CHAPTER 4. ENERGY USE CHARACTERIZATION

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CHAPTER 4. ENERGY USE CHARACTERIZATION

4.1 INTRODUCTION

The national energy savings characterization described in this chapter provides estimates of the energy savings consumers would realize from the establishment of standards at the levels set forth in American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE) Standard 90.1-2010 for specific classes of equipment analyzed as well as for more energy efficient standards for the same equipment classes.

This chapter describes the energy use analysis for water-cooled and evaporatively cooled commercial packaged air conditioners, computer room air conditioners (CRACs), and water-source variable refrigerant volume (VRF) equipment with a cooling capacity greater than 135,000 British thermal units per hour (Btu/h). These equipment types are described in separate subsections of this chapter, which detail the analysis and the determination of baseline annual per unit energy consumption (UEC) estimates for specific product classes within each of the three product types examined. This chapter also describes the development of UEC estimates for more efficient equipment, up to the maximum technology feasible (max-tech) levels defined in either chapter 3 or this chapter.

For each of these equipment types, the Federal standard is expressed as an efficiency metric or metrics: energy efficiency ratio (EER) for cooling efficiency, coefficient of performance (COP) for heating efficiency, or sensible coefficient of performance (SCOP) for computer room air conditioners. For each equipment class, this chapter describes how an estimate of the UEC is developed corresponding to different rated efficiencies.

4.2 ENERGY USE ANALYSIS BY EQUIPMENT TYPE

4.2.1 Water-Cooled Air Conditioners

The analysis to assess the per unit energy savings of water-cooled air conditioners began with a review of the existing market as well as the review of historical shipments data provided by the Air-Conditioning, Heating, and Refrigeration Institute (AHRI) and discussed in chapter 8 of the technical support document (TSD). The review suggested that most water-cooled air conditioner units on the market are designed for installation inside commercial buildings (as opposed to on building rooftops). The shipments data suggested that, in recent years, shipments were dominated by larger equipment (greater than 240,000 Btu/h capacity), with a trend toward relatively few shipments of smaller capacity units. For that reason, the U.S. Department of Energy (DOE) analysis of energy savings focused on typical applications for equipment in the 240,000 Btu/h or greater class. Manufacturer literature suggested that a common application is floor-by-floor cooling in a multi-story building.[1]

To estimate the energy use of water-cooled air conditioners in this application, DOE used annual hourly simulation data developed from computer simulations of a prototypical, medium sized, commercial office building. The prototype building model was a three-story, 53,600 ft^2

commercial office building developed as part of DOE's commercial reference building models.[a] In this building model, referred to as a "medium office," each floor is served by a separate packaged air conditioning unit. The hourly data used for this analysis was previously developed from simulations using this model and the DOE EnergyPlus building simulation software, and reflected building simulations in 15 climate locations in the United States.[b] Each climate location used is representative of a climate in 1 of 15 climate regions that have been developed in DOE's Building Energy Codes Program and subsequently used in the development of the commercial reference building models.[2,3] Characteristics of the DOE reference medium office building model used in this analysis are provided in appendix 4A.

The medium office reference building model used packaged variable air volume (VAV) rooftop cooling units, one unit serving each floor of the building, and the hourly thermal load and supply fan air volume data used for this analysis reflects this. DOE determined that the hourly cooling thermal loads from modeling of this type of equipment would be representative of hourly loads for systems served by larger water-cooled equipment also providing floor-by-floor cooling and serving multiple building thermal zones with a VAV air distribution network. Thus, these loads would be accurate for the analysis of water-cooled air conditioners in this application. EnergyPlus does not have an equipment component model developed around a water-cooled air conditioner used in this application. EnergyPlus does have equipment models of water-source heat pumps, but these equipment models do not appear to provide for variable volume airflow. For this reason, DOE used the previously developed hourly cooling thermal load, hourly airflow, and hourly air temperature data for the air-cooled packaged rooftop equipment used in the medium office reference building model as a starting point for its analysis.

To convert the hourly data to annual equipment energy consumption for water-cooled air conditioners for baseline-level equipment, DOE developed a spreadsheet model of the equipment performance of a water-cooled air conditioner using actual manufacturer performance data for an existing 25-ton water-cooled air conditioner.[4] Cooling capacity and condenser power consumption curve fits to this data were developed using polynomial relationships and the independent variables recommended for modeling of cooling efficiency for water-source heat pumps in EnergyPlus (with minor variations). Discussion of the modeling of the water-cooled condenser and supply fan power follows.

4.2.1.1 Water-Cooled Condenser Performance

EnergyPlus provides two methods of modeling equipment performance of water-source heat pumps. The method used for this work is referred to as the "equation-fit model." The equation-fit model uses non-dimensional equations developed from curve-fits to manufacturer's performance data. These curve fits are used to predict the condenser performance in cooling (and

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[a] The commercial reference building models are available on DOE's Office of Energy Efficiency and Renewable Energy website as Energy Plus input files at: <http://www1.eere.energy.gov/buildings/commercial_initiative/new_construction.html>. Documentation of the model development is provided in Deru, M. et al., *U.S. Department of Energy Commercial Reference Building Models of the National Building Stock* (NREL/TP-5500-46861) (2011).

[b] Hourly simulation results for the medium office reference building model were developed at Pacific Northwest National Laboratory and provided for support of the ASHRAE 90.1 Mechanical Subcommittee in April 2009. Simulation results used reflect code requirements for buildings designed to ANSI/ASHRAE/IESNA 90.1-2004.

heating) mode. Least squares regression is used to generate a set of performance coefficients from catalog performance data at various conditions for the equipment modeled. Then, the respective coefficients from the least squares regressions are used in the model to simulate the heat pump performance in the EnergyPlus software. The variables used by EnergyPlus to describe the water-cooled condenser performance are the entering air wet bulb and dry bulb temperature, the condenser water inlet temperature, the entering air volumetric flow rate, and the condenser water flow rate. Each of these variables is made non-dimensional by first dividing by corresponding values for each variable at a defined reference condition. Two minor variations from the implementation in EnergyPlus included the use of inch-pound units in developing the DOE performance curves and the selection of reference temperature conditions that correspond precisely to the DOE rating conditions. The governing equations for the cooling performance are as follows:

$$\frac{Q_{total}}{Q_{total\_ref}} = A_1 + A_2 \times \left(\frac{T_{wb}}{T_{wb\_ref}}\right) + A_3 \times \left(\frac{T_{w,in}}{T_{w,in\_ref}}\right) + A_4 \times \left(\frac{\dot{V}_{air}}{\dot{V}_{air\_ref}}\right) + A_5 \times \left(\frac{\dot{V}_w}{\dot{V}_{w\_ref}}\right)$$

Eq. 4.1

$$\frac{Q_{sen}}{Q_{sens\_ref}} = B_1 + +B_2 \times \left(\frac{T_{db}}{T_{db\_ref}}\right) + B_3 \times \left(\frac{T_{wb}}{T_{wb\_ref}}\right) + B_4 \times \left(\frac{T_{w,in}}{T_{w,in\_ref}}\right) + B_5 \times \left(\frac{\dot{V}_{air}}{\dot{V}_{air\_ref}}\right) + B_6 \times \left(\frac{\dot{V}_w}{\dot{V}_{w\_ref}}\right)$$

Eq. 4.2

$$\frac{Power}{Power_{ref}} = C_1 + C_2 \times \left(\frac{T_{wb}}{T_{wb\_ref}}\right) + C_3 \times \left(\frac{T_w}{T_{w-ref}}\right) + C_4 \times \left(\frac{\dot{V}_{air}}{\dot{V}_{air\_ref}}\right) + C_5 \times \left(\frac{\dot{V}_w}{\dot{V}_{w\_ref}}\right)$$

Eq. 4.3

Where:

A_1 through C_5 = equation fit coefficients for the cooling mode,
Q_{total} = cooling capacity (Btu/h),
Q_{total_ref} = cooling capacity at reference condition (Btu/h),
Q_{sens} = sensible cooling capacity (Btu/h),
Q_{sens_ref} = sensible cooling capacity at reference condition (Btu/h),
Power = condenser power (W),
$Power_{ref}$ = condenser power at reference condition (W),
T_{wb} = entering air wet bulb temperature (Rankine (°R)),
$T_{wb\text{-}ref}$ = entering air wet bulb temperature at reference temperature (°R),
T_{db} = entering air dry bulb temperature (°R),
$T_{db\text{-}ref}$ = entering air dry bulb temperature at reference temperature (°R),
T_w = entering water temperature (°R),
$T_{w\text{-}ref}$ = entering water temperature at reference temperature (°R),
V_{air} = volumetric airflow rate (cubic feet per minute (cfm)),
$V_{air\text{-}ref}$= volumetric airflow rate at reference (gallons per minute (gpm)),
V_w = volumetric water flow rate (cfm), and
$V_{w\text{-}ref}$ = volumetric water flow rate at reference (gpm).

The equations above are based on steady state full-load operation at the rating (reference) conditions and off-rating conditions. To account for operation at part-load and non-steady-state operation, part-load performance modifier curves for air-source air conditioners—which were used in the EnergyPlus medium office reference building model—were also used for this analysis. Because these part-load modifier curves reflect the effects of compressor cycling at part-load for air-cooled systems, it was determined that they should be representative of the compressor cycling impacts for water-cooled air conditioners as well.

The part-load modifier curves are of the following equation form:

$$CoolPLFFPLR = C_1 + C_2 \times PLR$$

Eq. 4.4

Where:

CoolPLFFPLR = cooling power part-load factor modifier,
PLR = ratio of cooling load to cooling capacity under the conditions of interest, and
C_1 and C_2 = equation fit coefficients for the part-load performance curves.

C_1 and C_2 values of 0.771 and 0.229, respectively, were used based on the part-load modifier curves in the reference building simulations and for this analysis.

The steady state power input is then adjusted according to the following equation:

$$Power_{Hourly} = Power \times \frac{PLR}{CoolPLFFPLR}$$

Eq. 4.5

Where:

$Power_{Hourly}$ = hourly average power in response to hourly cooling load,
Power = steady state power at conditions as defined in Eq. 4.3,
PLR = part-load ratio as defined in Eq. 4.4, and
CoolPLFFPLR = part-load modifier as defined in Eq. 4.4.

For each climate zone, the original system air volumes were sized based on design day runs and the maximum hourly supply air for each system was used as the basis for the system air volumes and reference volumetric flow rate. The peak cooling loads were adjusted for rated conditions using Eq. 4.1 and used to determine the equipment nominal capacity for each system. This is also the cooling capacity defined at the reference condition. No hourly condenser-water flow rates were available, and for the modeling purpose the condenser water flow rate at all hours was assumed to be at the reference (rated) condition. Thus, the condenser water flow rate did not affect the performance curves in the analysis.

