APPENDIX C. DESIGN OPTION ANALYSIS - SUPPORTING DOCUMENTATION

TABLE OF CONTENTS

LIST OF TABLES

LIST OF FIGURES

APPENDIX C. DESIGN OPTION ANALYSIS - SUPPORTING DOCUMENTATION

C.1 MODELING TOOLS

The task of modeling the performance of rooftop units and evaluating the impact of multiple design options involved the use of several different engineering models, each serving a specific role in the overall process. The primary model was the Oak Ridge National Laboratory (ORNL) Heat Pump model, which predicted the overall performance of the rooftop unit. The Department used secondary models, which accounted for certain components of the rooftop unit, to obtain necessary data to input into the ORNL model. Compressor performance data (in the form of a nine-term polynomial function) provided data on the compressor operation—namely, the refrigerant mass flow and power consumption. The Department found the air-side pressure drop across the condenser and evaporator coils using a custom heat exchanger performance model called UAIRE. The Department then used the pressure drop in conjunction with the corresponding fan or blower curve to determine the fan or blower power.

C.1.1 DOE/ORNL Heat Pump Design Model

The ORNL developed the Hypertext Markup Language (HTML)-based model, with Department of Energy support, for use as a reference tool with the ability to predict performance trends for air-to-air heat pumps and air conditioners. Available on the ORNL website at http://www.ornl.gov/~wlj/hpdm/doehpdm.html, the model accepts user input, performs selected calculations, and presents results in a comprehensive manner.

Input

The required inputs for the ORNL model are numerous, and are summarized but not explicitly listed here. The user first chooses to model either a heat pump or air-conditioning system, and selects one of 15 possible refrigerants. The model provides the following three options for calculating system refrigerant charge balance: one in which the high side of the system (in terms of either condenser exit subcooling or flow control requirements) and low-side (compressor inlet superheat) conditions are specified; and two options that allow the user to specify a refrigerant charge, and then either the high- or low-side specifications. In each case, the third, unspecified variable is calculated when the model is run. If the low side of the system is to be specified, the model allows for the use of four options for detailing flow control: condenser subcooling, capillary tube, short-tube orifice, or thermal expansion valve.

For compressor selection, the model offers ten different compressors, using four different refrigerants and with capacities ranging from 26,000 Btu/h to 36,500 Btu/h, that are preconfigured and can be used with or without modification. It is also possible to input data from any other compressor map for use in the model, using coefficients determined according to the Air-Conditioning and Refrigeration Institute (ARI) Standard 540-99, “Positive Displacement Refrigerant Compressors and Compressor Units.” In general, the variables for both the evaporator and condenser coils are extensive, and allow for both air-side and refrigerant-side specifications. The final details that the model requires as input variables are the fan locations within the unit and dimensional data for the refrigerant lines.

ORNL designed and verified the simulation model for basic air conditioners and heat pumps, but the model has optional flexibility that allows users to simulate a wider range of air-conditioner designs. The model's flexibility is, in part, attributed to scaling factors: customizable multipliers for compressor capacity, compressor power, compressor mass flow, coil heat transfer (refrigerant-side and air-side), and coil pressure drop (refrigerant- and air-side). These scaling factors enable users to customize and fine-tune the model to suit their specific needs. The Department, for example, used the scaling factors to calibrate the model for each baseline product to published test data. Once the Department established the scaling factors for each baseline, it held them constant during all of the design option simulations.

Calculations

Depending on the user-selected input, the model calculates the performance of the air-conditioning system following one of the three logic diagrams shown in Figures C.1.1 and C.1.2. Quantities calculated in this system balance are capacity (in Btu/h), sensible heat ratio (SHR), condenser heat transfer coefficient multiplied by area (UA) (Btu/h-˚F), evaporator UA (Btu/h-˚F), and refrigerant charge (lbm).

