CPUC Ex-Ante Team

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# Summary

The object of this investigation was to perform simulation-based comparisons between variable refrigerant flow (VRF) systems and conventional heating and air conditioning systems. Representative DEER building prototype models were developed for four building types with a range of configurations of both conventional systems and variable refrigerant flow systems. Expanded capabilities to allow more accurate modeling of the range of commonly installed VRF systems were developed and implemented into the building energy use simulation software tool used for previous DEER analysis – eQUEST using DOE-2.3. The models used for the conventional packaged heat pump systems were also improved to ensure comparison with the VRF systems are reasonable and represent current expected comparative performance. An energy use and demand performance comparison analysis was then performed for climates across California.

# General Scope and Modeling Methods

## Scope of Study

The simulation study encompassed four building types, all 16 California climate zones, and 6 system configurations. The four building types, listed in Table 1, were selected as typical applications for VRF technology that cover a range of load conditions.

Table 1 Building Types for VRF Assessment



### VRF Technology Definitions

VRF technologies were developed based on VRF outdoor unit system size as shown in Table 1. The outdoor unit efficiency is the only parameter that changed between the VRF baselines and the VRF measures.

Table 2 VRF Measure Levels vs System Capacity

|  |  |  |  |  |
| --- | --- | --- | --- | --- |
|  | System | AHRI Rated EER | | |
| Building Type | Tons | Title 24 | Tier 1 | Tier 2 |
| Hotel Guest Room | 8 | 11.0 | 13.0 | 15.0 |
| Small Office | 14 | 10.6 | 12.0 | 13.5 |
| Primary School, Hotel Public Areas | 18 | 10.6 | 12.0 | 13.5 |
| Large Office | 20 | 9.5 | 11.5 | 13.0 |

### Alternate-Technology Baseline Definitions

Several different alternate-technology baselines were evaluated in order to demonstrate the relative performance of VRF technologies compared to non-VRF systems, as shown in Table 3.

Table 3 Alternate-Technology Baselines for VRF Measures

|  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- |
|  |  | Building Type Applicability | | | |
|  |  | Small Office | Primary School | Large Office | Hotel |
| All Electric Title-24 Baselines | |  |  |  |  |
| 1. | Packaged single zone heat pump, (< 65 kBtuh) without economizer | X | X | X |  |
| 2. | Packaged single zone heat pump, (< 65 kBtuh) with economizer | X | X | X | X |
| 3. | Packaged single zone heat pump, (65 to 135 kBtuh), with economizer | X | X | X | X |
| Fuel Heat Title-24 Baselines | |  |  |  |  |
| 1. | Packaged single zone AC/furnace, (< 65 kBtuh) without economizer | X | X |  |  |
| 2. | Packaged single zone AC/furnace, (< 65 kBtuh) with economizer | X | X |  |  |
| 3. | Packaged single zone AC/furnace, (65 to 135 kBtuh), with economizer | X | X |  |  |
| 4. | Packaged variable air volume AC/ gas boiler, with economizer | X | X |  |  |
| 5. | Four pipe fan coil system with water-cooled chiller and gas boiler |  |  | X | X |
| Fuel Heat High Performance Baselines (for Three-Prong Test) | |  |  |  |  |
| 1. | Packaged single zone AC/furnace, (< 65 kBtuh) with economizer | X | X |  |  |
| 2. | Packaged single zone AC/furnace, (65 to 135 kBtuh), with economizer | X | X |  |  |
| 3. | Packaged variable air volume AC/ gas boiler, with economizer | X | X |  |  |
| 4. | Four pipe fan coil system with water-cooled chiller and gas boiler |  |  | X | X |

#### All-Electric Reference Runs

Key parameters for each of the all-electric baselines are listed below.

1. Packaged single zone heat pump with economizer (<65 kBtuh)

* SEER 14, HSPF 7.7 (hotel guest room: EER 10.7, COP 3.1)
* One speed fan
* Fan power 0.294 W/cfm (hotel guest room: 0.125 W/cfm)
* No economizer

1. Packaged single zone heat pump without economizer (<65 kBtuh)

* SEER 14, HSPF 7.7 (hotel guest room: EER 10.7, COP 3.1)
* One speed fan
* Fan power 0.294 W/cfm (hotel guest room: 0.125 W/cfm)
* Air economizer (except hotel guest room)

1. Packaged single zone heat pump with economizer (65 to 135 kBtuh)

* EER 10.8, COP 3.3 (hotel guest room: EER 10.7, COP 3.1)
* Two speed fan
* Fan power 0.4 W/cfm (hotel guest room: 0.125 W/cfm)
* Air economizer (except hotel guest room)

In addition to the three Title-24 heat pump baselines, two high performance heat pump alternates were run as follows:

1. Packaged single zone heat pump with economizer (<65 kBtuh)

* SEER 17, HSPF 9.0 (hotel guest room: EER 11.9, COP 3.7)
* Two speed fan
* Fan power 0.271 W/cfm (hotel guest room: 0.125 W/cfm)
* No economizer

1. Packaged single zone heat pump with economizer (65 to 135 kBtuh)

* EER 10.8, COP 3.3 (hotel guest room: EER 11.9, COP 3.1)
* Two speed fan
* Fan power 0.4 W/cfm (hotel guest room: 0.125 W/cfm)
* Air economizer (except hotel guest room)

system most building areas were equipped with a two speed unit rated at SEER 18 and HSPF 9.7. For hotel guest rooms, a high performance packaged terminal heat pump rated at 11.9 EER/3.7 COP was utilized.

The systems were configured with one thermal zone per heat pump unit in order to prevent simultaneous heating and cooling. With the exception of the hotel guest rooms, all units were assumed to be larger than 54 kBtu/hr, thus requiring outside air economizers.