4.2.1.2 Supply Water Temperature

The performance equations developed in this spreadsheet model separately account for the water-cooled gross cooling capacity and power consumption as a function of entering air conditions and supply water temperature and flow rate. To function, the spreadsheet model requires an hourly entering water temperature and entering water flow rate. For this analysis, a simple cooling tower supply water temperature model was developed based on a defined control profile with minimum 70 °F return water temperature and using a 7 °F approach temperature. Thus, the return water temperature from the cooling tower is calculated as

$$T_w = \min\left[T_{wb,o} + 7\,,70\right] + 459.67$$

Eq. 4.6

Where:

T_w = building supply water temperature from cooling tower (°R), and
$T_{wb,o}$ = outdoor air wet bulb temperature (°F).

The supply water temperature from the cooling tower is thus estimated at 7 °F above the current wet bulb temperature during most cooling hours. Condenser water flow rates were assumed to be equivalent to the nominal rating condition water flow rates for all cooling hours.

Using these inputs and performance curves, it was possible to calculate the hourly condenser energy consumption of each of the three water-cooled air conditioner units for this building model.

4.2.1.3 Supply Fan Power

In addition, DOE's spreadsheet model calculated the fan energy consumption for the equipment. Fan energy for each system at design conditions (peak fan capacity) was estimated using the following assumptions:

$$Power_{fan,max} = \frac{\Delta P \times Q}{6350 \times n_f \times n_m \times 0.746}$$

Eq. 4.7

Where:

$Power_{fan,max}$ = design fan power (kW),
ΔP = total fan static pressure (in. H_20),
Q = design supply airflow rate (cfm),
n_f = fan efficiency, and
n_m = motor and drive efficiency.

A design total fan static pressure of 4.35 in. H_2O, fan efficiency of 0.65, and motor efficiency of 0.9 were used, providing for a design fan system power of 0.87 W/cfm at design conditions. The total fan static pressure was judged likely to be less than that used in the reference building design (4.35 in. H_2O versus 5.35 in. H_2O) based on the determination that, due to the length of the duct runs, the total supply and return path pressure drop would be less with water-cooled units installed inside the building as opposed to packaged units installed on rooftops.

Because this was a VAV system with varying hourly airflow rate, the fan power was adjusted hourly using a fan power part-load curve used for the DOE reference building VAV system. The fan power curve provides the relationship between fan system power and system airflow rate, which varied hourly for the VAV system. The fan power versus flow rate relationship is of the form used in EnergyPlus:

$$PLF_{fan} = C_1 + C_2 \times FF + C_3 \times FF^2 + C_4 \times FF^3 + C_5 \times FF^3$$

Eq. 4.8

Where:

PLF_{fan} = a part-load factor multiplier equal to the ratio of system fan power to design fan power,
FF = fraction of system design airflow rate, and
$C1$ through $C5$ = equation coefficients for the VAV fan power performance curve.

The coefficients C_1 through C_5 used for the VAV fan power performance curve are those from the original packaged VAV system used in the reference building model and are as follows:

$C_1 = 0.040759894$
$C_2 = 0.08804497$
$C_3 = -0.07292612$
$C_4 = 0.943739823$
$C_5 = 0$

4.2.1.4 Water-Cooled Condenser-Only Cooling Coefficient of Performance vs. Energy Efficiency Ratio Relationship

For analysis of energy use at the base efficiency (EER) level, DOE developed estimates for the condenser-only efficiency (condenser-only cooling COP) for the water-cooled air conditioner based on the nominal rating conditions for an 11 EER unit. Condenser-only cooling COP at rating conditions is an input to the EnergyPlus simulation model. This parameter can be calculated by backing out the estimated fan power at nominal rating conditions from the rated energy consumption while accounting for the effect of fan heat. This is done by calculating the condenser-only cooling COP according the following equation:

$$COP_{CU\_Cool} = \frac{\frac{EER}{3.413} + R}{1 - R}$$

Eq. 4.9

Where:

EER = EER rating of the equipment being modeled,
R = ratio of supply fan power to total equipment power at the AHRI rating condition, and
COP_{CU_Cool} = condensing-only cooling COP at rated conditions.

The parameter R can be estimated from available manufacturer fan power data at the external static rating condition and a known airflow rate. Generally, however, the airflow rate at rating is not provided in the manufacturer's literature. DOE used an R of 0.164 based on review of available data for a large evaporatively cooled air conditioner and assuming a supply airflow rate of 350 cfm/ton.[2] For an 11.0 EER unit that DOE modeled directly, the calculated condenser-only COP corresponding to an 11.0 EER unit was 4.052.

4.2.1.5 Baseline Annual Energy Use/Ton for Climates – Water-Cooled

For each of the 15 climates, DOE used the spreadsheet model to calculate the hourly condenser energy use and supply fan energy use for each of the three water-cooled air conditioner units in the reference medium office building model and summed the results to provide annual energy use estimates for condenser and for supply fan separately. DOE also calculated the cooling capacity for each unit and summed these for the whole building. Dividing the energy use figures for both the condenser and the supply fan by the rated cooling capacity provides a normalized energy-use-per-ton figure for each end use at the baseline efficiency level. These climate-specific results, along with the climate weighting factors used, are shown in Table 4.2.1. The climate weighting factors are developed from construction statistics for commercial office floor space for small and for medium sized office buildings (likely to use commercial packaged air-conditioning equipment regardless of system size) for the period from 2003 to 2007.[5] For each building type, weights were first calculated for the building class and then averaged. Figure 4.2.1 shows the floor space weighting factors for this analysis compared with that of all 15 DOE reference buildings types with construction floor space tallied in the study.

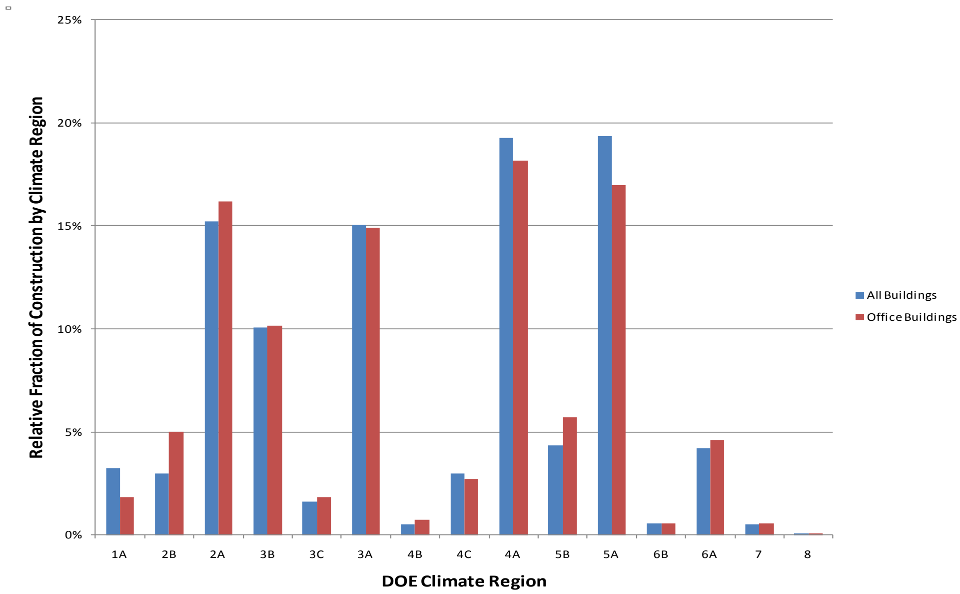

Figure 4.2.1 Comparison of Floor-Space Weighting Factors by Region, Small and Medium Office Buildings versus All Buildings

In addition, Table 4.2.1 shows a parameter called the "full load equivalent operating hours" (FLEOH), which is calculated as the annual cooling load served divided by the system cooling capacity. In this case, the FLEOH shown is the result from weighting of the FLEOH calculated for each individual air-conditioning unit by the relative capacity of that unit to the total capacity of all units serving the reference building model. While not used in DOE's energy calculation, FLEOH values provide a useful reference point to compare with other energy calculation methods.

Table 4.2.1 Normalized Energy Use Ton by Climate for Water-Cooled Air Conditioners, 11.0 Energy Efficiency Ratio

Climate Zone	Representative Climate Location	Construction Wt.	Annual Condenser Energy *kWh/ton*	Annual Fan Energy *kWh/ton*	Annual System Energy *kWh/ton*	FLEOH
1A	Miami, FL	1.82%	1,999	257	2,256	2,718
2A	Houston, TX	16.20%	1,440	215	1,654	1,966
2B	Phoenix, AZ	4.99%	1,446	354	1,800	2,052
3A	Memphis, TN	14.91%	1,051	204	1,255	1,440
3B	El Paso, TX	10.15%	973	284	1,258	1,413
3C	San Francisco, CA	1.82%	497	273	770	736
4A	Baltimore, MD	18.18%	824	216	1,040	1,135
4B	Albuquerque, NM	0.72%	776	357	1,133	1,154
4C	Salem, OR	2.71%	517	247	764	736
5A	Chicago, IL	16.96%	674	215	889	926
5B	Boise, ID	5.70%	542	283	825	807
6A	Burlington, VT	0.56%	545	230	775	759
6B	Helena, MT	4.62%	446	360	807	682
7	Duluth, MN	0.56%	352	249	600	496
8	Fairbanks, AK	0.10%	259	436	696	398
National		100.00%	947	242	1,189	1,319

FLEOH = full load equivalent operating hours; kWh = kilowatt-hours

4.2.1.6 Energy Use Estimates for Baseline and Higher Efficiency Water-Cooled Air Conditioner Equipment by Equipment Class.

National average estimates of the energy use for efficiencies other than those directly modeled in the spreadsheet tool were developed by first scaling the national average condenser energy use per ton from the spreadsheet simulation of 11.0 EER equipment shown in Table 4.2.1 by the condenser-only cooling COP corresponding to the higher EER equipment. The condenser-only cooling COP is developed based on Eq. 4.9 for higher EER values. To that result is added the baseline fan energy use per ton values, and the sum is then multiplied by the selected representative tonnage for each equipment class; or in equation form,

$$UEC_{EER} = CAP \times [CondUEC_n \times \frac{COP_{CU\_Cool\_Modeled}}{COP_{CU\_Cool\_EER}} + FanUEC_n]$$

Eq. 4.10

Where:

CAP = nominal equipment capacity in cooling tons,
UEC_{EER} = unit energy consumption at the specific EER level being evaluated,
$CondUEC_n$ = normalized condenser energy consumption from the modeling, kWh/ton – national basis,
$COP_{CU\_Cool\_Modeled}$ = the cooling condenser-only COP used in the spreadsheet equipment model,
$COP_{CU\_Cooled\ EER}$ = the cooling condenser-only COP corresponding to the specific EER level being evaluated, and

$FanUEC_n$ = normalized condenser energy consumption from the modeling, kWh/ton – national.