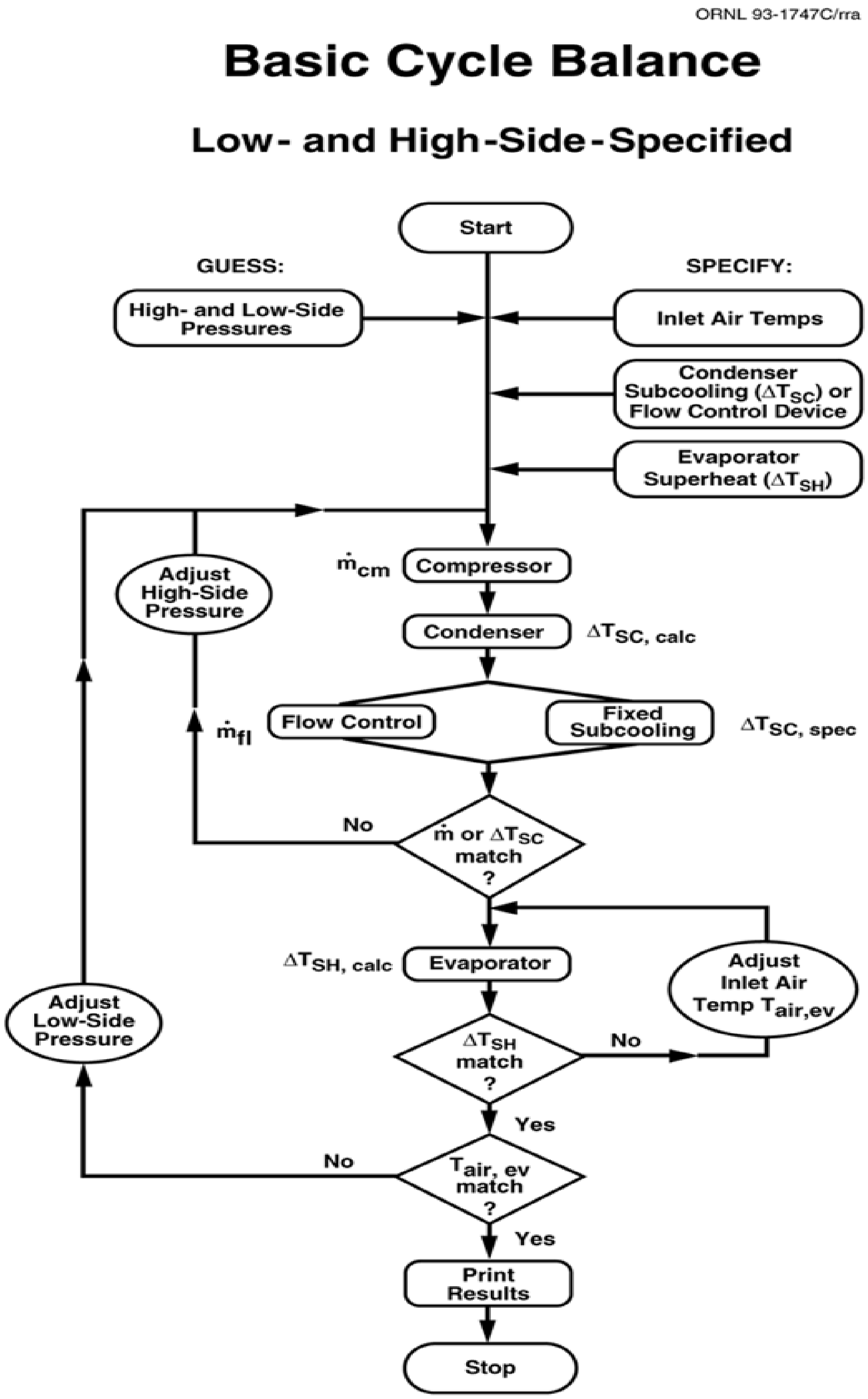


Figure C.1.1 Cycle Balance Logic Diagram

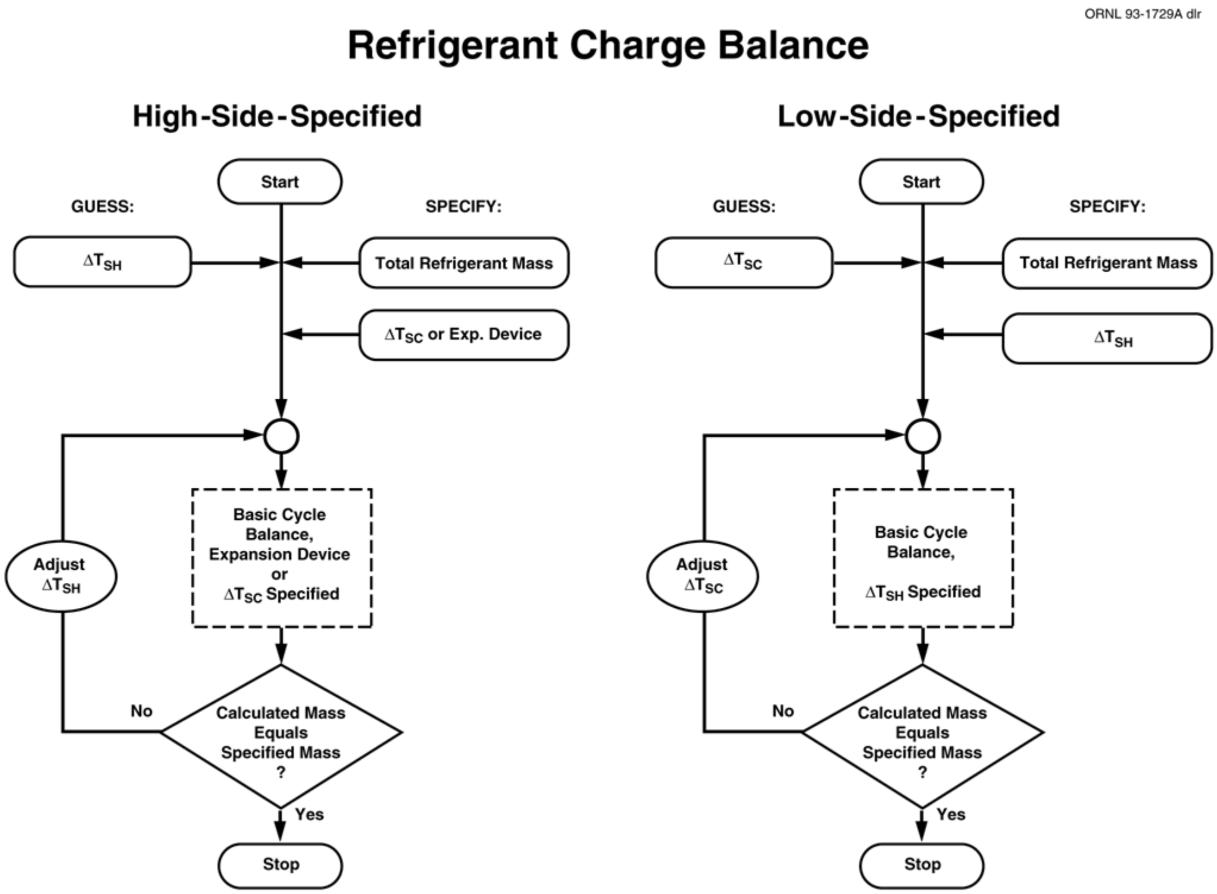


Figure C.1.2 Refrigerant Balance Logic Diagram

Presentation of Results

The results page of the ORNL model contains information on the system as a whole as well as information on each of the components. The system-wide information is based on the capacity, using that value to calculate the energy-efficiency ratio (EER) and the SHR. The EER is calculated using the system capacity and the total power, which is the sum of user-supplied power values for the condenser and evaporator fans and the calculated compressor power. The sensible capacity is calculated from the appropriate indoor air conditions, then compared to the overall system capacity. The SHR is reported as a decimal value. All of these values are presented immediately at the top of the results page, with the remainder of page one devoted to general system operating conditions and page two presenting detailed evaluations of each individual component.

The majority of the first page of results presents the refrigerant conditions at any of seven points within the system. Each point that is represented displays the refrigerant pressure, temperature, and saturation temperature, as well as properties that are specific to that point within the system. For example, at the condenser inlet, the refrigerant enthalpy and superheat are reported, and at the condenser exit, the refrigerant subcooling is shown. Also shown on the first results page is the calculated compressor mass flow and power, as well as the relevant air conditions for both the condenser and evaporator.

The second page of results provides a detailed analysis of each component of the system. Data given for the compressor include, but are not limited to, EER, capacity speed (rpm), pressure ratio, and any scaling factors or multipliers that were used. Data reported for both the condenser and the evaporator include face area, face air velocity, total air-side area, and coil overall heat transfer coefficient multiplied by the face area (UA). Details are also given for the flow-control device. The refrigerant-charge calculation results complete the page, displaying not only the total charge of the system, but the dispersion of the refrigerant throughout the system—shown as the percentage of refrigerant present in different locations.

C.1.2 Heat Exchanger Model

The custom heat exchanger model, called UAIRE, accepts input regarding the physical specifications of a heat exchanger coil (e.g., rows, area, fin pitch) and the airflow over the coil (cfm), and then calculates the associated air properties concerning airflow through the coil. The central interest in this program for the rooftop unit modeling is the calculation of air pressure drop across the coil, and the Department used the UAIRE program to find this value for both condenser and evaporator coils. The Department then used the air-side pressure drop (DPAIR) in baseline and design-option modeling.

C.1.3 Compressor Map Analysis

For both of the units, DOE obtained compressor maps from the compressor manufacturer for the exact compressors that it used. From this map, DOE created a curve fit to determine the coefficients of the following equation, from ARI 540-99:

$$F(T_S,T_D) = C_1 + C_2T_S + C_3T_D + C_4T_S^2 + C_5T_DT_S + C_6T_D^2 + C_7T_S^3 + C_8T_DT_S^2 + C_9T_ST_D^2 + C_{10}T_D^3$$

Where:

T_s is the compressor suction saturation temperature,
T_D is the compressor discharge saturation temperature, and
C_{1-10} are the performance coefficients.

The Department carried out this type of analysis for the compressor mass flow and power tables from the map (see Table C.1.1), and used the analysis accordingly in the ORNL model.

Table C.1.1 Compressor Map Values

	7.5-Ton Standard Efficiency		7.5-Ton High Efficiency		15-Ton (Standard and High Efficiency)	
Coefficients	**Mass flow**	**Power**	**Mass flow**	**Power**	**Mass flow**	**Power**
C_1	-1569.8	1762.18	-789.24	274.874	466.636	6855.719
C_2	-25.981	95.684	-12.129	32.255	42.869	33.268
C_3	50.346	-41.812	27.712	19.526	10.088	-66.524
C_4	0.0037	1.4892	0.0188	0.9154	0.0740	0.4502
C_5	0.5312	-2.4394	0.2550	-0.8211	-0.3209	-0.7235
C_6	-0.4497	0.9138	-0.2265	-0.0207	-0.0345	1.0297
C_7	0.0006	0.0027	0.0003	0.0068	0.0004	0.0012
C_8	0.00003	-0.0171	-0.0002	-0.0114	0.0011	-0.0061
C_9	-0.0020	0.0179	-0.0007	0.0051	0.0012	0.0053
C_{10}	0.0012	-0.0047	0.0005	0.0006	-0.0001	-0.0009

C.1.4 Fan Curve Analysis

Given specifications for either the evaporator or condenser coil (e.g, rows, depth, area) and airflow, DOE established a baseline air-side pressure drop for the coil using the UAIRE program. It then used the pressure drop with fan performance data provided in public product literature to determine the baseline fan power, which is typically given in terms of shaft horsepower. The Department multiplied the horsepower value by the number of fan motors in the rooftop unit (typically two condenser fans and one evaporator blower), and converted it to watts. It finally corrected this value for motor efficiency (estimated at 70 percent for condenser fans and 85 percent for an evaporator blower, unless stated otherwise), resulting in the input power to the fans.

C.2 BASELINE MODELS

C.2.1 Introduction

The Department established a baseline model for each of the four units that it used in the design option process. This baseline served as a starting point for all of the design options, and was equivalent to the published ARI rating data for each of the units. For this process, DOE gathered all input data from unit catalogs taken from each manufacturer, and from the physical teardowns of the units. Once it had entered all relevant data into the ORNL model, DOE matched the unit capacity and then the EER, determined by the total unit power. Once these three values corresponded with the published data, DOE checked other performance variables against available test data and engineering judgement to ensure that the model was reasonably predicting unit performance. Among these variables were refrigerant pressures throughout the system, superheat and subcooling values, refrigerant mass flow, and air conditions around both coils.

C.2.2 7.5-ton Standard-Efficiency Rooftop Unit

The subject of the first model was a standard-efficiency rooftop unit with a nominal capacity of 7.5 tons.