The conventional heat pump alternates included in this assessment are identical to those used for the DEER 2015 release, except for the following changes:

1. A problem was found in the bypass factor performance curves in the DEER 2015 heat pump simulations. This was resolved by changing to the default bypass factor curves from the DOE2 library.
2. One significant change from DEER2015 relates to frost control of the heat pump outdoor coil during heating mode. Defrost heat for the conventional heat pumps in this study was provided by the unit operating in a reverse cycle (DEER2015 assumed electric resistance). Moreover, the assumed defrost control scenario operates on a cycling period and defrost run time that are both based on demand (DEER2015 assumed a time initiated/temperature terminated control). In addition, the DOE2 algorithm for frost control for conventional heat pumps was updated for DOE2.3. The details of this update are summarized in Appendix C.
3. One minor control change for heat pump operation relates to the temperature above which supplementary electric resistance heat was disabled. In DEER2015, this value was defaulted to 40°F, but this resulted in some hours with insufficient heating capacity. For the current assessment, the value has been changed to 52°F. The DOE-2.3 model gives the heat pump cycle first priority as long as it is enabled, and only allows the electric resistance heat to provide additional capacity when the heat pump is running at 100%.
4. Heat pumps measures were not updated in DEER 2015 for systems in the 65 to 135 kBtuh range, so DEER 2014[[1]](#footnote-2) was used as the source for these baselines. The heating performance curves from DEER 2014 did not account for changes in air temperature entering the indoor coil, and this was resulting in unreasonable heating energy calculations. To resolve this, heat pump performance curves from DEER 2015 for the 2 speed heat pumps in the 0 to 65 kBtuh category were utilized.

A noteworthy assumption that is unchanged from DEER2015 is the method of ventilation control for two speed heat pump units. The ventilation rate is assumed to be held constant as supply air flow is adjusted. This provides a steady supply of fresh air to the conditioned spaces so as to ensure comparable indoor air quality during operation when compared with a VRF system using a dedicated outdoor air system, which also supplies a steady outdoor air quantity with respect to terminal unit supply air flow.

#### Gas-Heat Reference Runs - Title-24

Key parameters for each of the baselines with fuel heat are listed below.

1. Packaged single zone air conditioner with economizer (<65 kBtuh)

* SEER 14, Furnace 80%
* One speed fan
* Fan power 0.294 W/cfm
* No economizer

1. Packaged single zone air conditioner without economizer (<65 kBtuh)

* SEER 14, Furnace 80%
* One speed fan
* Fan power 0.294 W/cfm
* Air economizer

1. Packaged single zone air conditioner with economizer (65 to 135 kBtuh)

* EER 10.8, Furnace 80%
* Two speed fan
* Fan power 0.4 W/cfm
* Air economizer

1. Packaged VAV with gas boiler

* EER 10.0, Boiler 80%
* Variable speed fan
* Fan power 0.72 W/cfm
* Air economizer

1. Four pipe fan coil with water cooled chiller and gas boiler

* Chiller 0.56 kW/ton
* Boiler 80% efficiency
* Two speed fans, except hotel guest rooms are one speed
* Fan power same as VRF
* Dedicated outside air system with 0.32 W/cfm fan power
* No economizer

#### Gas-Heat Reference Runs - High Performance

Any comparison of VRF with systems that use fuel heating must use the CPUC three-prong test, which is listed in Appendix E . The CPUC three-prong test for fuel-substitution programs requires that the proposed technology not increase source BTU consumption as calculated using the current CEC-established heat rate. The latest value for the heat rate is 7,760 Btu/kWh, and the following formula is used to calculate the source energy:

Source Energy = [Building Site Electric kWh] X [7,760 Btu/kWh] + [Building Site Fuel Btu]

Another key element of the CPUC three-prong test is that the technology offered by a program must be compared with the most efficient cost-effective technology available that uses the fuel that is to be substituted with electricity. Cost-effectiveness is defined as having a TRC and PAC benefit-cost ratio of 1.0 or greater.

Key parameters for each of the high performance fuel heat baselines required for the three-prong test are listed below.

1. Packaged single zone air conditioner with economizer (<65 kBtuh)

* SEER 18, Furnace 80%
* Two speed fan
* Fan power 0.271 W/cfm
* Air economizer

1. Packaged single zone air conditioner with economizer (65 to 135 kBtuh)

* EER 13, Furnace 80%
* Two speed fan
* Fan power 0.4 W/cfm
* Air economizer

1. Packaged VAV with gas boiler

* EER 12.5, Condensing Boiler 95%
* Variable speed fan
* Fan power 0.72 W/cfm
* Air economizer

1. Four pipe fan coil with water cooled chiller and gas boiler

* Chiller 0.476 kW/ton
* Condensing Boiler 95% efficiency
* Two speed fans, except hotel guest rooms are one speed
* Fan power same as VRF
* Dedicated outside air system with 0.32 W/cfm fan power
* No economizer

## DEER Prototypes

This study was performed using building models derived from the DEER 2015 prototypes[[2]](#footnote-3), since that version of DEER was the last one in which SEER rated heat pumps were updated. The starting point for the assessment was a set of DOE2 input files generated by MASControl version 3.00.27[[3]](#footnote-4). The enhanced VRF capabilities were only added to DOE2.3, thus some minor modifications were needed to the DEER 2015 non-residential prototypes in order to be able to be utilized with DOE2.3 version. DOE-2.3 was utilized for DEER2017 and DEER2018 for the residential modeling but the non-residential modeling was not updated for that release of DEER. The building shell and non-HVAC potions of the prototypes were generally kept unchanged from the DEER 2015 version for comparability. However, one significant change was made for the small office building: the interior walls that border the core zone were changed from massless air-walls to uninsulated frame walls to be more realistic in terms of interior heat transfer.