Thus, for example, to estimate the annual energy consumption of a 35-ton water-cooled air conditioner at an EER of 12.4, the initial step is to calculate a condenser-only COP_{EER} using Eq. 4.9. For an EER of 12.4, the ASHRAE 90.1-2010 level for this product class, this corresponds to a COP of 4.543. UEC is then calculated as

$$UEC_{12.4\,EER} = 35\ tons\ \times [947\ kWh/ton \times \frac{4.052}{4.543} + 242\ kWh/ton]$$

Eq. 4.11

A representative capacity for each product class was selected, as shown in Table 4.2.2, based on the approximate average capacity of the units available in the market as identified in DOE's market analysis.

Table 4.2.2 Water-Cooled Air Conditioner Representative Capacities by Product Class

Product Class	Representative Cooling Capacity
Water-Cooled Air Conditioner, ≥65,000 and <135,000 Btu/h, Electric Resistance Heating or No Heating	8 tons
Water-Cooled Air Conditioner, ≥65,000 and <135,000 Btu/h, All Other Heating	8 tons
Water-Cooled Air Conditioner, ≥135,000 and <240,000 Btu/h, Electric Resistance Heating or No Heating	15 tons
Water-Cooled Air Conditioner, ≥135,000 and <240,000 Btu/h, All Other Heating	15 tons
Water-Cooled Air Conditioner, ≥240,000 Btu/h, Electric Resistance Heating or No Heating	35 tons
Water-Cooled Air Conditioner, ≥240,000 Btu/h, All Other Heating	35 tons

For this analysis, the annual supply fan power was not adjusted for different efficiency levels. A more detailed analysis of the engineering of higher efficiency water-cooled air conditioners might show that some improvement can come from reducing supply fan power. One path to reducing fan power in rooftop equipment is believed to have been through larger case designs and consequently lower internal fan pressure drop. However, based on the assumption that these units are typically installed in mechanical rooms as opposed to rooftops, increased size may come at the expense of installation cost and available (rentable) floor space. It is noted that for a given EER level, assuming that fan efficiency is improved to reach that level would mean less improvement to the condenser performance and less energy savings from the condenser portion of the packaged air conditioning system.

The resulting UEC values for each class of water-cooled air conditioner are provided in Table 4.2.3 for selected efficiency levels analyzed for the national energy savings analysis.

Table 4.2.3 Water-Cooled Air Conditioner Unit Energy Consumption by Product Class

	Small Water-Cooled Air Conditioners Electric or No Heat 65-135 kBtu/h	Small Water-Cooled Air Conditioners Other Heat 65-135 kBtu/h	Large Water-Cooled Air Conditioners Electric or No Heat 135-240 kBtu/h	Large Water-Cooled Air Conditioners Other Heat 135-240 kBtu/h	Very Large Water-Cooled Air Conditioners Electric or No Heat 240-760 kBtu/h	Very Large Water-Cooled Air Conditioners Other Heat 240-760 kBtu/h
Average Cooling Capacity (*tons*)	8	8	15	15	35	35
Efficiency Level (*EER*)						
Base Case – Federal Standard	11.5	11.3	11.0	11.0	11.0	10.8
Efficiency Level 1	12.1	11.9	12.5	12.3	12.4	12.2
Efficiency Level 2	13.0	13.0	13.0	13.0	13.0	13.0
Efficiency Level 3	14.0	14.0	14.0	14.0	14.0	14.0
Efficiency Level 4	15.0	15.0	15.0	15.0	NA*	NA*
Efficiency Level 5 – "Max-Tech" –	16.4	16.4	16.1	16.1	14.8	14.8
Unit Energy Consumption (*kWh/yr*)						
Base Case – Federal Standard	9,199	9,322	17,838	17,838	41,621	42,205
Efficiency Level 1	8,855	8,966	16,206	16,402	38,041	38,504
Efficiency Level 2	8,396	8,396	15,743	15,743	36,733	36,733
Efficiency Level 3	7,953	7,953	14,911	14,911	34,793	34,793
Efficiency Level 4	7,566	7,566	14,186	14,186	NA*	NA*
Efficiency Level 5 – "Max-Tech" –	7,101	7,101	13,490	13,490	33,422	33,422

*An efficiency level 4 was not identified for very large water-cooled air conditioners.

4.2.2 Evaporatively Cooled Air Conditioners

The analysis to assess the per unit energy use of evaporatively cooled air conditioners began with a review of the existing market (see chapter 2 of the TSD). DOE did not identify any current models of evaporatively cooled air conditioners with less than 240,000 Btu/h cooling capacity. The review of the market suggested that the evaporatively cooled equipment models currently available are designed as packaged rooftop cooling units. Based on the available models identified, DOE estimated that the average capacity was 40 tons. Therefore, DOE's analysis of energy savings focused on typical applications for this very large equipment class. Because of the large size, DOE believes that a common application would also be packaged VAV system serving a building. To assess energy use for evaporatively cooled equipment, DOE modified the three-story office reference building model used for the water-cooled equipment discussed previously to serve as the simulation model for the very large evaporatively cooled air conditioner equipment class.

The EnergyPlus simulation tool can model evaporatively cooled unitary air conditioners directly, with only minor modifications from the air-cooled unitary air conditioner equipment models that were used in the original DOE medium office reference building model. EnergyPlus offers two alternatives for modeling evaporatively cooled air conditioners. One is to use performance curves that characterize the steady state efficiency as a function of ambient wet bulb temperature and coil entering air dry bulb temperature (in contrast to the performance curves for air-cooled equipment, which use ambient dry bulb temperature and coil entering air dry bulb temperature). DOE was not able to derive separate performance curves for evaporatively cooled equipment as neither compressor nor overall condenser power data at different operating conditions were available in the manufacturer literature reviewed. Therefore, DOE utilized the second approach provided for in EnergyPlus, which is to use performance curves for air-cooled equipment that are a function of the condenser entering air dry bulb conditions, but provide for a evaporative condenser using an evaporative condenser effectiveness parameter that reduces the condenser coil entering air conditions to below the ambient air temperature according to the following equation:

$$T_{C,i} = (T_{wb,o} + (1 - Eff) \times (T_{db,o} - T_{wb,o})$$

Eq. 4.12

Where:

$T_{c,i}$ = temperature of the air entering the condenser coil (°F),
$T_{wb,o}$ = wet bulb temperature of the outdoor air (°F),
$T_{db,o}$ = dry bulb temperature of the outdoor air (°F), and
Eff = evaporative condenser effectiveness.

DOE modified the medium office reference building model to use this latter modeling approach and defined an evaporatively cooled condenser for each of the three packaged VAV systems in the model. An evaporative condenser with an effectiveness of 0.9, equivalent to the default value in EnergyPlus, was used.

DOE developed estimates for the condenser-only cooling COP based on the nominal rating conditions. This was done by backing out the estimated fan power at nominal rating conditions and separately accounting for the impact of fan heat using Eq. 4.9, as discussed for water-cooled equipment. The fan power ratio parameter R was estimated from a review of available manufacturer fan power data for an evaporatively cooled rooftop air conditioner. DOE estimated an R of 0.137 based on a review of available data for a large evaporatively cooled air conditioners and assuming a supply airflow rate of 350 cfm/ton of net cooling capacity.[6] For an 11.0 EER unit that DOE modeled directly, the calculated condenser-only COP was then estimated at 3.893. DOE used the fan power performance curves and peak fan power assumptions from the original reference medium office building model.

4.2.2.1 Baseline Annual Energy Use/Ton for Climates – Evaporatively Cooled

DOE performed simulations in the 15 climates identified previously using the 11.0 EER evaporatively cooled air conditioners. Equipment was sized using design day sizing. DOE extracted the cooling capacity, annual equipment condenser energy consumption, and blower energy consumption for each evaporatively cooled air conditioner equipment unit simulated. The annual energy consumption values were then normalized by dividing by the equipment capacity in cooling tons. The sum of the resulting condenser energy per cooling ton and blower energy per cooling ton represents the annual energy consumption per cooling ton for equipment at that 11 EER efficiency level. These climate-specific results, along with the climate weights, are shown in Table 4.2.4.

Table 4.2.4 Normalized Energy Use Ton by Climate for Evaporatively Cooled Air Conditioners, 11.0 Energy Efficiency Ratio

Climate Zone	Representative Climate Location	Construction Wt.	Annual Condenser Energy *kWh/ton*	Annual Fan Energy *kWh/ton*	Annual System Energy *kWh/ton*
1A	Miami, FL	1.82%	1,683	380	2,068
2A	Houston, TX	16.20%	1,293	329	1,626
2B	Phoenix, AZ	4.99%	1,117	431	1,551
3A	Memphis, TN	14.91%	979	305	1,287
3B	El Paso, TX	10.15%	873	420	1,295
3C	San Francisco, CA	1.82%	420	343	764
4A	Baltimore, MD	18.18%	782	318	1,101
4B	Albuquerque, NM	0.72%	620	405	1,027
4C	Salem, OR	2.71%	426	332	759
5A	Chicago, IL	16.96%	569	313	883
5B	Boise, ID	5.70%	448	374	823
6A	Burlington, VT	0.56%	473	340	814
6B	Helena, MT	4.62%	335	356	692
7	Duluth, MN	0.56%	318	374	693
8	Fairbanks, AK	0.10%	186	383	569
National		100.00%	838	341	1,189

4.2.2.2 Energy Use Estimates for Baseline and Higher Efficiency Evaporatively Cooled Air Conditioners by Equipment Class.

Energy use consumptions were developed for representative evaporatively cooled equipment sizes corresponding to each product class, as was done for water-cooled air conditioners. A representative capacity for each product class was selected, as shown in Table 4.2.5, based on the approximate average capacity of the units available in the market as identified in DOE's market analysis.

Table 4.2.5 Evaporatively Cooled Air Conditioner Representative Capacities by Product Class

Product Class	Representative Cooling Capacity
Evaporatively Cooled Air Conditioner, ≥ 240,000 Btu/h, Electric Resistance Heating or No Heating	40 tons
Evaporatively Cooled Air Conditioner, ≥ 240,000 Btu/h, All Other Heating	40 tons

The national average energy consumption per ton figures from Table 4.2.4 were then multiplied by the selected equipment capacities for the evaporatively cooled equipment class analyzed to establish the UEC values for the evaporatively cooled equipment classes at an 11 EER level.