ITS Testing Results

Intertek Testing Services (ITS) performed a series of tests on the unit that was to be torn down to determine the capacity and EER at ARI (80/67 return air temperature, 95˚F ambient) and integrated part-load value (IPLV) (80/67 return air temperature, 80˚F ambient) rating conditions. The outcome of this testing indicated capacity and EER numbers better than the published levels, so the modeling process moved forward with the ITS results as the target values. The tested capacity was 92,741 Btu/h, with a 10.453 EER value (total unit power of 8872 W). ITS performed several trials at each set of test conditions and reported an extensive set of data for each trial. The data included refrigerant temperatures and pressures and air conditions recorded at various points within the system. With the new numbers set as the targets, DOE completed the modeling process as described in the following section.

Model Inputs

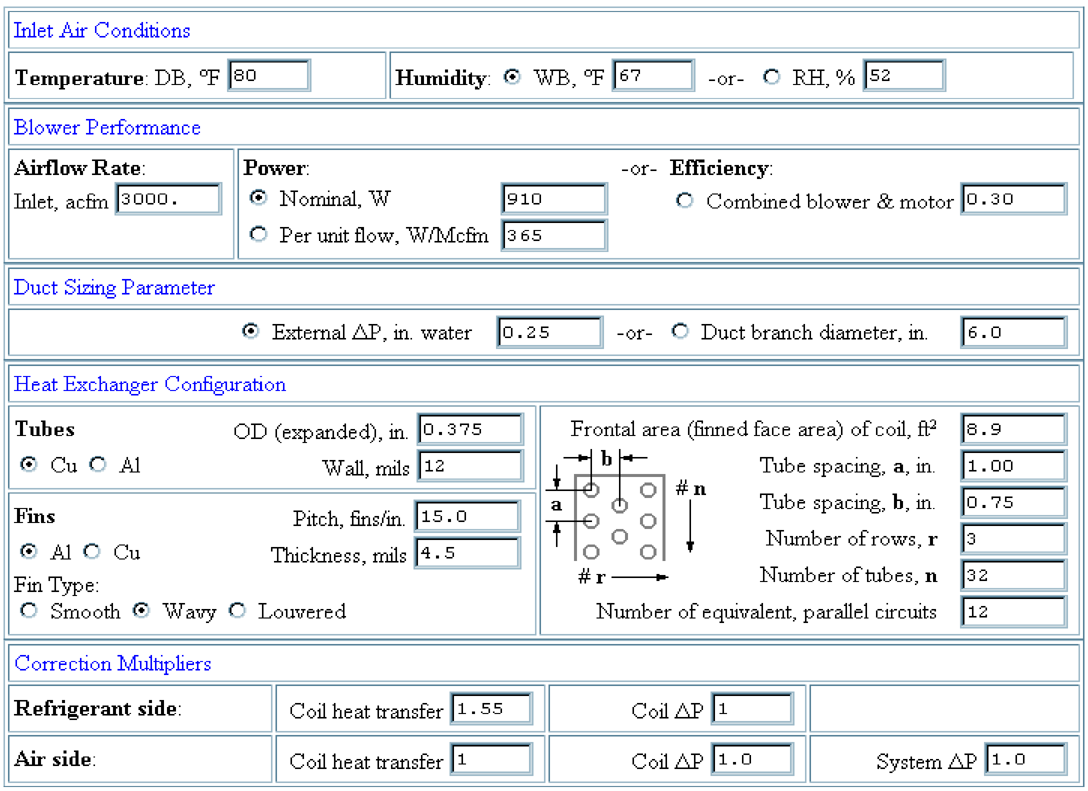
The first major piece of information that was used for this baseline model was a set of data from the appropriate compressor map. This unit has two compressors set up in a parallel arrangement. The particular compressor used in this product has a rated capacity of 41,700 Btu/h and a rated EER of 11.0. The Department input these values as needed into the ORNL model, along with coefficients for mass flow and power. The Department modeled the two compressor circuits as a single circuit with capacity, power, and mass-flow scaling factors set to 2.0.

The fixed-orifice flow control technique was selected in the model. The modeled unit used 12 short-tube orifices to regulate refrigerant flow, so the measured length and diameter of the orifices was entered along with the total number of parallel orifices in the system.

The next major information needed for the ORNL model was a complete physical description of the evaporator. The input data for the evaporator can be seen in Figure C.2.1 below, which is an image directly from the ORNL model. The user supplies all of the data that are seen in the various text boxes on the page, many of which are determined previously to being input into the model. The air conditions (temperature, humidity, and external dP, seen in Figure

C.2.1) are specified by the ARI standard, and all coil information is taken from the manufacturer catalog or from the teardown data. The evaporator blower power was taken from the ITS data, and verified with the appropriate catalog data. The last significant input is the refrigerant-side heat-transfer coefficient of 1.55. This value was supplied by Oak Ridge engineers in their version of the baseline model (discussed in a subsequent section below) and verified by the Department.

Indoor Unit Data



Inlet Air Conditions

Temperature: DB, °F 80 | **Humidity**: ◉ WB, °F 67 -or- ○ RH, % 52

Blower Performance

Airflow Rate: Inlet, acfm 3000.

Power: ◉ Nominal, W 910 ○ Per unit flow, W/Mcfm 365 -or- **Efficiency**: ○ Combined blower & motor 0.30

Duct Sizing Parameter

◉ External ΔP, in. water 0.25 -or- ○ Duct branch diameter, in. 6.0

Heat Exchanger Configuration

Tubes ◉ Cu ○ Al — OD (expanded), in. 0.375; Wall, mils 12

Fins ◉ Al ○ Cu — Pitch, fins/in. 15.0; Thickness, mils 4.5

Fin Type: ○ Smooth ◉ Wavy ○ Louvered

Frontal area (finned face area) of coil, ft² 8.9

Tube spacing, **a**, in. 1.00

Tube spacing, **b**, in. 0.75

Number of rows, **r** 3

Number of tubes, **n** 32

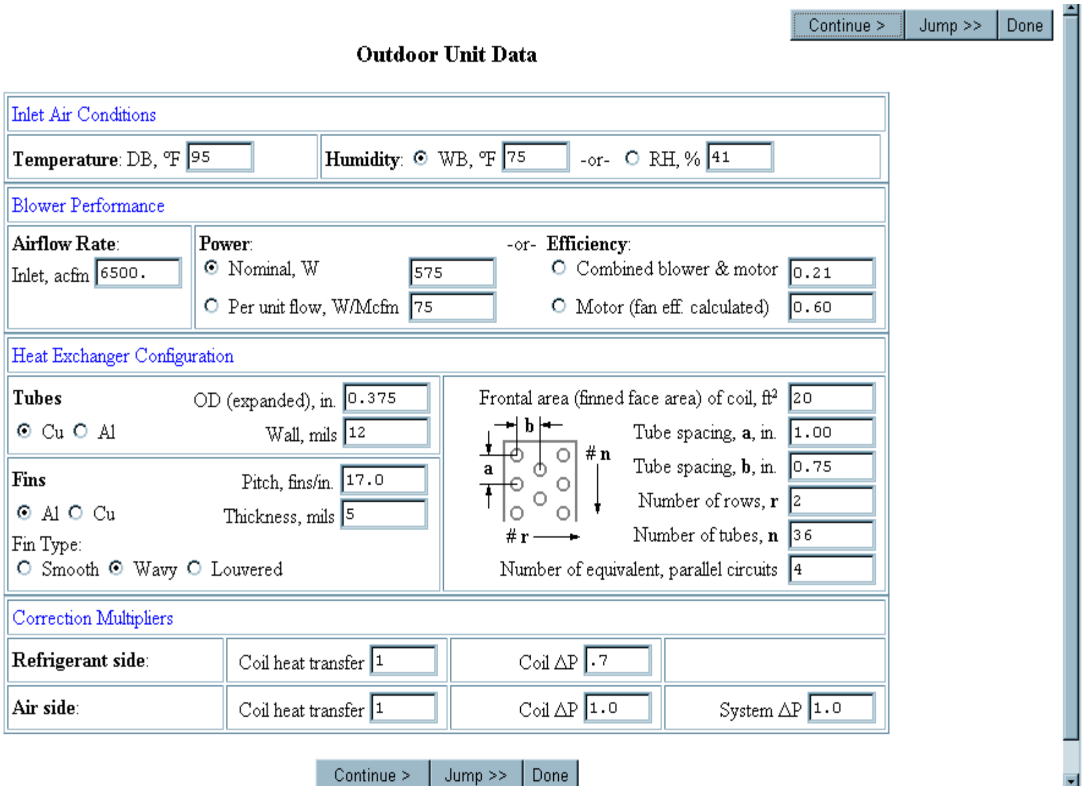
Number of equivalent, parallel circuits 12