## Simulation Management Tool

A Microsoft Excel workbook tool was used to manage the simulations required for this study. The tool contains tables of data to define the heat pump reference cases and the VRF alternates. Macro routines in the tool are used to create new input files, run simulations and extract results for each model variation. Additional description of the simulation tool is provided in Appendix A.

## System Sizing

DOE2.3 includes a number of features that improve the ability to calculate design HVAC sizes. These include:

* The addition of alternate weekend design day schedules to capture morning start-up loads after a weekend.
* New zone keywords to limit the temperature rise per hour that can be delivered by a system during startup.
* Improved accuracy when sizing of air flows due to iterative calculation process accounting for inter-zonal heat transfer.
* Better handling of coincident loads due to multiple steps to sizing process, starting with zone air flows and ending with central equipment.
* Definition of multiple design days for cooling and multiple days for heating in order to represent different design conditions for different months. This more accurately captures the interplay between weather conditions and sun angles, which change differently through the year.

Based on the methods described in the 2016 T24 Nonresidential Compliance Manual, the new sizing features in DOE2.3 were applied in this assessment using system sizing factors of 1.1 to account for "unexpected loads or changes in space usage"[[4]](#footnote-5). The same section of the compliance manual also indicates the use of 0.5% cooling drybulb with mean coincident wet bulb temperature for cooling design conditions, and winter median of extremes for heating design. Monthly design day weather values fitting these criteria were calculated from the CZ2010 weather data files[[5]](#footnote-6) Since the weather files represent a single typical year, the data were not sufficient for calculation of a winter median of extremes for heating. In lieu of this, the monthly 0.5% heating drybulb temperature was used. The simulations utilized cooling design days for the months of July through October and heating design days for November through January. Detailed tables of the design day data are listed in Appendix B.

# Variable Refrigerant Flow Energy Model

The VRF scenarios were evaluated using newly expanded capabilities in the DOE2.3 energy simulation program. Previous versions of the program included a heat pump only algorithm for VRF systems. The latest version also handles VRF systems operating in heat recovery mode, and a new component has been added to the program to model dedicated outdoor air systems.

The VRF routines in DOE2.3 are component-based, with separate calculations for the compressor, the outdoor heat exchanger coils, the indoor heat exchanger coils and the piping. There are new keywords to allow specification of key values to determine the sizing and operation of each of the components. This component based analysis provides more accurate assessment of indoor and outdoor coil control strategies, and the ability to accurately model piping energy impacts.

## Outdoor Units

The outdoor units in the VRF model are comprised of the compressors, the outdoor heat exchanger coils and the trunk refrigerant piping.

### System Configuration

The program allows multiple equally sized compressors to be specified for one system. The program also allows a multiplier to used in a VRF outdoor unit model to represent an appropriate system size when the model is abstracted with large zones and systems. This enables piping loops to be realistically sized and evaluated by the program.

### Compressor Performance Curves

One challenge in developing and using the component based VRF model has been establishing appropriate compressor performance data. Existing engineering data published by the manufacturers list performance in terms of outdoor dry bulb temperature and indoor coil entering air temperature. This approach does not support the component based model, which requires compressor performance as a function of refrigerant operating temperatures.

One reference point for addressing the issue of compressor performance is the recently published "physics based" EnergyPlus model for variable refrigerant flow, which is uses a component model very similar to the DOE2.3 version. The EnergyPlus release includes a sample run file[[6]](#footnote-7), which contains a set of three compressor curves for a 12 ton compressor at three different operating speeds. Another reference point that was found was performance data for variable speed compressors manufactured by Danfoss[[7]](#footnote-8). Figure 1 shows the performance of the EnergyPlus compressor along with two different sizes of Danfoss compressors and the 2-speed conventional heat pump. Note that the smaller Danfoss compressor is more in line with typical VRF outdoor units, which normally use compressors in the 6 to 8 ton range. It should also be noted that, while the Danfoss compressor can operate as low as 15% part load ratio, the operating temperatures at those speeds appear to be quite limited. Thus, for the current assessment, the 6.6 ton Danfoss performance curve was used, and the minimum speed was set to 25%. The curves in Figure 1 reflect a minimum speed of 25%, and curve values below that speed are based on the application of cycling loss calculations.



Figure 1 Sample Part Load Performance Curves

### Outdoor Unit Rated Efficiency

In order to characterize the performance of the outdoor unit, the rated efficiency of the unit must be specified along with the refrigerant temperatures and outdoor temperatures that correspond to the rated efficiency. Most manufacturers publish a unit EER based on AHRI rating conditions, which are listed in Table 3. A detailed review of EER values from the AHRI product database[[8]](#footnote-9) reveals a significant trend of decreasing efficiency as system capacity increases, as shown in Figure 2. One explanation for this trend is the requirement in the AHRI test procedure to use longer refrigerant pipe runs for higher capacity systems (Table 3). Another explanation is the tendency towards higher air flow per unit capacity in the outdoor heat exchanger for smaller units. This is likely related to the larger form factor per unit capacity for smaller units, as well as the tendency for larger units to be comprised of smaller units that are attached together side-by-side, thus blocking some pathways for air flow.