To assess the annual energy consumption at the other efficiency levels analyzed, DOE developed estimates of the condenser-only cooling COP for each other efficiency level using Eq. 4.9. To establish annual energy consumption metrics for the higher efficiency levels, DOE used Eq. 4.10, adjusting the baseline annual condenser energy consumption for each climate by the ratio of the baseline condenser-only cooling COP at 11 EER to the condenser-only cooling COP at the efficiency levels analyzed. The annual fan energy consumption estimates were held constant at the baseline level for all higher standards. The UEC for each efficiency level analyzed is the sum of the annual condenser energy consumption and the annual fan energy consumption.

The resulting UEC values for each class of water-cooled air conditioner are provided in Table 4.2.6 for selected efficiency levels analyzed for the national energy savings analysis.

Table 4.2.6 Evaporatively-Cooled Air Conditioner Unit Energy Consumption

	Large Evaporatively-Cooled Air Conditioner Electric or No Heat 240-760 kBtu/h	Large Evaporatively-Cooled Air Conditioner Other Heat 240-760 kBtu/h
Average Cooling Capacity (*tons*)	40	40
	Efficiency Level (*EER*)	
Base case	11.0	10.8
Level 1 – ASHRAE	11.9	11.7
Level 2	12.5	12.5
Max Tech	13.1	13.1
	Unit Energy Consumption (*kWh/yr*)	
Base case	47,171	47,766
Level 1 – ASHRAE	44,732	45,243
Level 2	43,294	43,294
Max Tech	41,983	41,983

4.2.3 Computer Room Air Conditioners

DOE estimated the per unit energy savings of computer room air conditioners using a temperature-bin-method analysis developed specifically for computer room air conditioners and implemented in a commercial spreadsheet. The spreadsheet calculates an annual UEC for air-, water-, or glycol-cooled computer room air conditioners of specific capacities based on the representative sizes analyzed.

The spreadsheet calculations presume that the heat load in the space being served is primarily sensible and that the energy consumption can be determined using the SCOP values for a given set of thermal load and outdoor conditions. The spreadsheet calculations further assume that conditions outside the server room environment have no effect on the thermal load (*i.e.*, the thermal loads are a function of internal gains from the computer servers and associated equipment and do not take into account variations in envelope load or ventilation induced load that might occur through the year).

The calculation proceeds in two principal stages. First, an hourly average power consumption performance map for a given class of computer room air conditioners (at a defined efficiency level) is developed using user entered design parameters for the equipment and either outdoor air dry bulb temperature or outdoor air wet bulb temperature as the dependent variable, depending on equipment class. The performance map results are in terms of average hourly power usage and are calculated at the midpoint of defined 5 °F temperature bins. Second, a dot-product multiplication of the performance map (an average hourly power use versus temperature vector) and a vector of the annual hours associated with each temperature bin in a given climate is used to generate the expected annual energy consumption for that climate.

4.2.3.1 Development of Computer Room Air Conditioner Performance Map

The calculation steps to represent the average hourly power consumption for a given temperature bin are based on the following analysis steps:

1. Estimate the average computer server power heat load served by a system.
2. Estimate the condensing fluid or dry bulb air temperature entering the CRAC condensing unit.
3. Estimate the load that would be met by an economizer (if incorporated) for that hour.
4. Calculate the thermal load served by the direct expansion (DX) system (after accounting for any economizer load reduction).
5. Calculate the full load power consumption of the CRAC unit.
6. Calculate the part load ratio for that hour (the ratio of the actual load to the full load capacity).
7. Calculate the part load factor for that hour (an adjustment to the part load ratio to deal with how the equipment responds to part load levels).
8. Calculate the hourly run fraction and the corresponding hourly average compressor/condenser power consumption for that hour.
9. Add in the fan power, assuming continuous fan operation.

These steps are outlined in more detail below.

4.2.3.2 Estimate the Average Computer Server Heat Load

CRAC units are designed for a given cooling capacity. However, due to sizing considerations and the fact that server loads will vary during the day or year, the ratio of the typical heat load in the space being served to the capacity of the CRAC unit will be less than 1.0. DOE estimated the average sensible load on a CRAC unit to be 65 percent of the sensible capacity. This value was based on an Australian Minimum Energy Performance Standards (MEPS) study of close controlled air conditioners used for data processing applications, which used a 65 percent average load factor for this equipment (operating on average at 65 percent of full load capacity) based on industry consultation.[7] DOE found that a different analysis presented to the California Energy Commission (CEC) assumed that 25 percent of the load would be at each of the following load points (25, 50, 75, and 100 percent), resulting in an average load factor of 62.5 percent over the year, nearly equivalent to the 65 percent used in this analysis.[8]

There is likely significant variation in CRAC load factors in the field. Server loads can vary due to computer usage, but more recently, computer operating strategies like operating system virtualization can make it possible to more easily migrate computer server requirements between physical servers, allowing for individual computer servers to be placed in low activity states until needed. In addition, because of the requirements for high cooling reliability, purchase of redundant cooling units is common in the industry. Industry literature often speaks of purchasing $n + 1$ units for redundancy, where n is the number of units required to meet a design load and + 1 refers to the purchase of a redundant unit in case of system failure. One manufacturer refers to a general rule that the capacity of CRAC-rated cooling must be 1.3 times the anticipated information technology load rating plus any capacity added for redundancy, stating this approach works well with smaller network rooms of under 4,000 ft^2.[9] However, DOE did not find any large studies that examined actual data on electrical load being cooled over the year relative to capacity installed, and used the 65 percent load factor figure for the energy analysis.

4.2.3.3 Estimate the Condensing Fluid or Air Temperature for the Computer Room Air Conditioner Unit

The 15 classes of CRAC units can be grouped into three main categories based on condensing system type: air-cooled systems, water-cooled systems, and glycol-cooled systems. Air-cooled systems utilize air-cooled condensers. In most cases, the condensers are located outdoors in a split-system arrangement where refrigerant piping connects the outdoor condenser to the indoor CRAC equipment. In some systems, outdoor air can be ducted into the building for condenser cooling. However, in both instances, the air used for cooling the condenser is at the prevailing outdoor air temperature.

Lower outdoor air temperatures during the year provide a cooler thermal sink and can improve the operating efficiency of the system by reducing the condensing temperature and corresponding refrigerant head pressure. However, for several reasons, including providing for oil migration and proper expansion valve performance, minimum head pressure requirements do exist in refrigerant system designs, including CRAC equipment designs. Various means of head pressure control exist in the air-cooled condenser, including reducing fan speed, cycling of fans in multiple fan systems, and flooding compressors with liquid refrigerant to reduce the condenser volume that can be used for active refrigerant condensation.

Water- and glycol-cooled CRAC equipment utilizes either a water- or a glycol-cooled heat exchanger for condensing the refrigerant in the CRAC unit. These heat exchangers are constructed as an integral part of the CRAC unit. In the example of a water-cooled unit, water enters the condenser and is heated by the condensing refrigerant. Warmer water then leaves the CRAC unit, is cooled, and flows back to the CRAC condenser. There are a variety of methods to cool the condensing water, including directly via dedicated open evaporative cooling towers serving one or more CRAC units, central cooling towers serving both CRAC units as well as other heating, ventilating, and air-conditioning (HVAC) equipment, closed evaporative cooling towers, and dry coolers. Glycol-cooled CRAC units are designed similarly to water-cooled CRAC units except that the refrigerant to fluid condenser is designed for a glycol-water mix. The warmed glycol-water mix leaves the CRAC condenser and must be cooled prior to being returned to the CRAC unit for additional condenser cooling. As with water cooling systems, a single glycol fluid loop can serve multiple CRAC units. However, this glycol-cooling fluid loop is a closed system to prevent the loss of glycol. Thus, the fluid cooling is generally accomplished by a dry liquid air heat exchanger, referred to in the industry as a dry cooler or fluid cooler. One feature of glycol-cooled systems is that the heat rejection system can readily operate at low (below freezing) temperatures. Since CRAC units serve computer cooling loads that are year round, this is an important consideration.

As with air-cooled CRAC units, a lower condenser fluid temperature can result in improved efficiency of the refrigeration system, down to the minimum head pressure allowed in the refrigeration system. Head pressure control in water- and glycol-cooled systems is typically accomplished by mixing or controlling the flow rate of the fluid in the condenser.

For the CRAC energy analysis tool, DOE made a number of generalized assumptions on how the fluid condenser temperature would be controlled:

- For air-cooled systems, it was assumed that the condensing fluid (air) would be at the prevailing ambient dry bulb air temperature for a given hour.
- For water-cooled systems, it was assumed that the entering water would be provided by an open cooling tower with a constant 7 °F approach temperature (*i.e.*, the temperature of the water returned to the CRAC unit would be 7 °F above the ambient wet bulb temperature).
- For glycol-cooled systems, it was assumed that the glycol-water fluid would be cooled using a dry cooler and that the temperature of the fluid returned to the CRAC unit would be 10 °F above the ambient dry bulb air temperature.

In addition, for each system, a "low temperature" limit was also included, which helps establish the lower limit on the refrigerant head pressure. In water-cooled systems, water must be held above the freezing temperature, and this sets a practical limit on the water temperature irrespective of refrigerant head pressure considerations. In glycol- and air-cooled systems, the "low temperature limit" is defined not by the air or glycol properties, but by the need to maintain a minimum head pressure and thus condensing temperature of the refrigerant. Since the heat exchangers are passive, active controls are used to limit either the heat transfer capabilities of the condenser (*e.g.*, by reducing airflow or flooding condensers in air-cooled equipment) or the condenser heat sink temperature (through mixing or flow rate control for glycol- or water-cooled equipment). The key use of these minimum temperatures for the spreadsheet model is to restrict the model such that that there is no improvement in CRAC efficiency with lower temperatures below these defined points. The control of the fluid temperature and the "minimum's" established in the model are shown in Table 4.2.7.

Table 4.2.7 Condensing Conditions Established for CRAC Equipment

CRAC System Type		
Air Cooled	Water Cooled	Glycol Cooled
Assumed Approach Temperature *°F*		
0	7	10
Assumed Low Temperature Limit *°F*		
40	40	35

In the model, weather data binned into 5 °F ambient dry bulb temperature bins are used to develop the annual performance for air-cooled and glycol-cooled CRAC equipment for different climates. Weather data in 5 °F wet bulb temperature bins is used to develop the annual performance for water-cooled CRAC equipment.