Correction Multipliers

Refrigerant side:	Coil heat transfer 1.55	Coil ΔP 1	
Air side:	Coil heat transfer 1	Coil ΔP 1.0	System ΔP 1.0

Figure C.2.1 Evaporator Data (ORNL input page)

As with the evaporator, DOE completed a similar input page for the condenser coil; this is shown in Figure C.2.2. Again, the user supplies all information in text boxes, with the same sources for air conditions—with the exception of external dP, which is not specified for the condenser, and coil dimensions. The condenser fan power is found from the fan manufacturer's fan performance curve, and is a function of the air-side pressure drop across the coil, as determined by the UAIRE model. The refrigerant-side coil dP correction factor was set at 0.70 to correspond with the refrigerant pressure characteristics seen in the ITS data (DPCOIL ~ 10 psi).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Outdoor Unit Data

Inlet Air Conditions

Temperature: DB, °F 95 | **Humidity**: ⦿ WB, °F 75 -or- ○ RH, % 41

Blower Performance

Airflow Rate: Inlet, acfm 6500. | **Power**: ⦿ Nominal, W 575; ○ Per unit flow, W/Mcfm 75 | -or- **Efficiency**: ○ Combined blower & motor 0.21; ○ Motor (fan eff. calculated) 0.60

Heat Exchanger Configuration

Tubes ⦿ Cu ○ Al — OD (expanded), in. 0.375; Wall, mils 12

Fins ⦿ Al ○ Cu — Pitch, fins/in. 17.0; Thickness, mils 5

Fin Type: ○ Smooth ⦿ Wavy ○ Louvered

Frontal area (finned face area) of coil, ft² 20

Tube spacing, **a**, in. 1.00

Tube spacing, **b**, in. 0.75

Number of rows, **r** 2

Number of tubes, **n** 36

Number of equivalent, parallel circuits 4

Correction Multipliers

Refrigerant side:	Coil heat transfer 1	Coil ΔP .7	
Air side:	Coil heat transfer 1	Coil ΔP 1.0	System ΔP 1.0

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Figure C.2.2 Condenser Data (ORNL input page)

The last inputs involve unit configuration and the refrigerant-line data. For this unit, the indoor blower, condenser fan, and the compressor shell are located after the respective coil. This option was chosen in the ORNL model, which adds the appropriate waste heat for each device to the airstream. Due to the method of modeling two circuits with one compressor, the refrigerant line flow area was doubled and the overall length cut in half. This was an attempt to correctly predict pressure drop within the refrigerant line.

Model Results

With the data input as described in the preceding section, the model yields results that are quite close to the values reported in the ITS data—a calculated capacity of 91,037 Btu/h (1.8 percent difference from ITS) and an EER of 10.334 (1.1 percent difference). These results were within the range of error allowed in the ARI test standard. Figure C.2.3 shows a comparison of the data.

	Predicted by Model	Measured by ITS Testing[1]	*Difference*
Capacity (Btu/hr)	91,037	92,741[2]	*-1.8%*
EER	10.334	10.45[2]	*-1.1%*
Compressor Power (W)	7,325	7,430 +/- 2.0%	*-1.4%*
Evaporator Blower Power (W)	910	880 +/- 2.0%	*+3.4%*
Condenser Fan Power (W)	575	562 +/- 2.0%	*+2.3%*
Evaporator Pressure (PSIA)	96.8	98.7 +/- 2.0%	*-1.9%*
Evaporator Temperature (°F)	48.8	Average: 50[2] (48.7 - 51.2)[3]	*1.2 °F*
Condenser Pressure (PSIA)	279.2	282.2 +/- 2.0%	*-1.1%*
Condenser Temperature (°F)	121.3	Average: 122.1[2] (120.6 - 123.7)[3]	*0.8 °F*

[1] Using Test "A" methods
[2] Calculated from measured data
[3] Range calculated from pressure error band

Figure C.2.3 Model Calibration with Test Data

Oak Ridge Participation

Oak Ridge National Laboratory engineers simultaneously completed their own version of this baseline model. Because these engineers are continuously involved with all details of their HTML-based model, the Department considered their input valuable as an addition to its own work. The model the ORNL engineers created utilized a different, but equivalent, approach to modeling the dual compressors (doubling the compressor map before finding coefficients, then using scaling factors set to one), but modeled the rest of the unit in essentially the same manner. The results were slightly lower than the final values in the model and closer to the published ratings with a capacity of 89,500 Btu/h (0.36 percent lower than the model), and an EER of 10.11 (0.72 percent lower than the model's value).

C.2.3 7.5-ton High-Efficiency Rooftop Unit

Modeling the 7.5-ton high-efficiency unit was much the same as the standard efficiency, due to the vast similarity of the two units. Most of the equipment in the unit (e.g., condenser and evaporator coils, condenser fans) was the same in both of the units; the only major difference was a change from a reciprocating compressor to a scroll compressor. The scroll technology offers greater efficiency than the reciprocal compressor and this is reflected in the compressor coefficients used in the model. The Department used the same approach to model the dual-compressor configuration, where it set scaling factors for capacity, mass flow, and power equal to 2.0. The compressor's rated capacity is 45,500 Btu/h with an EER of 11.4.

The flow control device and coil characteristics (both evaporator and condenser) for this unit were the same as that of the standard-efficiency unit described in Section B.2.2.

The high-efficiency unit has a more powerful evaporator fan blower, rated at 2.9 horsepower, compared to the standard-efficiency unit's 2.4-horsepower motor. This difference has little impact on the modeling, however, because the ARI rating is measured at standardized air conditions (dP External = 0.25 inches of water). This pressure setting, along with the use of the same evaporator coil, ensures that the evaporator blower power will remain relatively constant despite the higher peak motor horsepower rating.

The results of the model were very close to the published values: model capacity is 90,148 Btu/h and the EER 11.029 (total power of the unit is 8173.4 W).

C.2.4 15-ton Standard-Efficiency Rooftop Unit

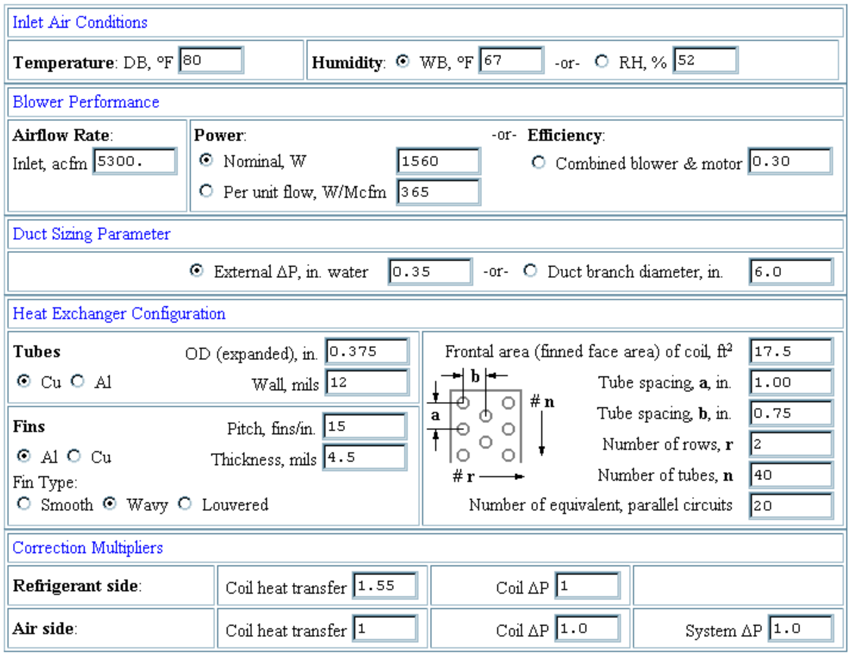
Modeling the compressors in the 15-ton unit presented a challenge that had not been previously encountered, since the unit contains one 10-ton compressor and one 5-ton compressor. The Department could not model these as it had the 7.5-ton units; instead, with an adjusted scaling factor, it combined the performance maps for both compressors to model a single compressor. The Department obtained coefficients for the capacity, mass flow, and power of each compressor, and created a standard performance map for each compressor. It then added corresponding values from each of the two maps, simulating the two compressors operating simultaneously under the same conditions. The Department performed a regression analysis (see section B.1.3) on the resultant maps for mass flow and power, and entered these coefficients into the ORNL model. The Department also determined a standard capacity from the combined map, and entered it into the model.

