ref} \quad (26)$$

Along with the reference entering air conditions, the algorithm then searches along the saturation curve of the psychrometric chart until the slope of the process line between the point on the saturation curve and the inlet air conditions match SlopeReference. Once at this point, the apparatus dew point (ADP) is found on the saturation curve and the coil bypass factor (BF) at the reference conditions is calculated as shown in Equation 27.

$$\text{BF}_{ref} = \frac{h_{out,ref} - h_{adp}}{h_{in,ref} - h_{adp}} \quad (27)$$

The coil bypass factor (Equation 28), by definition, is one minus the heat exchanger effectiveness for both the latent and sensible calculations and can be described in terms of the number of transfer units (NTU).

$$\text{BF} = 1 - \varepsilon = e^{-NTU} = e^{-\frac{\left(\frac{UA}{c_p}\right)}{\dot{m}}} = e^{-\frac{A_o}{\dot{m}}} \quad (28)$$

For a given coil geometry, the rated bypass factor is purely a function of air mass flow rate (Carrier 1967). The model calculates the parameter A_o in Equation 28 based on BF_{ref} and

the reference air mass flow rate. With A_o known, the coil BF can be determined for nonreference airflow rates.

The VRF DX cooling coil model calculates the total cooling capacity and coil bypass factor at the actual operating conditions. The coil bypass factor is used to calculate the coil surface temperature (Equation 30), which in turn is used to calculate the operating sensible heat ratio (SHR) using Equation 31. Here is where the difference in computer models occurs between the VRF DX cooling coil and the original single-speed DX cooling coil model (Henderson et al. 1992). The original coil model calculates the coil's full-load enthalpy difference (i.e., total cooling capacity divided by air mass flow rate) and, considering the bypass factor, finds the coil surface condition (h_{ADP}) at full load (i.e., $\text{PLR} = 1$) using Equation 29. Conversely, the VRF cooling coil model modulates the VRF DX cooling coil capacity, hence the use of the full-load coil capacity multiplied by the part-load ratio as shown in Equation 30. This effectively finds that the coil surface condition for varying DX cooling coil loads and the operating SHR can be calculated using Equation 31.

Single-Speed DX Coil Model (h_{adp1} in Figure 3)

$$h_{adp} = h_{in} - \frac{\left(\frac{\dot{Q}_{coil(i),cool}}{\dot{m}} \right)}{1 - \text{BF}} \quad (29)$$

VRF DX Coil Model (h_{adp1-3} in Figure 3)

$$h_{adp} = h_{in} - \frac{\left(\frac{\dot{Q}_{coil(i),cool}(\text{PLR})}{\dot{m}} \right)}{1 - \text{BF}} \quad (30)$$

$$\text{SHR} = \left(\frac{h_{Tin,wadp} - h_{adp}}{h_{in} - h_{adp}} \right) \quad (31)$$

Using this SHR, the properties of the air leaving the cooling coil are calculated using Equations 32–35.

$$h_{out} = h_{in} - \left(\frac{\dot{Q}_{coil(i),total,cool} \times \text{PLR}}{\dot{m}} \right) \quad (32)$$

$$h_{Tin,\omega out} = h_{in} - (1 - \text{SHR})(h_{in} - h_{out}) \quad (33)$$

$$w_{out} = f(T_{in}, h_{Tin,\omega adp}) \quad (34)$$

$$T_{db,out} = f(h_{out}, w_{out}) \quad (35)$$

Figure 3 shows this process on a psychrometric chart. The VRF DX cooling coil model follows the dotted process line from the coil inlet condition (h_{in}) toward the outlet air condition. The coil surface temperature is found by drawing a straight line through these points. The process line from h_{in} to h_{adp1} represents the full-load ($\text{PLR} = 1$) process line from coil

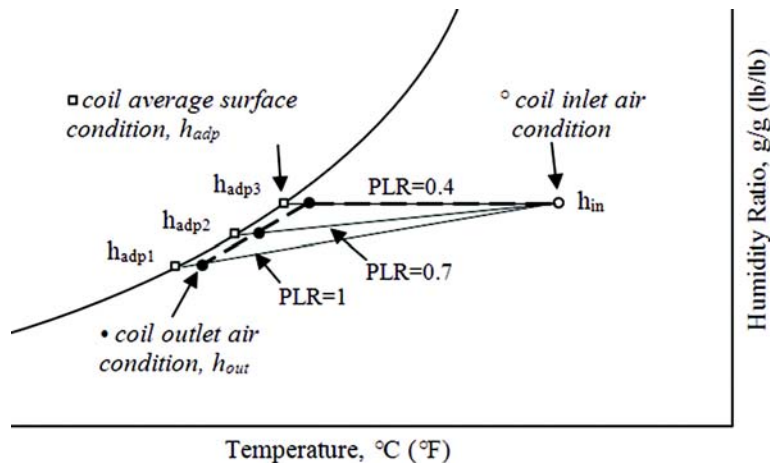


Figure 3 Variable refrigerant flow DX cooling coil process example.