4.2.3.4 Estimate the Load That Would Be Met by an Economizer (If Incorporated) for That Hour

Economizers for DX cooling systems are system design features that allow for part or all of the thermal cooling load from the space to be removed prior to reaching the refrigerant evaporator coil. For CRAC equipment, the typical economizer is incorporated into the equipment as a water-to-air or glycol-to-air cooling coil placed in the air stream prior to the DX evaporator coil. When weather and system operations permit, the water or glycol economizer coil can be provided with cool entering fluid that can partially or wholly meet the cooling load from the space. This entering fluid temperature is the same as that provided to the CRAC unit from the external fluid cooling loop. Remaining heat load not met by the economizer coil can be met by the refrigeration coil. Because the economizer coil sits in the air stream, it results in an increased fan pressure and fan energy over the same equipment design without the economizer coil. The full load performance rating of equipment with economizer coils thus reflects this increased fan energy use.

To estimate the load reduction for the economizer coil at a given hour, it was necessary to estimate both the entering fluid temperature and the overall heat transfer capability of the economizer coil. Limited information from manufacturer data indicates that, for CRAC systems with economizer coils, the economizers can generally meet the cooling load with fluid entering temperatures near 45 °F.[10,11,12] In addition, ASHRAE 90.1-2010[13] has a requirement that evaporatively cooled water economizers serving primarily computer rooms shall be able to meet 100 percent of the expected cooling load at 40 °F dry bulb / 35 °F wet bulb temperature outdoor conditions, and a system using dry-cooler water economizers shall be able to meet 100 percent of the expected cooling load at 35 °F dry bulb outdoor conditions.

For CRAC classes using water or glycol economizers, DOE estimated the overall heat transfer coefficient (UA) for a fluid-to-air economizer cooling coil that could meet the net sensible cooling capacity of the representative CRAC cooling unit at 75 °F return air temperature and entering fluid conditions equal to the entering water or glycol conditions at 35 °F outdoor wet bulb temperature or 35 °F outdoor dry bulb temperature respectively, as follows:

$$UA = \frac{Qs}{RAT - EFT_{design}}$$

Eq. 4.13

Where:

Qs = rated cooling capacity (sensible),
UA = overall economizer coil heat transfer coefficient (Btu/h-F),
RAT = return air temperature (assumed equal to 75 °F), and
EFT_{design} = entering fluid temperature to economizer coil (°F).

For water-cooled economizers, EFT_{design} was estimated at 42 °F (35 °F outdoor wet bulb temperature + 7 °F approach temperature). For glycol-cooled economizer, EFT_{design} was estimated at 45 °F (35 °F outdoor dry bulb temperature + 10 °F approach temperature).

The economizer coil is assumed to be capable of operation any time the entering fluid is within a defined operating temperature range provided for by a low limit and high limit temperature control based on the entering water or glycol temperature. For water-cooled economizers, the low limit set point was set to 35 °F. For glycol-cooled systems, the low limit set point of the range was set to -20 °F, the lowest operating temperature provided for with a typical glycol loop. From a practical point of view, the economizer can meet 100 percent of the expected load under higher entering fluid temperatures than are defined by these conditions and will control the mixed fluid temperature in the coil to meet space conditions. Manufacturer literature examined suggested a range of economizer high limit controls set points, from systems where the economizer does not work when it can only partially meet the load to systems that would function with up to 65 °F entering fluid temperature. A temperature of 58 °F was chosen to represent a typical value within the range of high limit control strategies.

The heat load reduction from the economizer was calculated as the larger of either the maximum potential heat transfer at the economizer coil for that hour or the hourly computer server load:

$$Qecon = Max[UA\ x\ (RAT - EFT), \qquad \text{Server Load})]$$

Eq. 4.14

Where:

$Qecon$ = economizer load reduction,
EFT = entering fluid temperature to economizer coil (°F), and
Server Load = sensible heat load from the space being served.

The server loads for the analysis were based on 65 percent of the sensible capacity, as discussed in section 4.2.3.2.

The analysis did not investigate the use of air side economizers. There appear to be questions within the industry on the use of air side economizers in spaces served by CRAC units. Air side economizers have significant potential to reduce the annual load on all types of CRAC equipment as well as computer room air handler equipment using chilled water coils. Primary concerns within the industry include the introduction of particulates, or chemical contaminants, in the supply air stream as well as the need to humidify the air entering through the economizer during cold weather when the outside air humidity ratio is low, which could lower the indoor space humidity conditions below a defined minimum humidity threshold. The cost of any needed humidification depends on the type of humidifier (*e.g.*, steam, ultrasonic), fuel costs, and the number of economizer hours and amount of humidification required for a given system and climate.

ASHRAE 90.1-2010 introduced tables of economizer requirements for cooling systems in server computer rooms as a function of climate and system cooling capacity (greater than 65,000 Btu/h), but at the same time provided significant exceptions that limited the application of the economizer requirements. These exceptions include (1) any building where the total design cooling load of all computer rooms in the building is less than 3,000,000 Btu/h and the building in which they are located is not served by a centralized chilled water plant; (2) individual computer rooms where the total design cooling load is less than 600,000 Btu/h and the building in which they are located is served by a centralized chilled water; (3) spaces where less than 600,000 Btu/h of computer room cooling capacity is being added to an existing building; and (4) computer spaces that can be defined as serving specific critical needs. DOE had no data on which to establish the frequency of air side economizer usage with the different classes of CRAC equipment, but notes that the exemptions provided in ASHRAE 90.1-2010 would seem to indicate that their use would be limited to very large computer server applications. DOE notes that the analysis assumption of no air side economizer usage likely sets an upper bound on the cooling loads and average energy use of air-cooled CRAC equipment.

4.2.3.5 Estimate the Cooling Load

The cooling load at any hour is estimated as the computer server load plus the heat from operating CRAC supply fans, minus the load met by the economizer coil in that hour. Base fan power was estimated at 0.418 W/cfm for systems without an economizer coil. Fan airflow was estimated at 472 cfm per ton of sensible capacity based on review of manufacturer data for a variety of CRAC products. For systems with an economizer coil, the fan power was increased by the amount necessary to increase the total system power and reduce the EER of a corresponding baseline water- or glycol-cooled system by the 0.05 EER reduction seen in the ASHRAE 90.1 requirements. The estimated fan power for water-cooled equipment with economizers was 0.461, 0.464, and 0.467 W/cfm for less than 65,000 Btu/h, greater than or equal to 65,000 Btu/h but less than 240,000 Btu/h, and greater than or equal to 240,000 Btu/h CRAC equipment, respectively. The estimated fan power for glycol-cooled equipment with economizers was 0.461, 0.464, and 0.467 W/cfm for less than 65,000 Btu/h, 65,000 but less than 240,000 Btu/h, and greater than or equal to 240,000 Btu/h CRAC equipment, respectively. It is assumed that all of the fan power is converted to heat that must be removed by the CRAC system.

4.2.3.6 Estimate the Hourly Power Usage

To estimate the average energy consumed for any hour of operation, it was assumed that the supply fan would run in any hour, but that the average hourly compressor power could be reduced from the nominal rating condition due to reduced compressor run time (corresponding to reduced cooling load) or external conditions.

Compressor power and run time were estimated based on methods outlined in the EnergyPlus building simulation software for DX cooling equipment. DOE developed three curves to modify the full load performance (expressed in terms of the energy input ratio (EIR)—the ratio of compressor/condenser power to the cooling output at the evaporator coil, in consistent units) as a function of condenser air or entering fluid temperatures. DOE had no detailed equipment performance data from manufacturers on which to base these relationships. DOE instead used example tables presented in ASHRAE Standard 127 informative appendix D,[14] which show the variation of full load SCOP as a function of outside air temperature (for air-cooled products) or as a function of entering fluid temperature (for glycol-cooled products). DOE converted the SCOP data to sensible EIR data (ratio of compressor/condenser power to sensible capacity) using the following formula:

$$EIR_s = \frac{1 - R}{SCOP + R}$$

Eq. 4.15

Where:

EIR_s = sensible energy input ratio for condenser (kW/kW), and
R = ratio of fan power to total system power.

The fan power ratio used for these initial calculations was approximately 0.2. Four SCOP points, corresponding to four outside air temperatures, were converted to sensible EIR points for air-cooled equipment.[c] These sensible EIR values then normalized by dividing by the EIR at the nominal 95 °F rating condition, and the normalized values plotted against outdoor air temperature. A second order polynomial curve fit using the outdoor temperature as the independent variable was used to characterize the relationship. This curve fit is referred to as COOL-EIR-FT (cooling EIR as a function of temperature).

A similar analysis of the ASHRAE 127 appendix D EIR versus entering fluid temperature for glycol-cooled equipment was also conducted. The curve fits are provided in Table 4.2.9 and shown in Figure 4.2.2 for air- and glycol-cooled equipment.

ASHRAE 127 appendix D does not provide similar data for water-cooled equipment as a function of condenser entering fluid temperature. For water-cooled equipment, DOE developed a COOL-EIR-FT curve fit parallel to that of the glycol-cooled systems, but which passed through the ASHRAE 127 nominal rating point for water-cooled CRAC equipment. This curve fit is also shown in Figure 4.2.2.

[c] Only points corresponding to variable capacity equipment were provided in ASHRAE 127 appendix D for these temperatures, but the ratings were full load ratings points. Non-variable capacity equipment are not required to test performance at other than the nominal outside air temperature or entering fluid conditions.