Considering the approximations made with the compressor modeling, it was necessary to use the scaling factors in the model to achieve the desired unit performance. Of particular importance were the compressor mass flow and power-scaling factors. Through an iterative process, DOE adjusted the mass flow so the capacity would approach 174,000 Btu/h. The Department also adjusted the compressor power to make a change in the overall unit power and, in effect, the unit EER, because condenser fan power and evaporator blower power were fixed. The Department set the mass flow factor at 1.102, and the power factor at 1.125. Both remained constant for all subsequent design option modeling.

As with the 7.5-ton units, this system uses a short-tube orifice to regulate the refrigerant flow through the evaporator coil. The Department entered the measured orifice information into the model.

The input data for the evaporator and condenser coils appear in Figures C.2.4 and C.2.5, respectively; they were taken from the manufacturer's catalog and the teardown data. The evaporator blower power of 1560 W comes directly from the manufacturer's catalog for airflow and pressures specified by the ARI test standard.

Indoor Unit Data



Inlet Air Conditions

Temperature: DB, °F 80 | **Humidity**: ◉ WB, °F 67 -or- ○ RH, % 52

Blower Performance

Airflow Rate: Inlet, acfm 5300.

Power: ◉ Nominal, W 1560; ○ Per unit flow, W/Mcfm 365 -or- **Efficiency**: ○ Combined blower & motor 0.30

Duct Sizing Parameter

◉ External ΔP, in. water 0.35 -or- ○ Duct branch diameter, in. 6.0

Heat Exchanger Configuration

Tubes: ◉ Cu ○ Al — OD (expanded), in. 0.375; Wall, mils 12

Fins: ◉ Al ○ Cu — Pitch, fins/in. 15; Thickness, mils 4.5

Fin Type: ○ Smooth ◉ Wavy ○ Louvered

Frontal area (finned face area) of coil, ft² 17.5

Tube spacing, **a**, in. 1.00

Tube spacing, **b**, in. 0.75

Number of rows, **r** 2

Number of tubes, **n** 40

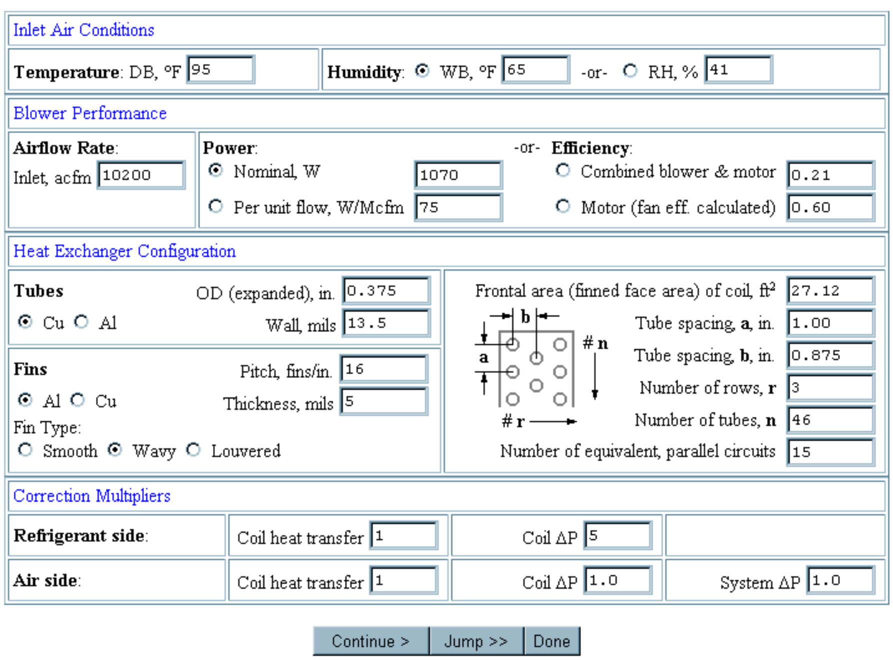
Number of equivalent, parallel circuits 20

Correction Multipliers

Refrigerant side:	Coil heat transfer 1.55	Coil ΔP 1	
Air side:	Coil heat transfer 1	Coil ΔP 1.0	System ΔP 1.0

Figure C.2.4 Evaporator Data (ORNL input page)

Outdoor Unit Data



Inlet Air Conditions

Temperature: DB, °F 95 | **Humidity**: ◉ WB, °F 65 -or- ○ RH, % 41

Blower Performance

Airflow Rate: Inlet, acfm 10200

Power: ◉ Nominal, W 1070; ○ Per unit flow, W/Mcfm 75 -or- **Efficiency**: ○ Combined blower & motor 0.21; ○ Motor (fan eff. calculated) 0.60

Heat Exchanger Configuration

Tubes: ◉ Cu ○ Al — OD (expanded), in. 0.375; Wall, mils 13.5

Fins: ◉ Al ○ Cu — Pitch, fins/in. 16; Thickness, mils 5

Fin Type: ○ Smooth ◉ Wavy ○ Louvered

Frontal area (finned face area) of coil, ft² 27.12

Tube spacing, **a**, in. 1.00

Tube spacing, **b**, in. 0.875

Number of rows, **r** 3

Number of tubes, **n** 46

Number of equivalent, parallel circuits 15

Correction Multipliers

Refrigerant side:	Coil heat transfer 1	Coil ΔP 5	
Air side:	Coil heat transfer 1	Coil ΔP 1.0	System ΔP 1.0

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Figure C.2.5 Condenser Data (ORNL input page)

The Department changed the refrigerant-side pressure drop scaling factor for the condenser to five, in order to maintain a 10-psi refrigerant drop through the coil. The condenser fan power calculation considers the two 0.5-hp condenser fans operating within the unit, resulting in a power input of 1070 W.

The final capacity of the unit, as determined through this process, was 174,091 Btu/h (+0.05 percent, compared to published rating), with an EER value of 9.703 (+0.03 percent, from the published rating).

C.2.5 15-ton High-Efficiency Rooftop Unit

The high-efficiency unit uses the same compressors as the standard unit, but has larger and deeper heat exchanger coils. As a result, DOE inserted the compressor coefficients that it used for the standard unit directly into the high-efficiency model.

A short-tube orifice is used in this unit, as with the standard-efficiency unit, to regulate the flow of refrigerant into the evaporator. Again, DOE entered the measured orifice parameters into the model.

As stated above, both the evaporator and condenser coils were quite different from those used in the standard-efficiency unit. The data pertaining to the coils are shown below in Figures C.2.6 and C.2.7.

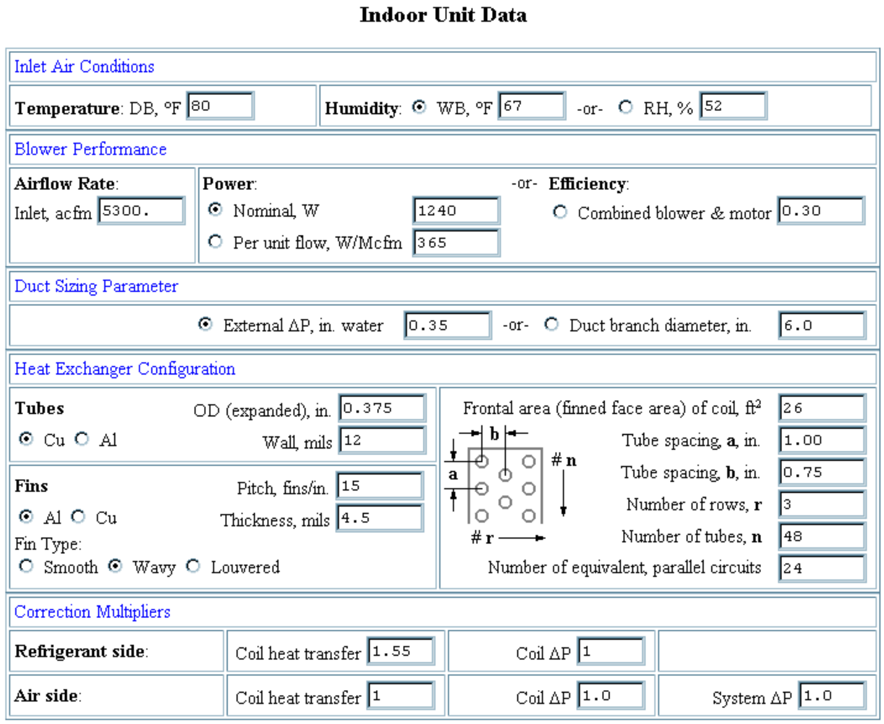

Figure C.2.6 Evaporator Data (ORNL Input page)