Table 5 AHRI Rating Conditions for Commercial VRF Systems[[9]](#footnote-10)

|  |  |  |  |  |  |  |
| --- | --- | --- | --- | --- | --- | --- |
| Capacity kBtu/hr | Capacity Tons | Pipe Length | Outdoor Fan | Indoor Fan | Outdoor Temperature | Indoor Temperature |
| 65 to 105 | 5.4 to11.3 | 50 ft | Included in EER calc. | Included in EER calc. | 95°F db  75°F wb | 80°F db  67°F wb |
| 106 to 134 | 11.3 to 20 | 75 ft |
| 135 to 350 | 20 to 29.2 | 100 ft |
| > 350 | > 29.2 | 150 ft |



Figure 2 Rated EER vs. Capacity for Heat Recovery VRF Systems in AHRI Database



Figure 3 Efficiency Versus Capacity for Two VSD Scroll Compressor Manufacturers[[10]](#footnote-11),[[11]](#footnote-12)

For this analysis, it was assumed that the efficiency characteristics of compressors in VRF systems do not change with unit capacity. This is supported by a set of compressor performance data collected for two manufacturers of variable speed scroll compressors , shown in Figure 3. If the compressors are assumed to be constant, then the operating conditions must vary to account for the trend of decreasing efficiency with increasing capacity. The compressor performance curves were utilized in a spreadsheet analysis to determine appropriate suction and discharge temperatures that would balance system capacities at the AHRI rated indoor and outdoor temperatures. The results of this procedure are listed in Table 4. A summary of manufacturer data used for this assessment is provided in Appendix D.

Table 6 Rated Refrigerant Temperatures and EIR vs. Capacity for VRF Outdoor Units



### Outdoor Unit Fan

Other rated conditions relate to the airflow and power of the outdoor fan. Figure 3 shows outdoor fan data for four manufacturers. The fan EIR is defined as the ratio of fan power to rated unit capacity. For the 6 to 10 ton units, which are usually single compressor systems, there is a clear trend of decreasing air flow with increasing capacity. This generally makes sense, since larger units would tend to have less surface area per unit volume of the cabinet.

For mid-sized units (typically 10 to 14 tons), some manufacturers interconnect smaller units, thus blocking airflow from the joined sides. Larger units are typically comprised of groups of smaller units mounted on a curb with a gap between them, so the reported air flow is usually referenced to the model numbers of the smaller units.

While the fan EIR values in the right-hand chart of Figure 3 show some trend of decreasing with increasing capacity, the trend was not as strong as for the air flow. Thus, a straight average of all the EIR data (0.034) was used for the fan power calculations in the current study.



Figure 4 Outdoor Fan Airflow and Power Ratio vs. Capacity

Modeled air flow values that were used in the present analysis are listed in Table 1. Air flow for 6 to 14 ton units was based on averages of the manufacturer data for those bins. For larger units, a straight average of the smaller units was used, since they can be built from various combinations of the smaller units.

Table 7 Rated Airflow per Unit Rated Capacity for VRF Study



### Outdoor Unit Control

The new DOE2.3 VRF routines have two options for controlling systems that include multiple compressors. The first option is to run the compressors together as much as possible to minimize motor speed, and the second is to run compressors in stages to minimize the number of units running. Based on review of the operations manuals for a couple of different manufacturers, the control strategy selected for the present assessment was running units together. It is important to note that there is some staging that occurs with this control strategy when systems are at low load levels. The control will only run two or more compressors if they can all be operated continuously. If the load drops below the minimum turn-down ratio of the compressors, then one compressor will drop off and the speeds of the remaining compressors will ramp up, as shown in Table 2. Also, the transition point for an increasing load is set at 20% above the minimum compressor speed in order to prevent short cycling (Table 3).

Table 8 Compressors Running Together with Decreasing Load for Compressors with 25% Minimum Speed



Table 9 Compressors Running Together with Increasing Load for Compressors with 25% Minimum Speed



There are also two means by which refrigerant temperatures can be controlled for the outdoor units. The first is a limit control of the fan on the outdoor unit. When the unit is operating in a cooling dominated mode, the fan runs at full speed as long as the condensing refrigerant temperature is above 60°F. When the condensing temperature hits this limit, the fan modulates to keep it from falling further. Similarly, in a heating dominated mode, the refrigerant temperature is not allowed to rise above 52°F.

The second control is a refrigerant temperature reset based on the demands of the indoor coils. For cooling coils, the suction temperature setpoint a the coil can be reset from the design value of 44°F to a maximum of 50°F. For heating coils, the discharge temperature at the coils can be reset from the design value of 115°F down to a minimum of 105°F.

The actual operating refrigerant temperatures can be different from the setpoints depending on the total system requirements. For example, there may be zones requiring cooling during winter that are requesting a suction temperature of 50°F, but the dominant mode is heating. In order to satisfy the heating loads, the system must draw heat from 40°F outdoor air, resulting in an actual operating suction temperature of 30°F.

### Defrost Model

The defrost model for the VRF system is based on many of the same principles as the model for the conventional heat pump. One significant difference between the two is how heat is provided during the defrost cycle. For the conventional heat pump, the outdoor coil is heated by a reversed refrigeration cycle during which the indoor coil is evaporating. In order to prevent the evaporating indoor coil from cooling the indoor zones, the supplemental heating coil is run. Thus there is a component of compressor power and a component of electric resistant heat power. For the VRF system, the indoor coils are not affected by the defrost process. Instead, the outdoor coil is split into two circuits: one that is evaporating and one that is condensing. This is effectively a hot gas bypass operation, and the outdoor fan is off during defrost, so the evaporating coil transfers heat by natural convection. Additional details regarding the VRF defrost model are provided in Appendix C.

## Indoor Units

### Indoor Unit Types

There are several different types of indoor units available from manufacturers, but they can generally be classified as surface mounted or ducted. The fan power consumption used in this study were taken from a review of published values from three different manufacturers. The results of this review are summarized in Table 4. For the small and large office building models, the zones represent a combination of activity areas, so a mixed unit type category was used, which is an average of the ducted and surface mounted types.