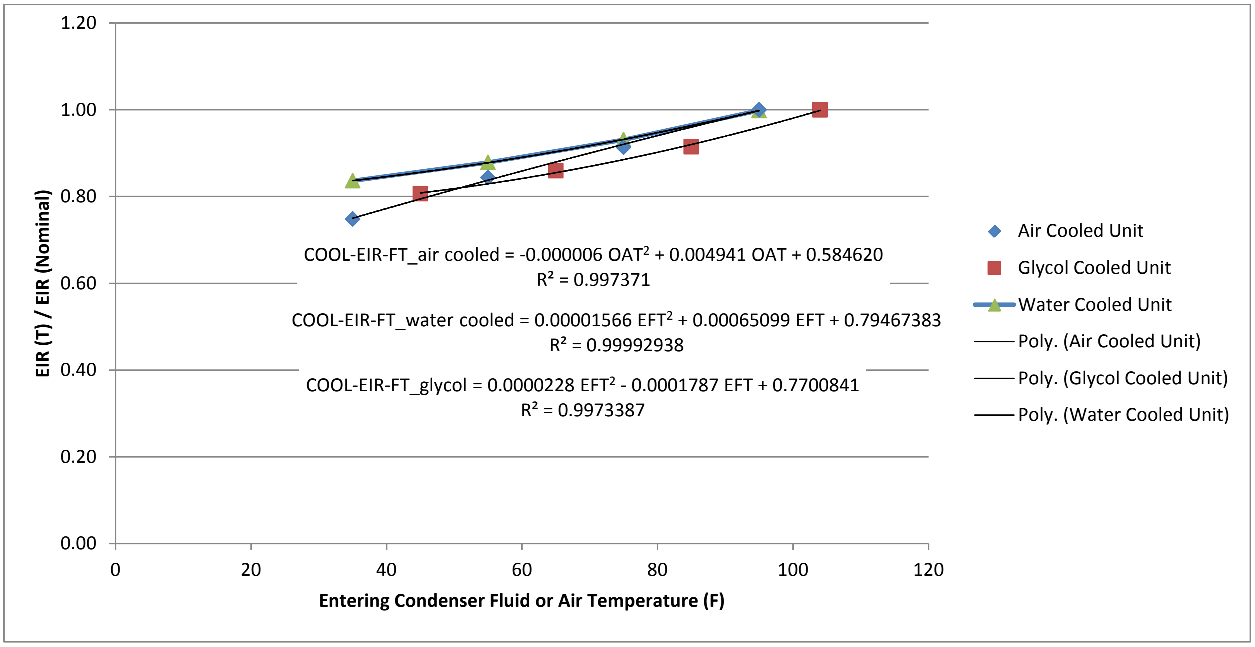

Figure 4.2.2 Cooling EIR as a Function of Entering Condenser Fluid Air Temperature

Table 4.2.8 CRAC Performance as a Function of Entering Condenser Fluid or Outdoor Air Temperature

Equipment	Energy Input Ratio Performance Curve
Air Cooled	COOL-EIR-FT = -0.000006 x OAT^2 + 0.004941 x OAT + 0.584620
Water Cooled	COOL-EIR-FT = 0.00001566 x EFT^2 + 0.00065099 x EFT + 0.79467383
Glycol Cooled	COOL-EIR-FT = 0.0000228 x EFT^2 – 0.0001787 x EFT +0.7700841

OAT = outside air temperature (°F), EFT = entering fluid temperature (°F)

These relationships were used to estimate the full load sensible EIR for a given outdoor air or entering condenser fluid temperature.

The actual power consumption for a given hour reflects the full load EIR at the operating conditions as well as the cooling load at that hour. DOE defines the part load ratio (PLR) for a given hour as the ratio of the actual load to cooling capacity at each hour. DOE calculated the PLR for each temperature bin. DOE also calculated a part load factor (PLF) correlation curve for each CRAC unit. The product of the rated EIR and Cool-EIR-FT modifier curve is divided by the calculated PLF to give the "effective" EIR for a given simulation hourly bin.

In EnergyPlus, the PLF correlation is designed to account for efficiency losses, primarily due to compressor cycling. EnergyPlus provides a typical PLF correlation curve for single compressor DX system as

$$PLF = 0.85 + 0.15\ x\ PLR$$

Eq. 4.16

Where:

PLF = part load factor (single compressor systems).

DOE developed a second average PLF relationship for two-compressor systems by assuming that two equally sized compressors were used. For a PLR of less than 0.5, one compressor is off and one compressor cycles to meet the load. For a PLR greater than 0.5, one compressor operates continuously without cycling and the second compressor cycles to meet the additional load. DOE developed separate PLF relationships for each compressor by examining the load on each compressor relative to the capacity of each compressor (assuming each provided 50 percent of the total capacity of the system). The overall PLF for two-compressor systems is the load-weighted average of the PLF for each single compressor in the system, and is defined by

$$PLR < 0.5, \quad PLF = 0.85 + 0.15\ x\ 2\ x\ PLR$$
$$PLR > 0.5, \quad PLF = \frac{1}{PLR}\ x\ (\frac{1}{2} + (PLR - 0.5)\ x\ (0.85 + 0.15\ x\ 2\ x\ (PLR - 0.5))$$

Eq. 4.17

The run time fraction (RTF) for the compressor/condenser system in a given hour bin is defined as

$$RTF = \frac{PLR}{PLF}$$

Eq. 4.18

The average power for the compressor/condenser in a given bin is calculated as the product of the cooling capacity, the adjusted EIRs, and the run time fraction for that bin, or

$$Power_{cond} = CAP_s\ x\ EIR_s\ \mathrm{x\ COOL - EIR - FT\ x\ RTF}$$

Eq. 4.19

Where:

CAP_s = the nominal sensible capacity of the CRAC unit.

The total system power is the sum of the fan power and the compressor/condenser power, or

$$Power = Power_{cond} + \ Power_{supply\ fan_s}$$

Eq. 4.20

Where:

$Power_{supply\ fan}$ = the supply fan power, assuming continuous operation.

Performance maps of estimated CRAC hourly power consumption for each temperature bin are provided in appendix 4B by equipment class.

4.2.3.7 Calculate Annual Energy Consumption by Climate and by State

The annual energy consumption for each representative equipment selection in a given climate is calculated as the dot product of the performance map vector for that CRAC unit and the corresponding bin hour vector for a given climate. DOE developed dry bulb and wet bulb temperature bin hours for 241 U.S. climates using Typical Meteorological Year 3 (TMY3) weather data, and from these developed 241 location-specific estimates of annual energy consumption for each class of equipment at each efficiency level. These location-specific energy use estimates were converted to state-level energy use estimates by equipment class at each efficiency level using previously developed TMY-location weighting factors. See appendix 4C for a discussion of the development of these climate weights. The resulting estimates of annual energy consumption by state are shown in appendix 4B and referenced directly in the life-cycle cost spreadsheet discussed in chapter 6.

4.2.4 Water-Source Variable Refrigerant Flow Heat Pumps

Annual UEC estimates for VRF water source heat pumps at or greater than 135,000 Btu/h cooling capacity were developed based on whole building energy simulation of a medium sized prototype office building in 15 locations around the United States, corresponding to each of 15 ASHRAE climate zones discussed previously.[5] The prototype office building model is a two-story, 20,000 ft^2 building that has five distinct HVAC zones, a core zone, and four perimeter zones, per floor. The building model used ASHRAE 90.1-2004 compliant construction to represent relatively recent construction practices and provide loads suitable for representing both new and replacement/retrofit construction. Assumptions as to the envelope and building form factor were drawn from previous analysis of small office buildings.[15] The HVAC model was modified to use two separate water-source VRF systems, one serving each floor, in conjunction with a gas fired boiler and a single cooling tower to serve the condensing water loop for the VRF systems. The simulation tool was a commercial version of the DOE2.1E building simulation tool, with the capability to model water source VRF equipment via custom DOE2.1E functions.[16] This simulation tool utilizes actual performance curves from equipment manufacturers for a large number of VRF units, including water-source VRF units. Results from simulations of water-source VRF heat pumps at three different efficiency levels in each of the 15 locations were developed and aggregated to estimates of national energy use for VRF systems using commercial construction weights developed for each climate. These national results in turn were used to estimate the annual energy consumption of the VRF units at five different efficiency levels that DOE analyzed. Details of this analysis follow.

4.2.4.1 Building Model Characteristics

The small office building prototype model uses characteristics typical of buildings of this size and use. The building is a 20,000 ft^2, two-story, square building. The square shape was originally chosen to provide an orientation-neutral simulation platform. The building has a window-area ratio of 21 percent with a 12-ft floor-to-floor height (a plenum is not directly modeled). The small office building is assumed to have slab-on-grade construction for the floor, mass walls, and a flat roof. A picture of the building structure is shown in Figure 4.2.3. The building is divided into five HVAC zones, four perimeter zones, and a core zone, per floor.

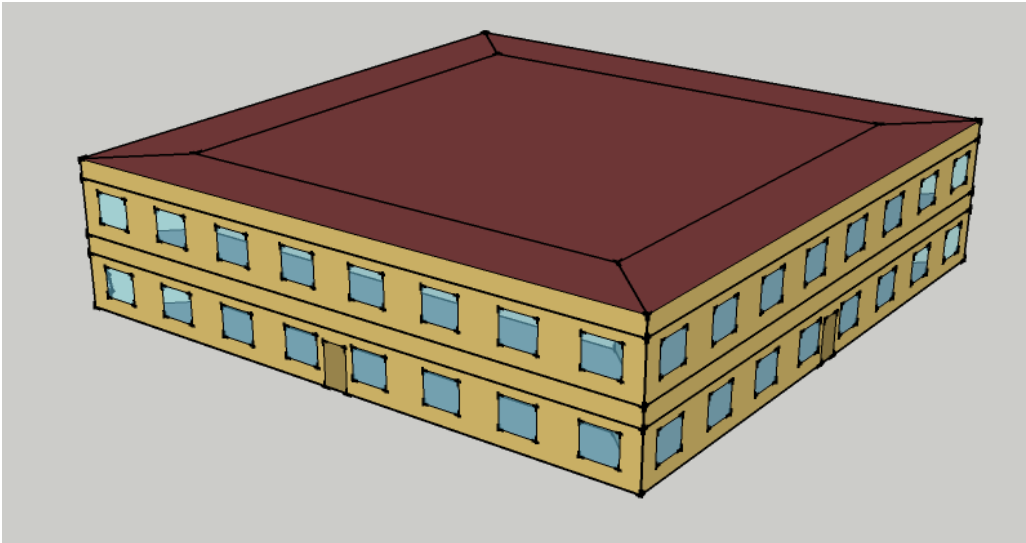

Figure 4.2.3 Small Office Building Design View

Where applicable, the building components for this model were assumed to "just meet" the minimum prescriptive requirements of ASHRAE Standard 90.1-2004. Envelope and window performance characteristics are shown in Table 4.2.9 for each ASHRAE climate zone.

Table 4.2.9 Envelope Performance Characteristics

Climate Zone	Characteristic Climate	Wall U-Factor	Roof U-Factor	Glass U-factor	Window SHGC / SC
1A	Miami, FL	0.58	0.063	1.2	0.25 / 0.287
2A	Phoenix, AZ	0.58	0.063	1.2	0.25 / 0.287
2B	Houston, TX	0.58	0.063	1.2	0.25 / 0.287
3A	Memphis, TN	0.58	0.063	0.57	0.25 / 0.287
3B	El Paso, TX	0.58	0.063	0.57	0.25 / 0.287
3C	San Francisco, CA	0.151	0.063	1.2	0.39 / 0.448
4A	Baltimore, MD	0.151	0.063	0.57	0.39 / 0.448
4B	Albuquerque, NM	0.151	0.063	0.57	0.39 / 0.448
4C	Salem, OR	0.151	0.063	0.57	0.39 / 0.448
5A	Chicago, IL	0.123	0.063	0.57	0.39 / 0.448
5B	Boise, ID	0.123	0.063	0.57	0.39 / 0.448
6A	Helena, MT	0.104	0.063	0.57	0.39 / 0.448
6B	Burlington, VT	0.104	0.063	0.57	0.39 / 0.448
7	Duluth, MN	0.09	0.063	0.57	0.49 / 0.563
8	Fairbanks, AK	0.08	0.048	0.46	0.49 / 0.563*

*No SHGC requirements for ASHRAE Climate Zone 8; however, selected values represent simulation inputs and are equivalent to Zone 7.