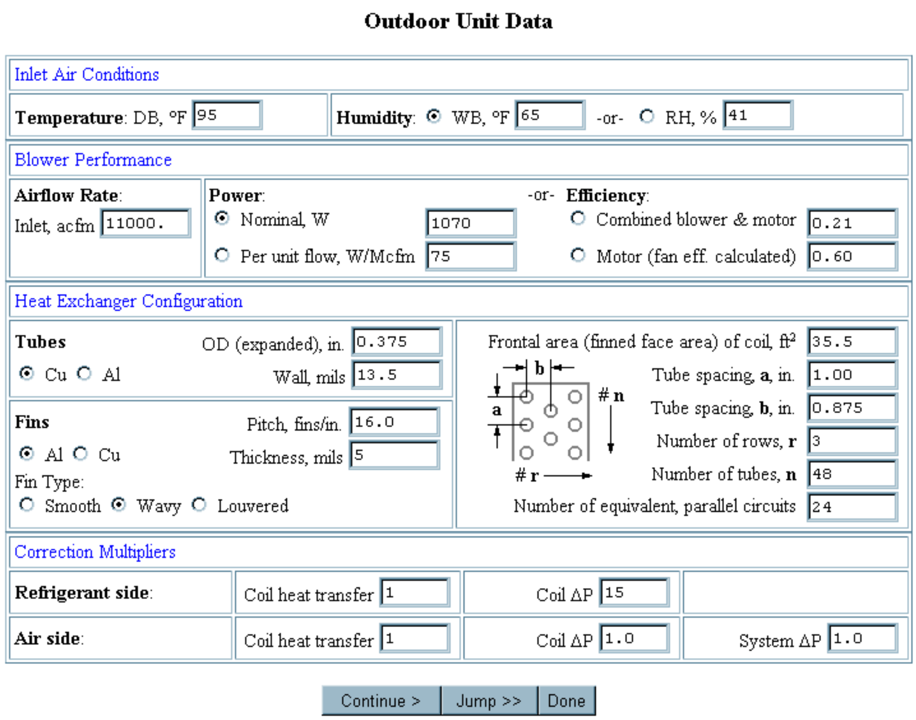

Figure C.2.7 Condenser Data (ORNL input page)

The final capacity of the unit as determined through this process was 180,984 Btu/h (+0.55 percent, compared to the published rating), with an EER value of 11.507 (+0.06 percent, from the published rating).

C.3 DESIGN MODIFICATIONS

C.3.1 Design Paths

The object of the design option simulation process was to increase the efficiency of each rooftop unit to at least a 12.0 EER value, while staying within performance limits established by the Department in discussions with manufacturers. The performance limits that were maintained are presented below in Table C.3.1 and, in almost every case, the modeled rooftop units were enhanced up to these limits. These performance limits prevented some of the design option units—essentially those models that were built on a standard-efficiency baseline unit—from reaching the target efficiency level. This is why the Department did not consider the standard-efficiency results for the presentation in Section 5.8. These results are included here for documentation purposes, and in order to add a further level of clarity to the design-modification approach.

The process of modeling the design modifications is described in detail in this section for each unit. For all units, DOE set the target efficiency level at 12.0 and modified the units until this level was reached or until the unit reached the stated operating limits. The Department completed modifications independently on each of the four baseline units; the process for each will be discussed in the following sections.

Table C.3.1 Design Constraints

Compressors	a.	With multiple compressors of the same size, each acts with the same performance characteristics. This enables the use of scaling factors.
	b.	Compressors are located after the condenser coil, so waste heat is added to the exiting condenser air stream.
	c.	Compressor capacity is scaled to maintain relatively constant system capacity. DOE performed compressor scaling in discrete increments to use capacities of commercially available compressors at all times.
Evaporator	d.	Evaporator blower motor efficiency is assumed to be 85%.
	e.	Refrigerant side heat transfer "correction multiplier" is 1.55 (taken from ORNL participation).
	f.	Evaporator blower is located after the coil, so the motor waste heat is added to the indoor air stream.
	g.	Maximum evaporator temperature is 52˚F.
	h.	Maximum sensible heat ratio is 0.750.
Condenser	i.	Condenser fan motor efficiency is assumed to be 70%.
	j.	Refrigerant pressure drop across the coil is maintained at approximately 10 psi. This is accomplished by changing the number of parallel circuits in the condenser coil when the pressure drop increases beyond this level.
	k.	Condenser fans are located after the coil, so the motor waste heat is added to the leaving condenser air stream.
	l.	Minimum condenser temperature is 117˚F for R-22, and 116˚F for R-410A.
	m.	Minimum refrigerant exit temperature is 102˚F, for each refrigerant.
Refrigerant Lines	n.	If there are parallel refrigerant circuits, the total area (sum of individual areas) is maintained by assuming one line, then setting the diameter to the necessary size for this larger tube. The tube length is kept at the original value.
Short Tube Orifice	o.	This method of flow control is used in every model, although actual systems may used a TXV (R-410A units especially). The orifice can approximate a TXV that is in a fixed position, as it would be at full load during ARI testing.
	p.	Diameter is variable to four significant figures.
	q.	Length is variable (only used as a coarse adjustment), although typically held constant.

C.3.2 7.5-ton Standard-Efficiency Unit

The standard-efficiency unit did not offer a high probability of making the jump from the 10.1 ARI rated EER to a 12.0, so DOE used a slightly different approach for the design option simulation process. Rather than attempting to attain a 12.0 EER, DOE modeled four different changes to determine the effect each would have on the unit performance. This information proved useful later when it was considered while choosing the specific paths that DOE selected to enhance the high-efficiency units.

The Department completed four separate design modifications for this baseline unit, all of which focused on the heat exchanger coils. For each coil, DOE increased the depth by one row (models 1 and 3), and then increased the coil face area along with the depth increase (models 2 and 4). The first two models allowed the unit to remain in the same box size, because the dimensions are more dependent on coil area than on coil depth, but the last two models included a box size "jump" (increase in box size to the next available volume). Each model is described below and a summary of the results is in Figure C.3.1.

Condenser Depth Increase

The addition of one row to the condenser coil of this unit resulted in a three-row coil and increased the ΔP_{AIR} by 0.0478 inches of water, which in turn decreased the airflow across the coil, 6200 cfm, and slightly increases the condenser fan power to 610 W. The larger condenser coil also initially resulted in an increase in overall unit capacity, but this was maintained by scaling back the compressor capacity by 2.5 percent to 40.6 kBtu/h. The result was a unit with a 10.62 EER, almost 0.3 EER higher than the baseline.

Condenser Area Increase

The condenser coil was increased by 25 percent, resulting in a 25-ft^2 face area and necessitating a box-size jump. The Department also added one row to the coil to maximize the EER gain. The resulting three-row coil had a ΔP_{AIR} that was slightly higher than the baseline, since the effects of the additional area and the added depth offset each other. The effect of this marginal increase in ΔP_{AIR} was a decrease in condenser airflow to 6400 cfm from the 6500 cfm baseline, with a negligible effect on fan power. The condensing temperature was 117.5˚F, and the exit temperature was 102.7˚F. In order to maintain capacity of this unit, DOE scaled back the compressor capacity by 2 percent to 41.1 kBtu/h. The result was a unit with a 10.77 EER, more than 0.4 EER higher than the baseline.

Evaporator Depth Increase

The addition of one row to the evaporator caused the ΔP_{AIR} to increase by 0.0428 inches of water, which translated to a 55 W jump in evaporator blower power to 965 W. The Department scaled back the compressors by 5 percent to account for the additional unit capacity that was caused by the enhanced evaporator, to a total value of 39.4 kBtu/h. The constraints pertaining to this modification were the maximum allowable evaporator temperature, and the maximum SHR. This unit pressed the allowable levels, but did not exceed them, as the evaporating temperature was 51.6˚F and the SHR was 0.745. The result was a unit with a 10.59 EER, almost 0.3 EER higher than the baseline.