Table 10 Type of VRF Indoor Unit Terminal by Activity Area



### Control of the Indoor Unit

Indoor units can be controlled either with adjustments to fan speed or supply temperature. For this assessment all activity areas except hotel guest rooms were assumed to use variable speed and variable temperature control. As load decreases from 100%, the indoor unit initially responds by decreasing air flow down to a minimum level. This minimum was set to 72% based on typical minimum flow values published in manufacturer literature. As load decreases further, the indoor unit begins to increase the supply air temperature (in cooling mode) to modulate the capacity. This is managed by control of an electronic thermal expansion valve in the unit.

Hotel guest rooms were assumed to use constant speed fans that cycle in proportion to space load. This constant speed assumption was a constant between the conventional heat pump and the VRF systems.

### Outdoor Air Conditioning and Delivery

All VRF system options were run in conjunction with dedicated outdoor air systems (DOAS). These systems provide constant outdoor air to zones whenever they are in a normally occupied mode of operation. Three DOAS configurations were considered: unconditioned, conditioned and energy recovery. For the unconditioned option, outdoor air is delivered by the DOAS to the mixed air section of the VRF indoor unit without any conditioning. For the conditioned option, the DOAS contains a VRF coil that tempers the air to 55°F for a heating condition and 75°F for a cooling condition. For the energy recovery option, an enthalpy wheel is used to recover energy from exhaust air before delivering to the mixed air section of the indoor unit. The energy recovery option does not include a VRF coil in the DOAS unit. All three DOAS options include a supply fan and ductwork to deliver the outdoor air. The heat recovery option also includes a return fan in order to provide balanced air flow. The assumed values for fan power for each of the systems are listed in Table 5.

Table 11 Fan Power for Dedicated Outdoor Air Systems



The hotel guest rooms were not directly provided with outdoor air in the models, with the assumption that outdoor air for those spaces would be taken as transfer air from the corridors.

## VRF System Design and Layout

The new VRF model in DOE2.3 includes detailed calculations of pipe loss that include both friction loss and thermal loss. These losses depend on both the length and the diameter of the pipes in the system. In order to accommodate a range of piping configurations, three different categories of pipe can be defined. The Leader pipe is attached to the outdoor unit, and is defined as the section of the main trunk that ends at the first branch to an indoor unit, as illustrated in Figure 3. The header pipe is the section of the main trunk along which branches to indoor units occur. The branch pipes lead from the header to the individual indoor units, and the model allows each indoor unit to have its own unique branch length and diameter. Any branch can be located either outdoors or in a particular zone for heat loss calculations, but normally the leader is outdoors, and the other pipes are in a plenum.



Figure 5 Components of VRF Piping System in DOE2.3

Detailed layout concepts were developed for each of the building types in this study. Attention was paid to key parameters that affect energy performance, such as number and sizes of systems, length of piping loops, and branching configurations. These are described in the following sections, with tables of key values.

Zoning in the DEER prototypes often use abstraction to combine multiple actual control zones in the building that have identical characteristics (i.e. interior load, envelope and orientation) into a single zone for the model. The DOE2.3 inputs allow the definition of multipliers on both indoor units and outdoor units in order to size those units and their corresponding piping loops accurately in light of this abstraction. While pipe lengths are included as part of the model inputs, the unit capacities and pipe diameters are determined by the program.

### Small Office Building

The small office prototype model is a two story building with 5,000 square feet per floor. The layout concept used for this analysis is shown in Figure 3. The system consists of one system serving each floor, and each system has two piping trunks that run through the building. The branches from the trunks are relatively short, with an average of 10 feet. The outdoor units are located on the ground, and there is a leader pipe for each trunk that runs outdoors for about 20 feet. The indoor portion of the trunk, referred to as the header in the model, is 65 feet. To account for friction through elbows and other fittings, each of these values is increased by 5 additional feet for the model.

Each system is assumed to have a rated capacity of 14 tons, and is comprised of two 7 ton sub-units. Thus, the rated EIR and refrigerant temperatures listed in Table 1 for a 14 ton unit are used for the small office VRF models.

Since the DEER small office model is abstracted relative to Figure 3, the configuration of the indoor units does not match the figure exactly. The number of indoor units is established to reach a target of approximately 450 sq ft per indoor unit, which is equivalent to about 3 private offices. This results in 11 indoor units per floor, which will be sized at approximately 1 to 2 tons per unit, depending on the climate.

The DOE2.3 piping model can only have one trunk line for each system. In order to handle the two trunks in the small office prototype, the system is split into two separate systems in the model. The efficiency and rated conditions are still based on a 14 ton system, and there are still two outdoor units per system. The system split is accomplished with the NUM-SYSTEMS keyword, which allows a larger group of zones to be attached to a system, while the modeled size is representative of the physical system (in this case the piping system). Connecting a larger group of zones to a system ensures that the heat recovery benefits of the VRF system are accounted for when some zones are in heating and others are in cooling.



Figure 6 Conceptual Layout of VRF System for Small Office Building

### Large Office Building

The large office prototype is a ten story building with 17,500 sq ft per floor. The assumed layout for this analysis consists of two VRF systems per floor, each with a rated capacity of 20 tons. The units are located in the center of the building as illustrated in Figure 6, with trunks running along the centerline. Branches from the trunk extend to core zones and perimeter zones with lengths of 25 ft and 45 ft, respectively.

The capacities of the indoor units for the large office model are based on the mix of activity areas on a given floor. For open areas such as lobbies and open offices, larger systems in the range of 3 to 5 tons are assumed. For private offices and other small rooms, smaller units on the order of 1 to 2 tons are assumed. There are 24 indoor units in the model for the first floor and 32 units per floor for each of the middle and upper floors.