The building is assumed to follow typical small office occupancy patterns, with peak occupancy occurring from 8 a.m. to 5 p.m. weekdays and limited occupancy from 6 a.m. until 8 a.m. and from 5 p.m. until midnight for janitorial functions. The building is assumed to be unoccupied on weekends and holidays. Schedules for lighting and miscellaneous equipment were default schedules for office buildings selected using the EnergyPro user interface for ASHRAE 90.1 compliance using a simulation performance approach option. These schedules reflect a 5-day per week occupancy, with principal occupancy between 8 a.m. and 6 p.m. and limited additional occupancy in the evenings for janitorial services. HVAC schedules followed occupancy with the HVAC system scheduled on from 6 a.m. to 9 p.m. at night. Thermostat set points were 70 °F heating, 75 °F cooling during occupied hours, and 55 °F heating, 99 °F cooling on weekends and holidays. There is a 2-hour thermostat warm-up period between 6 a.m. and 8 p.m. weekday mornings.

Figure 4.2.4 through Figure 4.2.7 show the schedules for lighting, plug loads, and occupancy. Figure 4.2.7 provides the building space temperature set points. The first three of these schedules represent multipliers to defined daily "peak" conditions. In the case of ventilation, it is assumed that ventilation occurs whenever the fan system is operational.

For the occupancy, lighting, plug use, and ventilation, the schedules represent multipliers to defined peak densities. Lighting density was defined using the ASHRAE 90.1-2004 maximum lighting power density requirement of 1.0 W/ft^2. Plug load density (*e.g.*, computers and other miscellaneous loads) was 1.0 W/ft^2. Occupancy peak density was defined as 88 persons for the building (4.4 persons per 1,000 ft^2). Total outdoor air exchange during the occupied period (effective ventilation) was based on ASHRAE Standard 62-1999 and was set to 0.13 cfm/ft^2 of floor area for each conditioned zone.

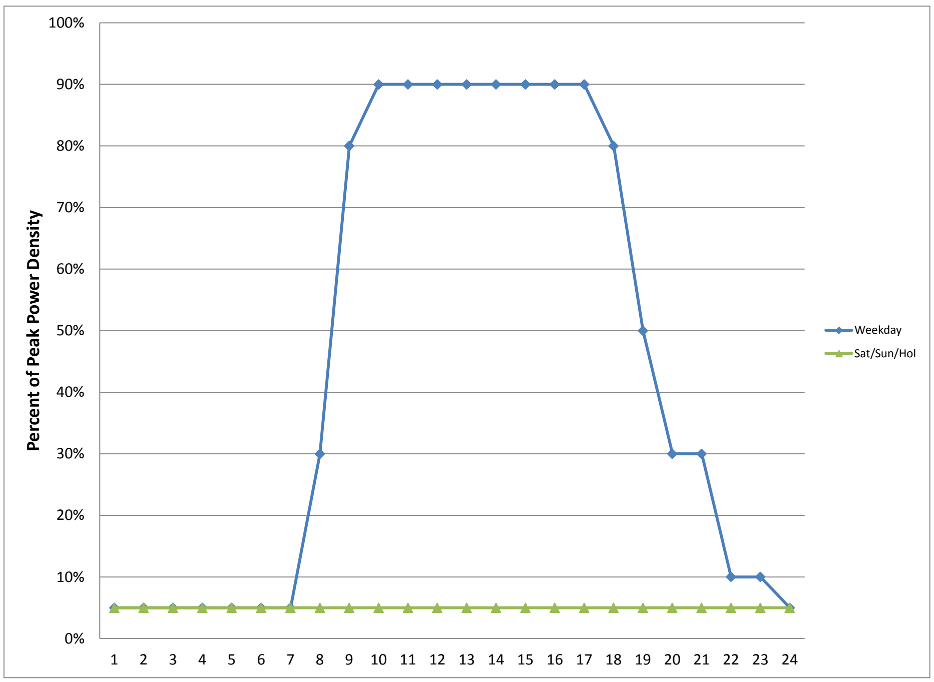

Figure 4.2.4 Small Office Lighting Schedule

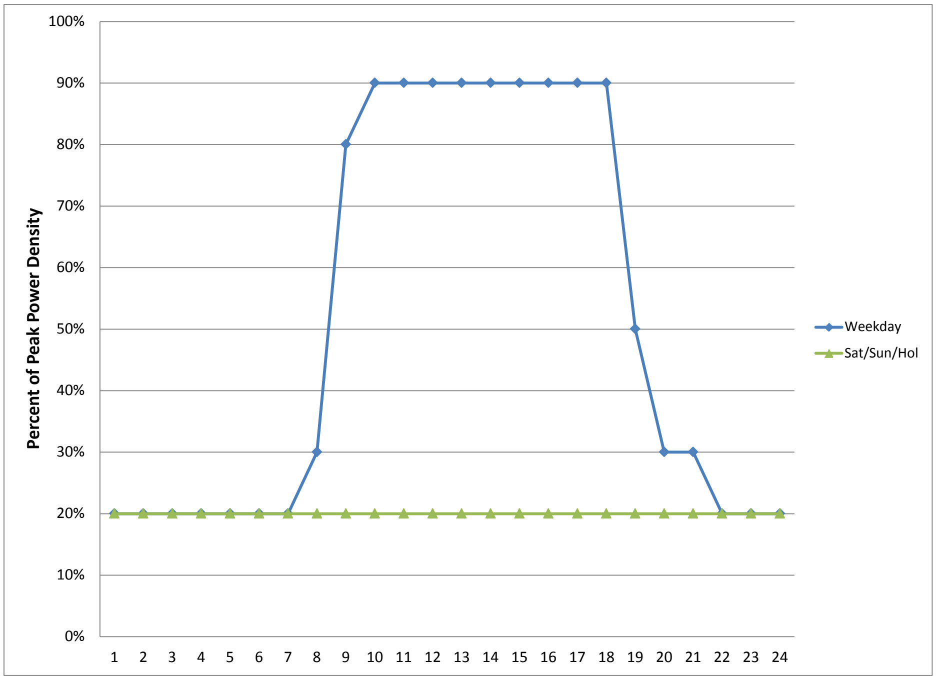

Figure 4.2.5 Small Office Plug Load Schedule

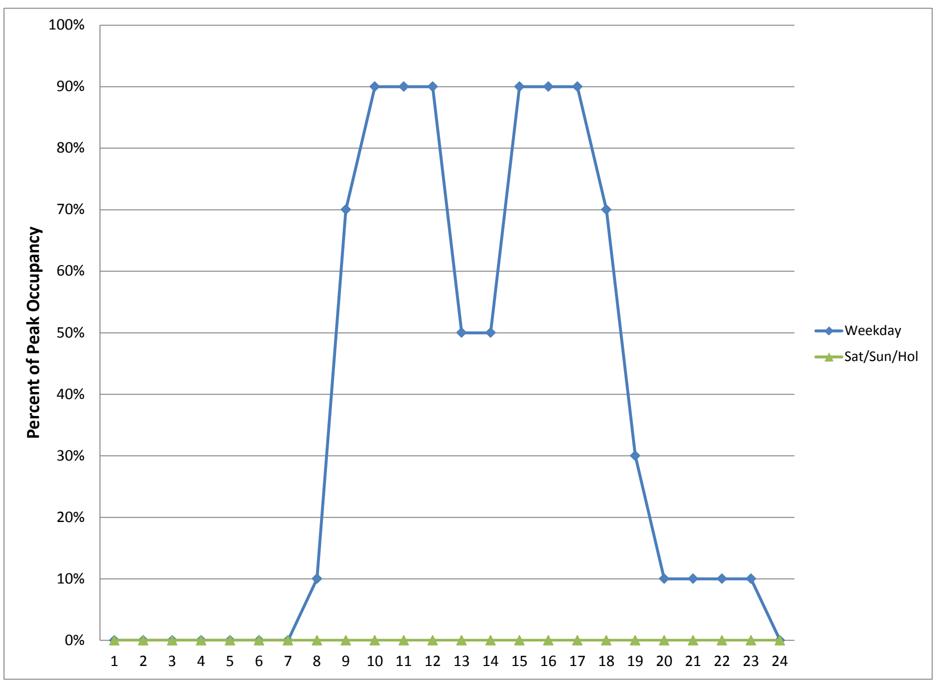

Figure 4.2.6 Small Office Occupancy Schedule

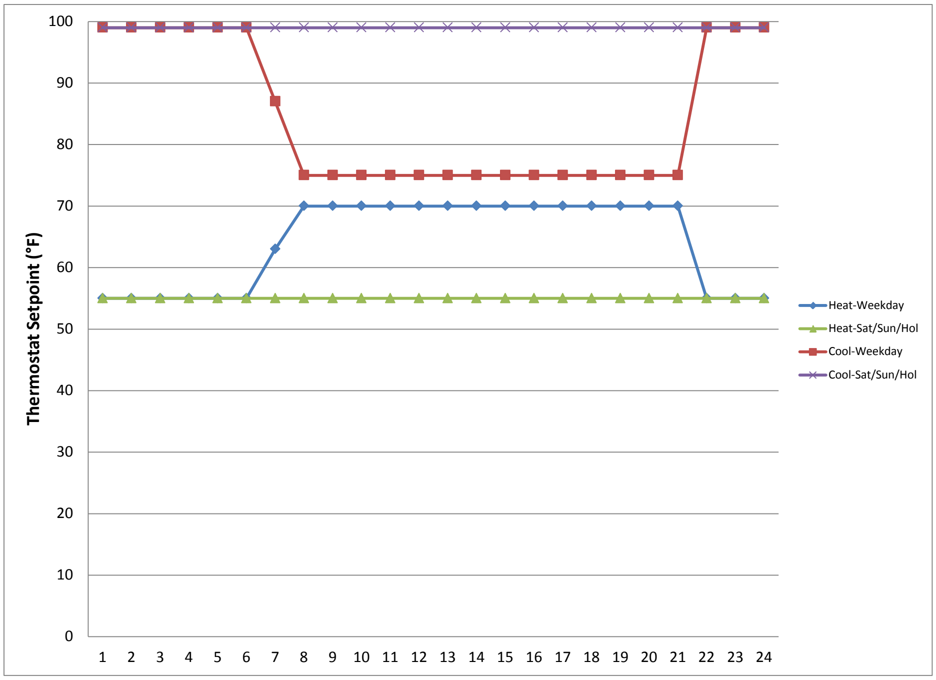

Figure 4.2.7 Small Office Thermostat Set Point Schedule

4.2.4.2 Variable Refrigerant Flow Modeling Characteristics

DOE developed estimates of the key equipment input parameters, including fan power and condenser COP at rating conditions, to represent systems at three different efficiency levels identified in the AHRI certified product directory for VRF multi-split air conditioners and heat pumps.[17] These efficiencies corresponded to (1) the lowest efficiency level identified in the directory and close to the ASHRAE baseline; (2) an efficiency level corresponding to the highest efficiency identified for ducted systems; and (3) an efficiency level near the highest efficiency identified for ductless systems. The AHRI 1230-2010[18] test procedure used for these ratings provides that each condensing unit be tested as both a ducted system, representing equipment using indoor units that are connected to short distribution ducts, and as a ductless system, representing equipment using ductless indoor sections that provide conditioned air directly to the building space served. Because of a higher external static pressure when testing ducted units, the rated efficiency (EER and COP) of a given condensing unit is lower when tested as a ducted system than when tested as a ductless system. The two higher efficiency levels simulated utilized the same condensing unit but represent ratings as a ducted and as a ductless system, respectively.