Evaporator Area Increase

The Department increased the evaporator area by 25 percent for this design option, resulting in a face area of 11.125 ft^2 and a ΔP_{AIR} that was 0.0437 inches of water less than the baseline. The lower ΔP_{AIR} resulted in a blower power that was 50 W lower than the baseline value of 910 W. Scaling the compressors back by 4 percent to a capacity of 40.0 kBtu/h controlled the capacity. Since the evaporator conditions were already approaching the design constraints, DOE made no further modifications to the evaporator. The evaporating temperature was 50.7˚F, and the SHR 0.736. The result was a unit with a 10.63 EER, 0.3 EER higher than the baseline.

C.3.3 7.5-ton High-Efficiency Unit

The 7.5-ton high-efficiency unit had a rated capacity of 90,000 kBtu/h and an EER of 11.0, and served as the baseline for the design option process that yielded the target 12.0 EER unit. In order to achieve this gain in efficiency, DOE modified the condenser coil, the evaporator coil, and the compressors. The final result was a unit with a capacity of 90,350 Btu/h and an EER of 12.05.

Condenser Area Increase

Increasing the condenser coil face area by 50 percent and adding a row resulted in a drop in ΔP_{AIR} of 0.0473 inches of water, which effected a 300 cfm increase in condenser airflow (6500 cfm, baseline) and a 60 W drop in fan power (620 W, baseline). The Department scaled down the compressors for this option by 2 percent to 42.0 kBtu/h to correct for the initial increase in unit capacity. The condensing temperature, 117.8˚F, and exit temperature, 102.1˚F, were within the design constraints. The result was a unit with an 11.71 EER, almost 0.6 EER higher than the baseline.

Condenser Depth Increase

The Department added one row to the condenser coil, bringing the total to three rows, which caused a rise in ΔP_{AIR} of 0.0478 inches of water. The change in ΔP_{AIR} led to a slightly lower condenser airflow value of 6200 cfm (6500 cfm, baseline) and an increase in fan power of 35 W (620W, baseline). Along with the coil change, DOE scaled down the compressors by 2 percent to 42.0 kBtu/h, in order to maintain the rated capacity. The condensing temperature for this unit was 118.5˚F and the exit temperature was 102.3˚F, both above the minimum values. The result was a unit with an 11.51 EER, almost 0.4 EER higher than the baseline.

__Design Combination__

Since the condenser area increase yielded the largest gain in unit EER, DOE chose that option to build on, in order to design a 12.0 EER unit. The condenser area increase made a jump in box size necessary, so the next step to increase unit performance would be an increase in the face area of the evaporator. Thus, additional cost beyond that of the coil itself would not be incurred.

The addition of the evaporator modification to this unit caused an increase in system capacity as well as EER, which was handled by scaling the compressors back again. The compressors chosen for this unit were 6 percent smaller in capacity than the baseline, rated at 40.6 kBtu/h, and each had the same EER. The unit had an evaporating temperature of 50.7˚F and a SHR of 0.739, which met design constraints. The result was a unit with a 12.05 EER, almost 0.9 EER higher than the baseline. The results are summarized in Figure C.3.1. below.

Approach	Baseline	Condenser Area +50% Area	Condenser Depth +1 Row		Area Combination +50% Cond., +25% Evap.
Product Specs	11.14-EER	+0.57-EER	+0.37-EER		+0.91-EER
Physical Characteristics		box: +70% volume condenser: +50% area compressor: 2 x 42.0 kBtu/hr	condenser: +1 row Compressor: 2 x 42.0 kBtu/hr		box: +70% volume condenser: +50% area evaporator: +25% area compressor: 2 x 40.6 kBtu/hr
Performance Characteristics	Tc = 121.0 F Te = 48.9 F Subcool = 10.8 F Superheat = 20 F Compressor: 6883 W Fan: 620 W Blower: 700W SHR = 0.728	Tc = 117.8 F Te = 48.8 F Subcool = 11.7 F Superheat = 20 F Compressor: 6533 W Fan: 560 W +300 CFM Blower: 700W SHR = 0.729	Tc = 118.5 F Te = 48.9 F Subcool = 11.9 F Superheat = 20 F Compressor: 6545 W Fan: 655 W -300 CFM Blower: 700W SHR = 0.730		Tc = 117.4 F Te = 50.7 F Subcool = 11.2 F Superheat = 20 F Compressor: 6289 W Fan: 560 W +300 CFM Blower: 650W SHR = 0.739

Figure C.3.1 7.5-ton High-Efficiency Baseline and Design Options

C.3.4 15-ton Standard-Efficiency Unit

As with the 7.5-ton standard-efficiency unit, the 15-ton standard-efficiency unit did not offer a high probability of making the jump from the 9.7 ARI rated EER to 12.0 EER. With this in mind, DOE modeled the same four changes as with the 7.5-ton unit to determine the effect each would have on the unit performance. These changes were the same for both the condenser and evaporator coils, and included a depth increase of one row (models 1 and 3), and then a coil face area increase which may or may not also have included a depth increase (models 2 and 4). Each model is described below, and a summary of the results is seen in Figure C.3.2.

Condenser Depth Increase

The additional row on the condenser coil of this unit made the coil depth four rows, and the corresponding ΔP_{AIR} increase by 0.0484 inches of water. In addition to the coil modification, DOE changed the condenser fans to produce a higher airflow over the coil. The Department increased the fan blade diameter to 28" (26", baseline), and applied the new ΔP_{AIR} to the larger fan curve to find the appropriate airflow and fan power values. The Department determined the new airflow to be 11800 cfm (10200 cfm, baseline), and the new fan power to be 1215 W (1070 W, baseline). These changes resulted in an increase in overall unit capacity, but this was maintained by scaling back the total compressor capacity by 3.4 percent to 168.0 kBtu/h. As with the earlier models, the design constraints that limited this option were the minimum condensing temperature and the minimum condenser exit temperature. Both of these limits were maintained, as the condensing temperature was 121.3˚F, and the exit temperature 102.4˚F. The result was a unit with a 10.47 EER, almost 0.8 EER higher than the baseline.

Condenser Area Increase

The Department increased the condenser coil face area by 35 percent to 36.6 ft^2, and added one row to make a four-row coil. The resulting ΔP_{AIR} was slightly lower than the baseline, and was again used to find airflow and fan power required by the 28" diameter condenser fans that were used in place of the 26" fans in the baseline unit. The effect of these changes was an airflow value of 12300 cfm (10200 cfm, baseline), and a fan power of 1400 W (1070 W, baseline). The Department changed the compressors in this unit to maintain overall capacity, and scaled them back by 3.8 percent to 167.4 kBtu/h. The condensing temperature was 118.2˚F, and the exit temperature was 102.2˚F. The result was a unit with a 10.71 EER, over 1.0 EER higher than the baseline.

Evaporator Depth Increase

The Department increased the evaporator by one row in depth, making the coil three rows deep. In addition to the depth increase, DOE changed the evaporator blower to an 18" blower, which nearly cut the blower power in half. The ΔP_{AIR} that corresponded to the coil change increased by 0.0268 inches of water when the airflow was set at 5100 cfm (5300 cfm, baseline), and the evaporator blower power was 830 W. The Department scaled back the compressors by 9 percent to account for the additional unit capacity that was caused by the enhanced evaporator, to a total value of 158.3 kBtu/h. The constraints that pertained to this modification were the maximum allowable evaporator temperature, and the maximum SHR. This unit was easily within the allowable levels, since the evaporating temperature was 50.0˚F and the SHR was 0.717. The result was a unit with a 11.11 EER, over 1.4 EER higher than the baseline.

Evaporator Area Increase

The Department employed a 50 percent increase in evaporator face area for this design option, resulting in a 26.25 ft^2 coil with a ΔP_{AIR} that was 0.0256 inches of water lower than the baseline. With the lower ΔP_{AIR}, DOE increased the evaporator blower to an 18" blower (15", baseline) to maximize the efficiency gain. This change resulted in a blower power of 790 W, slightly more than half the baseline value. The Department scaled back the compressors by 8 percent, to a capacity of 160.0 kBtu/h, to control the capacity. The evaporator conditions were well within the limits, with an evaporating temperature of 49.4˚F, and the SHR 0.711. The result was a unit with an 11.05 EER, more than 1.3 EER higher than the baseline. Figure C.3.2 shows the results.