Figure 7 Conceptual Layout of VRF System for Large Office Building

### Primary School Building

The primary school prototype consists of classrooms, a gymnasium, a cafeteria and a kitchen. System layouts for each of these areas are presented in Figure 5 through Figure 7. A summary of key geometry and capacity values is listed in Table 6. The predominant activity area in the model is the classroom, which has one surface mounted indoor unit per room.

Table 12 Key System Design Parameters for Primary School VRF Systems





Figure 8 VRF Layout for Classroom Wing of Primary School Prototype



Figure 9 VRF Layout for Gymnasium of Primary School Prototype



Figure 10 VRF Layout for Cafeteria and Kitchen of Primary School Prototype

### Hotel Building

Design parameters for the hotel prototype were established based on the actual construction documents from a recently designed building. The outdoor units that serve the guest rooms are all located on the roof of the building, and pipe chases feed down to each floor. For this analysis, it is assumed that two outdoor units serve each floor, with a typical system capacity of 8 tons. Systems that serve hotel guest rooms are limited in capacity due to risk of suffocation if the system were to leak into a guest room. Public areas in the model are located on the first floor of the building, and the outdoor units for these systems are assumed to be located on the ground away from the building. Thus, there is minimal height difference between the indoor and outdoor units for the public areas, but a greater outdoor trunk length.

Table 13 Key System Design Parameters for Hotel VRF Systems





Figure 11 VRF Layout for Typical Guest Room Floor of Hotel Prototype



Figure 12 VRF Layout for Public Floor of Hotel Prototype

# Results

## Parametric Testing

In order to understand the differences in energy consumption between the conventional heat pump systems and the VRF systems, a series of parametric simulation tests were conducted. These tests were run for the large office prototype, and the location selected for the tests was CZ16 in order to emphasize impacts on both cooling and heating. Table 9 shows the results of these tests, which include peak period kW and annual kWh for each HVAC end use. The first two simulations are for the conventional heat pump system, with Run 1.0 being the baseline model. Runs 2.0 through 2.7 represent a series of changes to the model implemented cumulatively, i.e. each run is a modification of the previous run. The objective of the cumulative changes is to modify the VRF model such that Run 2.7 has similar performance characteristics to Run 1.1, but to keep the model for Run 2.7 configured with the DOE2.3 VRF system type.

**Run 1.0: Conventional HP Baseline**

The first result in Table 9 is the high performance conventional heat pump, which is a two speed unit with SEER 18 and HSPF 9.7.

**Run 1.1: Remove Economizer for Conventional HP**

Run 1.1 removes air side economizers from the conventional heat pump model. This is the largest factor governing the differences between the conventional heat pump and the VRF systems. Removing the economizer results in a 49% increase in annual cooling energy, and a 15% decrease in fan energy.

Table 14 Parametric Testing for Large Office Building in CZ16



**Run 2.0: VRF Test Baseline: Heat Recovery VRF with Conditioned DOAS**

The reference run for the VRF tests is the heat recovery VRF system with a DOAS that provides unconditioned outdoor air to the indoor units.

**Run 2.1: Reduce Piping**

For this alternate the VRF refrigerant piping was reduced from approximately 175 feet of distance between the indoor units and the outdoor units down to 25 feet. The thermal losses are more relevant for heating, because heating energy is connected to the discharge piping, which is almost always significantly above ambient temperature. Fan energy is affected because refrigerant temperatures at the indoor unit are more favorable with shorter piping runs. The overall HVAC energy savings due to the piping reduction is 2.4%.

**Run 2.2: Reduce Outdoor Unit Typical Size**

The typical outdoor unit size for the VRF systems in the large office building was assumed to be 20 tons. For this test case, the unit size was reduced to 8 tons. This has the effect of decreasing the unit EIR and altering the rated refrigerant temperatures for the unit to less extreme levels as listed in Table 1, with an overall HVAC energy benefit of 1%.

**Run 2.3: Increase Minimum Part Load Ratio of Outdoor Unit to 50%**

The VRF baseline assumes a minimum part load ratio of 25%. Since the VRF part load curve used for this analysis is more efficient at 50% load (as shown in Figure 1), this test results in a reduction in compressor power during both heating and cooling. Total HVAC energy reduction compared to Run 2.2 is 5%.

**Run 2.4: Change Indoor Unit from Variable Speed to Two-Speed**

The baseline VRF indoor unit is controlled with a variable volume/variable temperature scheme. As load decreases from maximum capacity, the system first responds by reducing air flow down to the minimum setting, while supply air temperature is held constant. Once air flow reaches the minimum, the supply air temperature is reset to maintain space temperature. This test changes the variable speed fan to a two speed fan. The result is more operation at full speed, with a greater degree of temperature reset. The test also decreases the minimum flow ratio of the fan at low speed from 0.72 to 0.66 to match the value used for conventional heat pumps. The net effect is a fan energy penalty of about 16%, but only 3% total HVAC penalty due to the beneficial temperature reset effects on heating and cooling.

**Run 2.5: Remove DOAS Fan Power**

The dedicated outdoor air system contains an additional fan and duct network that does not exist in the conventional heat pump system. Removal of the DOAS fan energy in the model reduces the annual VRF fan energy by 39%. There is a corresponding increase in heating load and a decrease in cooling load as a result of the reduced fan heat in the air stream, with overall HVAC decrease of 13%.

**Run 2.6: Increase Indoor Unit Fan Power**

It was assumed for the large office that indoor units would be evenly split between surface mounted units with relatively low fan power, and ducted units with fan power similar to the conventional heat pump. For Run 2.6, fan power was changed to match the conventional heat pump. The result is a 36% increase in fan energy and 9% increase in HVAC overall.