As the ratings data do not identify the indoor units used, DOE selected a representative ducted indoor section and developed supply fan power estimates based on that unit for ducted systems representing the first two efficiency levels simulated. For non-ducted systems where there was a large variety of indoor sections available, DOE developed an average fan power estimate based on average supply fan power data for five different ductless indoor section designs. Key characteristics of the VRF units simulated are shown in Table 4.2.10.

Table 4.2.10 VRF Modeling Characteristics

System Type	Model*,**	Nominal Ratings		Condenser Performance†		Indoor Section Classification	Indoor Section Fan Power *W/cfm*
		EER	COP	COPcool	COPheat		
Without Heat Recovery	A	10.3	3.95	4.546	5.039	Ducted	0.2408
	B	11.8	4.51	5.366	5.942	Ducted	0.2408
	C	14.4	4.47	5.366	5.942	Non-ducted	0.1136
With Heat Recovery	A	10.3	3.95	4.546	5.039	Ducted	0.2408
	B	11.8	4.51	5.366	5.942	Ducted	0.2408
	C	14.4	4.47	5.366	5.942	Non-ducted	0.1136

* ASHRAE 90.1-2010 specifies a minimum efficiency of 10.0 EER for water source VRF heat pumps > 135,000 Btu/h without heat recovery and a 9.8 EER for water source VRF heat pumps > 135,000 Btu/h with heat recovery.
** Model A data was for the lowest efficiency model shown in the AHRI database for this equipment class. Model B data was for the model with the highest efficiency "ducted" rating from the same manufacturer (a second manufacturer had data at 11.9 EER for ducted equipment). Model C data represents the Model B unit tested as a ductless unit and thus has the same compressor performance. A second manufacturer had a 14.5 EER unit, which represented the maximum shown in the database, but with a lower heating COP.
† Based on EnergyPro simulation tool data and used as entry in DOE2.1E building description language (BDL) data files created by EnergyPro. COPcool referrers to condenser cooling COP at rated conditioner as utilized by EnergyPro. COPheat refers to compressor heating COP at rated condition. Compressor performance data checked against manufacturer literature showing compressor output capacity and power consumption.

DOE performed simulations of the prototype office building using these three VRF efficiency levels and using weather data for 15 specific cities. Each specific city location provided representative climate corresponding to one of the ASHRAE 90.1-2004 climate zones. A corresponding location-specific TMY3 data file was used as the source for hourly weather data. The locations simulated are shown in Table 4.2.11.[3]

Table 4.2.11 Climate and Locations for Simulations

ASHRAE 90.1 Climate Zone	City	Construction Wt.
1A	Miami, FL	3.24%
2A	Houston, TX	15.22%
2B	Phoenix, AZ	2.98%
3A	Atlanta, GA	15.03%
3B	Los Angeles, CA	10.08%
3C	San Francisco, CA	1.61%
4A	Baltimore, MD	19.29%
4B	Albuquerque, NM	0.52%
4C	Seattle, WA	2.98%
5A	Chicago, IL	19.37%
5B	Denver, CO	4.34%
6A	Minneapolis, MN	4.21%
6B	Helena, MT	0.57%
7	Duluth, MN	0.51%
8	Fairbanks, AK	0.06%

DOE first performed preliminary design and annual simulation runs to size the indoor equipment and two outdoor units serving the conditioned zones. Because each VRF manufacturer has limits on the overall ratio of indoor unit capacity to outdoor unit capacity, the sum of the capacity of the indoor units was not allowed to be greater than 150% of the outdoor unit system capacity for any VRF system.

Six sets of simulation were done, with each set utilizing all 15 climates: three sets for the “without heat recovery” equipment, corresponding to the three identified efficiency levels shown in Table 4.2.10, and three sets for equipment with heat recovery at the same efficiency levels. The annual electrical energy use for the VRF equipment, including each condensing unit and all associated evaporator units, was extracted from the simulation results for each building simulated and normalized by cooling capacity to provide annualized estimates of total VRF cooling, heating, and fan energy consumption at the average cooling capacity estimated by DOE for VRF systems. For VRF systems without heat recovery, DOE estimated the average capacity at 216,000 Btu/h based on the average for available equipment found in the AHRI certified products directory. For VRF systems with heat recovery, DOE estimated the average capacity at 192,000 Btu/h using the same data source. DOE estimated the national average energy use for VRF systems at both the ASHRAE 90.1-2010 efficiency level and the max-tech level identified for ducted VRF systems on a national basis using commercial building construction weights previously developed by DOE and assigned to each of the 15 climates. Because VRF can be used in many types of buildings and building designs, the construction weights used were those for all commercial buildings.[5] These weights are shown in Table 4.2.11. Table 4.2.12 shows the calculated national average energy use from the simulations for heating, cooling, and fan energy on a per-ton of capacity installed basis. Annual fan energy was later apportioned between heating and cooling for accounting purposes using the ratio of the heating to cooling annual energy consumption.

Table 4.2.12 National Average VRF Energy Use – Simulations Results

Type	Simulation Level	Cooling *kWh/ton*	Heating *kWh/ton*	Fan *kWh/ton*	Fan-Cooling *kWh/ton*	Fan-Heating *kWh/ton*
No Heat Recovery	A	827	331	365	261	104
	B	729	295	365	260	105
	C	729	295	172	122	49
Heat Recovery	A	786	211	365	288	77
	B	683	180	365	289	76
	C	683	180	172	136	36

For each equipment class, DOE developed a piecewise linear relationship between the national average cooling energy use and the reciprocal of the cooling EER for the three efficiencies modeled. DOE also developed a piecewise linear relationship between the national average heating energy use and the reciprocal of the heating COP for the three efficiencies modeled. DOE then used these relationships to estimate the annual average cooling and heating energy use at the ASHRAE baseline efficiency level[d] and at four higher efficiency levels, including the highest EER and COP levels found in the AHRI certified product directory for each product class (identified as max-tech levels for this analysis). These levels are shown in Table 4.2.13. Level 2 corresponded to the highest efficiency found for ducted VRF equipment.

[d] No water-source VRF equipment greater than 135 kBtu/h with efficiency levels lower than the ASHRAE 90.1-2010 efficiency levels was identified in the AHRI certified product directory or through manufacturer literature. Consequently, the lowest efficiency levels examined in the energy use characterization were the ASHRAE 90.1-2010 efficiency levels.

DOE held the fan energy use constant for levels at and below Level 2 to that estimated based on the ducted VRF simulations. Level 1 was approximately midway between the ASHRAE baseline and Level 2. The highest efficiency level was Level 4, which is considered the max-tech level and was based on the highest efficiency found in the AHRI certified directory. Level 4 corresponded to a ductless system. DOE estimated the energy use at the max-tech level using the linear relationship between the higher two efficiencies simulated. Level 3 was intermediate between Levels 2 and 4. Annual energy use at Level 3 was calculated based on interpolation between Level 2 and the max-tech level. In all, DOE developed annual energy consumption estimates for efficiency levels at EER values of 10.0, 11.0, 12.0, 13.0, and 14.5 for water-source VRF heat pumps without heat recovery. DOE developed annual UEC estimates for efficiency levels at EER values of 9.8, 11.0, 12.0, 13.0, and 14.5 for water-source VRF heat pumps with heat recovery. While ASHRAE 90.1-2010 provides for a 0.2 EER difference between the EER for products with and without heat recovery, a review of all manufacturer ratings for these two equipment classes in the AHRI certified directory did not show evidence of any difference in rated EER for product models with and without heat recovery (identified by models having the same or nearly the same base model number). The resulting annual UEC for each product classes at each efficiency level analyzed is shown in Table 4.2.13

Table 4.2.13 Unit Energy Consumption Estimates by Efficiency Level for Water-Source VRF Equipment

<table>
<tr><th colspan="2" rowspan="2"></th><th colspan="4">Product Classes</th></tr>
<tr><th colspan="2">>135 kBtu/h Water-Source VRF Without Heat Recovery</th><th colspan="2">>135 kBtu/h Water-Source VRF with Heat Recovery</th></tr>
<tr><td colspan="2">Representative Capacity</td><td colspan="2">216 kBtu/h</td><td colspan="2">192 kBtu/h</td></tr>
<tr><th rowspan="2">Efficiency Level</th><th rowspan="2">EER/COP</th><th colspan="4">UEC per Standard Level
kWh/yr</th></tr>
<tr><th>Cooling</th><th>Heating</th><th>Cooling</th><th>Heating</th></tr>
<tr><td>ASHRAE Standard</td><td>10.0/4.0 – VRF without heat recovery

9.8/3.9 – VRF with heat recovery</td><td>19,998</td><td>7,786</td><td>17,830</td><td>4,634</td></tr>
<tr><td>Level 1</td><td>11.0 / 4.1</td><td>18,693</td><td>7,607</td><td>16,353</td><td>4,450</td></tr>
<tr><td>Level 2</td><td>12.0 /4.3</td><td>17,606</td><td>7,440</td><td>15,348</td><td>4,309</td></tr>
<tr><td>Level 3</td><td>13.0 /4.4</td><td>16,496</td><td>7,026</td><td>14,265</td><td>4,030</td></tr>
<tr><td>Max Tech</td><td>14.5/4.6 – VRF without heat recovery

14.5/4.7 – VRF with heat recovery</td><td>15,031</td><td>6,414</td><td>12,836</td><td>3,621</td></tr>
</table>

COP = coefficient of performance; EER = energy efficiency ratio; kBtu = thousand British thermal units; UEC = unit energy consumption; VRF = variable refrigerant volume.

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