inlet to the coil surface condition. The outlet air condition is then calculated as a point on this line and located accordingly using BF (Equation 30). At this point the coil is fully loaded (PLR = 1) and the sensible heat ratio is at the design point (assuming h_{in} is the reference point and the coil operates at the reference airflow rate). As the coil load is reduced, the refrigerant flow rate is restricted and the outlet air condition rides up the dotted line. For example purposes, the outlet air condition and associated h_{adp2} is shown for a PLR of 0.7. Here the sensible heat ratio is higher than that found at full-load operation. As the load continues to reduce, the refrigerant flow rate continues to throttle back, and there comes a point where the coil's ADP is equal to the inlet air dew-point temperature (h_{adp3}). At this point, and for all other PLRs less than this value, the majority of the coil surface becomes dry (at PLR = 0.4 in this example) and the coil's sensible heat ratio equals 1. From this point to PLR = 0, the coil outlet air condition follows the dotted line back toward h_{in} (i.e., at PLR = 0, $h_{out} = h_{in}$).

VARIABLE REFRIGERANT FLOW HEATING COIL MODEL

VRF heating coil model calculations are nearly identical to those described for the VRF cooling coil. The only difference is that the heating coil has only a sensible component and the sensible heat ratio (SHR) is always 1. The model calculations are the same as used for the EnergyPlus single-speed DX heating coil as described in the EnergyPlus engineering reference (DOE 2011).

SUMMARY AND DISCUSSION

The EnergyPlus VRF HP computer model is an empirical equation fit model based on manufacturers' performance data. This model uses equations similar in form to other DX equipment computer models used in EnergyPlus. The VRF HP operates in either cooling or heating mode and, at the time this paper was written, does not currently support heat recovery. Preliminary work has been performed to validate the VRF HP

computer model and shows that simulation results compare well to manufacturers' data. Efforts are also underway to compare the VRF HP computer model to a field demonstration. Additional work is warranted to fully understand the interactions of multiple indoor-terminal units connected to a single outdoor unit, the impact of different control algorithms, and how performance changes when heat recovery mode is active. In conclusion, this specific VRF model is in an infancy stage and may evolve over time as additional performance data become available from manufacturers, field demonstrations, or laboratory tests.

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NOMENCLATURE

ADP	=	apparatus dew point
$a-f$	=	equation coefficients determined through regression analysis (unique for each equation)
BF	=	bypass factor
COP	=	coefficient of performance
DX	=	direct expansion cooling system
ε	=	effectiveness
h	=	enthalpy, J/kg (Btu/lb)
HP	=	heat pump
\dot{m}	=	mass flow rate, kg/s (lb/h)
P	=	power, W (Btu/h)
PL	=	pipng loss due to refrigerant tubing
PLR	=	part-load ratio
\dot{Q}	=	rate of heat transfer, W (Btu/h)
RTF	=	runtime fraction
T	=	temperature, °C (°F)
UA	=	heat transfer coefficient, W/K (Btu/h°F)
VRF	=	variable refrigerant flow
w	=	moist air humidity ratio, g/g (lb/lb)

SUBSCRIPT

adp	=	apparatus dew point
avg	=	average
db	=	air dry-bulb
c	=	outdoor unit coil
$coil$	=	indoor unit coil
$cool$	=	cooling mode
fan	=	zone supply air fan
hp	=	heat pump
i	=	i^{th} coil
j	=	j^{th} performance metric
in	=	process inlet state point
min	=	minimum
oa	=	outdoor air
out	=	process outlet state point
ref	=	reference or rated performance parameter
wb	=	air wet-bulb

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DISCUSSION

Paul Doppel, Senior Director Industry and Government Relations, Mitsubishi Electric, Suwanee, GA: This is a positive comment for Richard's work on trying to expand the understanding of how VRF systems are tested, how they operate in the field, and how to best model them. His efforts have led to expanded efforts to work actively on evaluation of VRF systems.

Richard Raustad: I wish to acknowledge the five manufacturers that supported this work. These manufacturers actually co-sponsored this effort through co-funding and also provided guidance as to the actual operation of this system type. Two of these manufacturers were present in the laboratory during the

performance testing phase of the project. Each of these manufacturers provided feedback on testing methodology and measured system performance. Of course, further research is necessary to eventually create a robust and technically sound VRF computer model. These efforts are ongoing.

Neal Krus, Engineer, Big Ladder Software, Boulder, CO:

Are you using the average wet bulb because you are not modeling each evaporator separately? Is this average weighted in any way? ASHRAE is developing a standard (205) to get manufacturers' data into the hands of energy modelers; it would be good to discuss this and get your input.

Richard Raustad: Since this is an empirical performance model, only the resulting air-side capacity and system power are modeled. This result is based purely on the average coil entering air wet-bulb temperature and outdoor air temperature

(when cooling mode is active). The refrigerant properties are not modeled, which means the model does not have access to refrigerant-side characteristics such as evaporator suction temperature or pressure. I suppose that some form of suction temperature control (or high-side control for heating) could be added to the existing model (i.e., the model would attempt to attain some leaving air temperature). In the current model, the average wet-bulb temperature is weighted by the coil load to capacity ratio so that if there were a 1000 W load in one zone and a 1 W load in another zone, given each zone contained a 1000 W coil, then the coil entering air wet-bulb temperature in the zone with a 1000 W load will be predominant. Equation 2 in the paper describes the load-weighted average coil entering air wet-bulb temperature.