**Run 2.7: Remove Indoor Unit Reset Controls**

The heat recovery VRF baseline includes reset control of the refrigerant temperature setpoint based on zone coil loads. For this test, the reset controls are removed, so refrigerant temperatures are always held at their design temperatures for heating and cooling, as long as there is at least one zone in heating or cooling mode, respectively. Without reset controls, fans can be turned down to low speed more frequently, but the compressor runs less efficiently due to the more severe refrigerant setpoints. Thus, there is a fan benefit, but heating and cooling penalties. The net effect is a penalty of about 15% of annual HVAC energy.

**Run 2.7 vs Run 1.1: Parametric Comparison**

The cumulative changes that build up to Run 2.7 result in a VRF model with key performance parameters that are very similar to those in the conventional heat pump in Run 1.1. Consequently, heating, cooling and fan energy are relatively close in those simulations. One difference that persists is the use of supplemental heat for the conventional heat pump, both at colder outdoor temperatures and for makeup during defrost. Another difference is the heat recovery capability of the VRF system, which results in free heating or cooling when the building has some zones in heating and others in cooling. A third difference relates to cycling operation. When the conventional heat pump operates with loads below the low speed capacity, the compressor cycles. When it is cycled on, the unit operates with a relatively low coil surface temperature, which tends to remove excessive moisture, increasing the latent load. The indoor coil of the VRF system, on the other hand, can run continuously at low loads by modulating the electronic expansion valve. This gives a relatively higher surface temperature and lower latent load.

Appendix A: Detailed Model Parameters

The workbook RunHPComps.xlsm was used to manage simulations and organize input data for the VRF assessment. In this appendix the contents of the key worksheets of the workbook are described.

* Main: Contains locations of key files and controls for running the simulations
* Matrix: Tables that define the run variations are included here
* ODU\_Effic: This table lists rated conditions for each outdoor unit size.
* VRF1: Lists outdoor unit parameters that do not change from one system to the next.
* VRF2: Lists indoor unit parameters that change from one unit to another. These are mostly dependent on building type and/or activity areas served by the indoor unit.
* VRF3: Lists outdoor unit parameters that change from one unit to another. Mostly dependent on building type, but may also depend on activity areas.
* KwdProc: Lists keywords that need to be reset relative to the raw input files that come from DEER 2015. Most of these are SYSTEM keywords. Separate groups of values are applicable to PTHP systems.
* DOASdflt: Contains keywords for the dedicated outdoor air systems (DOAS).
* DOAS1: Makes assignments of condensing units to specific DOAS units.
* tblDesDay: Contains detailed design day data for all climate zones.
* DesDayX: Formulation of design day input lines for DOE2.

Appendix B: Design Day Weather

Design day weather conditions developed for the VRF prototype assessment are listed in Table 13. The methodology used for calculation of each value was as follows:

* Peak Dry Bulb for cooling: Sort all values for a given month by dry bulb. Choose the temperature for the record ranked 0.5% below the top (i.e. if there were 1000 records, choose the one that is in 5th place when counting down from the top).
* Low Dry Bulb for heating: Sort all values for a given month by dry bulb. Choose the temperature for the record ranked 0.5% above the bottom (i.e. if there were 1000 records, choose the one that is in 5th place when counting up from the bottom).
* Hour of High for cooling: Hour of day that the Peak Dry Bulb occurs.
* Hour of Low for heating: Hour of day that the Low Dry Bulb occurs.
* Drybulb Range: Difference between high and low dry bulb temperatures for time period that spans +/-12 hours from the hour of the design temperature.
* Hour of Low for cooling: Hour of day during which the low temperature of the dry bulb range occurs.
* Hour of High for heating: Hour of day during which the high temperature of the dry bulb range occurs.
* Day of Month: Day of the month in which the design dry bulb temperature occurs. For the winter design days, the day of month used in the simulation was changed in order to avoid vacation periods for the school prototype.
* Mean Coincident Wet Bulb: This is the average outdoor wet bulb temperature for all hours in the month for which the outdoor dry bulb temperature is within 1°F of the design value.
* Cloud Amount for cooling: This is calculated as the minimum cloud amount for all hours in the month for which the outdoor dry bulb temperature is within 5°F of the design value.
* Cloud Amount for heating: This is calculated as the maximum cloud amount for all hours in the month for which the outdoor dry bulb temperature is within 5°F of the design value.
* Wind Speed for heating: This is calculated as the maximum wind speed for all hours in the month for which the outdoor dry bulb temperature is within 5°F of the design value.
* Wind Speed for cooling: This was set to a fixed value of 0.5 mph, since wind on hot days is highly variable and tends to understate cooling load.

Table 15 Design Day Conditions Calculated for CZ 2010 Weather Files















Appendix C: Frost Control Calculations in DOE2.3

The frost control algorithms in DOE2.3 are based on information in an EPRI report from 1985[[12]](#footnote-13). This is the same document that was the source for the original DOE2.1 and DOE2.2 defrost algorithms, but some corrections have been made in the current implementation.

One issue with the original DOE2 defrost calculations was that they were based on a correlation calculation of coil surface temperature that was not based on the actual dynamics of the system in the model. This has been changed so that surface temperature is now calculated using coil bypass factor relationships, and accounting for the hourly system load and outdoor temperature and humidity.

Additional corrections to the frost control calculations for conventional heat pumps are documented in the workbook EPRI\_Defrost\_Notes.xlsx. In sheet DefF of the workbook, the formulas used for calculating the fraction of the hour during which defrost occurs and the time between defrost cycles are developed. For time initiated control, a 60 minute defrost cycle is assumed. For demand initiated control, the defrost cycle period is handled as accumulated run time and is calculated to range from 60 minutes at a moderately high humidity condition to 120 minutes at a low humidity condition.

In sheet Cap&Pwr, the adjustment factors for capacity and power due to the build-up of frost on the outdoor coil are developed. These factors are independent of whether timed or demand defrost is used, since that is accounted for by the cycle period.

The VRF system uses several of the same basic formulas for calculating frost accumulation, but there are a couple of important differences. The first is the calculation of the power and capacity adjustment due to frost on the coil. Since the VRF model explicitly evaluates the outdoor heat exchanger as a separate component with a calculated UA factor, the capacity adjustment due to frost accumulation is applied as an adjustment to the UA for any hour during which frost forms. This affects the suction temperature during normal operation, thus affecting both capacity and power.

The other difference is based on the VRF defrost configuration. During the defrost cycle, the outdoor coil is split, with one half operating in condensing mode to melt frost, and the other half evaporating to draw heat from the surroundings. The outdoor fan is turned off during the defrost cycle, so the evaporating coil operates under natural convection. Sheet VRFCoilUA in EPRI\_Defrost\_Notes.xlsx shows the calculations that were used to determine the UA adjustment factor for the VRF defrost cycle.

Appendix D: Manufacturer Data

Detailed summaries of data from manufacturers that were used in this study are contained in the workbooks described below.

Worksheets in VRF\_Mfg\_Data.xlsx:

* ODFanData: List of fan power and CFM for outdoor units for multiple manufacturers and capacities.
* AHRI\_gt\_65kbtuh: Calculation of unit EIR versus capacity from AHRI data.
* ID\_Data: Summary of indoor unit data and calculation of the following parameters:
  + Fan W/cfm for surface mounted and ducted units
  + Cooling capacity to heating capacity ratio
  + Fan CFM/rated Ton
  + Fan minimum flow ratio

Worksheets in Rated\_EIR\_vs\_Cap.xlsx:

* CalcRatedTemps: Calculations of the rated SST and SDT for each unit size are performed in this sheet.
* Summary: Lists all rated conditions and values for each size.

Appendix E : Fuel Switching Three-Prong Test

Fuel substitution programs may offer resource value and environmental benefits. Fuel-substitution programs should reduce the need for supply without degrading environmental quality. Fuel-substitution programs, whether applied to retrofit or new construction applications, must pass the following three-prong test to be considered further for funding:

1. The program must not increase source-BTU consumption. Proponents of fuel substitution programs should calculate the source-BTU impacts using the current CEC-established heat rate.

2. The program must have TRC and PAC benefit-cost ratio of 1.0 or greater. The TRC and PAC tests used for this purpose should be developed in a manner consistent with these Rules.

3. The program must not adversely impact the environment. To quantify this impact, respondents should compare the environmental costs with and without the program using the most recently adopted values for residual emissions in the avoided cost rulemaking, R.04-04-025. The burden of proof lies with the sponsoring party to show that the material environmental impacts have been adequately considered in the analysis.

For purposes of applying these tests, fuel substitution proponents must compare the technologies offered by their program with the most efficient same-fuel substitute technologies available to prospective participants that would have TRC and PAC benefit-cost ratio of 1.0 or greater. The burden of proof falls on the party sponsoring the analysis to show that the baseline comparison adheres to this requirement. Fuel substitution programs with a predominantly load building or load retention character are not eligible for funding, and the proponent of a fuel-substitution program carries the burden of proof to demonstrate that the program focuses on energy efficiency and creates net resource value.

1. http://www.deeresources.com/index.php/deer-versions/deer2013-update-for-2014-codes [↑](#footnote-ref-2)
2. http://www.deeresources.com/index.php/deer-versions/deer2015-code-update [↑](#footnote-ref-3)
3. MASControl is the simulation software used for the DEER database. The version used for DEER 2015 can be found at: http://www.deeresources.com/files/DEER2015/download/SetupMASControlX32\_3\_00\_27.msi [↑](#footnote-ref-4)
4. "2016 Nonresidential Compliance Manual", Section 4.6.2.2, CEC-400-2015-033-CMF, California Energy Commission, November 2015. [↑](#footnote-ref-5)
5. "2016 Reference Appendices for the 2016 Building Energy Efficiency Standards", Section JA2.1, CEC-400-2015-038-CMF, California Energy Commission, June 2015. [↑](#footnote-ref-6)
6. https://github.com/NREL/EnergyPlus/blob/8df72926681d9827a5ea69ab4303f30df39b1f43/testfiles/ VariableRefrigerantFlow\_FluidTCtrl\_5Zone.idf [↑](#footnote-ref-7)
7. Danfoss data taken from: "http://refrigerationandairconditioning.danfoss.us/documentation/literature/#/" with selections for VZH inverter compressors for air conditioning, April, 2016. [↑](#footnote-ref-8)
8. www.ahridirectory.org (Select "VRF Multi-Split Air Conditioning and Heat Pump Equipment" from COMMERCIAL") [↑](#footnote-ref-9)
9. "2010 Standard for Performance Rating of Variable Refrigerant Flow (VRF) Multi-Split Air-Conditioning and Heat Pump Equipment", ANSI/AHRI Standard 1230 With Addendum 2, June 2014. [↑](#footnote-ref-10)
10. Danfoss data taken from: "http://refrigerationandairconditioning.danfoss.us/documentation/literature/#/" with selections for VZH inverter compressors for air conditioning, April, 2016. [↑](#footnote-ref-11)
11. Copeland data taken from: ZPV series Performance Calculator workbooks, December, 2016. [↑](#footnote-ref-12)
12. "Performance of Air-Source Heat Pumps", EPRI EM-4226, 1985. [↑](#footnote-ref-13)