Approach	Baseline	Condenser Area +35% Area, +1 Row	Condenser Depth +1 Row (4 total)	Evaporator Area +50% Area	Evaporator Depth +1 Row (3 total)
Product Specs	9.7-EER	+1.01 -EER	+0.77 -EER	+1.35 -EER	+1.41 -EER
Physical Characteristics		**box:** +46% volume **condenser:** +35% area, +1 row **condenser fans:** +2' diameter **compressor:** 2 x 84.0 kBtu/hr	**condenser:** +1 row **condenser fans:** +2' diameter **compressor:** 2 x 84.0 kBtu/hr	**box:** +46% volume **evaporator:** +50% area **compressor:** 2 x 87.5 kBtu/hr	**evaporator:** +1 row **compressor:** 2 x 87.5 kBtu/hr
Performance Characteristics	**Tc** = 126.7 F **Te** = 45.5 F **Subcool** = 15.1 F **Superheat** = 10 F **Compressor:** 15,312 W **Fan:** 1070 W 10,200 CFM **Blower:** 1560 W/ 5300 CFM SHR = 0.694	**Tc** =118.2 F **Te** =45.3 F **Subcool** =13.2 F **Superheat** =10 F **Compressor: 13,296 W** **Fan: 1400 W 12,300 CFM** **Blower:** 1560 W/ 5300 CFM SHR =0.694	**Tc** =121.3 F **Te** =45.3 F **Subcool** =16.3 F **Superheat** =10 F **Compressor: 13,857 W** **Fan: 1215 W 11,800 CFM** **Blower:** 1560 W/ 5300 CFM SHR =0.694	**Tc** =125.9 F **Te** =49.4 F **Subcool** =14.4 F **Superheat** =10 F **Compressor: 13,964 W** **Fan: 1070 W 10,200 CFM** **Blower:** 790 W 5300 CFM SHR =0.711	**Tc** =125.7 F **Te** =50.0 F **Subcool** =14.3 F **Superheat** =10 F **Compressor: 13,795 W** **Fan: 1070 W 10,200 CFM** **Blower:** 830 W 5100 CFM SHR =0.717

Figure C.3.2 15-ton Standard-Efficiency Baseline and Design Options

C.3.5 15-ton High-Efficiency Unit

The high-efficiency 15-ton unit offered a couple of different options in order to get to a 12.0 EER, since it was already close to that efficiency level at the baseline rating of 11.5. The Department explored three different possibilities, one falling just short of the desired efficiency and the other two exceeding the target. The first design option was an increase in evaporator depth, the second an increase in condenser depth, and the last an increase in condenser area.

Evaporator Depth Increase

Since the baseline model for this unit already had an evaporator temperature of 51.5˚F, there was very little room for evaporator modification while keeping within the design constraints. For this reason, DOE modeled a one-row evaporator depth increase to determine whether it was a feasible design choice to modify the evaporator at all. The one-row increase brought the coil to four rows in all, and caused an increase of 0.0117 inches of water in ΔP_{AIR}. Along with this change, DOE reduced the evaporator airflow to 5000 cfm (5300, baseline), and scaled down the compressors by 2 percent to 176.4 kBtu/h. These changes did cause an increase in evaporator temperature over the baseline, to 52.7˚F, which is higher than the applicable limit. However, the SHR was much lower than the maximum level with a value of 0.717. Since the

SHR was well within the limit, DOE accepted the high evaporator temperature. This choice of allowing an elevated evaporator temperature was relatively inconsequential, however, since the modified unit did not reach the target EER level. The result was a unit with an 11.81 EER, 0.3 EER higher than the baseline.

Condenser Depth Increase

This design option increased the condenser depth from three rows to four, and included an increase in condenser fan diameter, as well as a slight change in compressors. The depth increase caused a jump in ΔP_{AIR}, which DOE calculated at the higher airflow that accompanied the 28" fans (26", baseline). The Department calculated the ΔP_{AIR} to be 0.0315 inches of water higher than the baseline, at an airflow of 12600 cfm (11000 cfm, baseline). Along with the condenser changes, DOE scaled down the compressor capacity by 2 percent to 176.4 Btu/h in order to maintain the overall unit capacity. After all of these changes were modeled, DOE verified the condenser temperature limits: the condensing temperature was 119.8˚F, and the exit temperature was 102.7˚F. The result was a unit with a 12.15 EER, over 0.6 EER higher than the baseline.

Condenser Area Increase

This design approach involved finding the minimum condenser area increase that would enable the unit to meet the 12.0 EER target. A 10 percent increase in coil face area, accompanied by a 2" increase in condenser fan diameter and a 2 percent decrease in compressor capacity, accomplished this goal. The face area increase resulted in a ΔP_{AIR} that was slightly higher than the baseline, when considered with the increase in fan size, and made the total condenser face area 39.05 ft^2. The Department used a total compressor capacity of 176.4 kBtu/h in the model, with a condenser airflow value of 13800 cfm (11000 cfm, baseline). The Department checked the condenser temperatures to verify unit performance, and met the design limits: the condensing temperature was 120.3˚F, and the exit temperature 103.1˚F. The result was a unit with a 12.03 EER, over 0.5 EER higher than the baseline. Figure C.3.3. summarizes the results.

Approach	Baseline	Condenser Area +10% Area	Condenser Depth +1 Row	Evaporator Depth +1 Row	
Product Specs	11.5-EER	+0.5-EER	+0.7-EER	+0.3-EER	
Physical Characteristics		**condenser:** +10% area **condenser fans:** +2' diameter **compressor:** 1 x 58.5 kBtu/hr 1 x 115.0 kBtu/hr	**condenser:** +1 row **condenser fans:** +2' diameter **compressor:** 1 x 58.5 kBtu/hr 1 x 115.0 kBtu/hr	**evaporator:** +1 row **compressor:** 1 x 58.5 kBtu/hr 1 x 117.0 kBtu/hr	
Performance Characteristics	**Tc** = 125.37 F **Te** = 51.5 F **Subcool** = 16.4 F **Superheat** = 10 F **Compressor:** 13,418 W **Fan:** 1070 W **Blower:** 1240 W SHR = 0.722	**Tc** =120.3 F **Te** =51.5 F **Subcool** =14.0 F **Superheat** =10 F **Compressor: 12,344 W** **Fan:** 1450 W+2,800 CFM **Blower: 1240 W** SHR=0.723	**Tc** =119.8 F **Te** =51.4 F **Subcool** =14.3 F **Superheat** =10 F **Compressor: 12,283 W** **Fan:** 1365 W+1,600 CFM **Blower: 1240 W** SHR=0.723	**Tc** =125.0 F **Te** =52.7 F **Subcool** =16.2 F **Superheat** =10 F **Compressor: 13,115 W** **Fan:** 1070 W **Blower: 1105 W-300 CFM** SHR=0.719	

Figure C.3.3 15-ton High-Efficiency Baseline and Design Options

C.4 R-410A ANALYSIS

C.4.1 7.5-ton R-410A Unit

The Department modeled the 7.5-ton R-410A unit to match the baseline performance of the 7.5-ton standard-efficiency unit (see A.2.2), since there are no commercially-available units that could be used as they were in the R-22 analysis. The Department constructed a baseline model for the 410A unit, using physical characteristics that were common in R-22 units of the same size. These characteristics included details about the condenser and evaporator coils, fans and blowers, refrigerant lines, and unit configurations. The only components that DOE used for the baseline models that were not based on the R-22 units were the compressors. The Department selected these from Copeland's line of R-410A compressors, which are currently commercially available. Once it had established valid baseline models, DOE completed the design option modeling following the same methodology as it used in the R-22 analysis.

7.5-ton R-410A Baseline Model

The evaporator coil was 8.9 ft^2 and 3 rows deep, with a 15" evaporator blower. The condenser coil was 16.5 ft^2 and 2 rows deep, with two 22" condenser fans powered by 0.25 hp motors. Refrigerant line sizes were identical to those of the R22 unit, and the flow was controlled with a short tube orifice. The short tube orifice is not typically used with R-410A units, based on available residential units that range in size up to five tons, but instead a thermal expansion valve is employed. For the modeling process, the short tube orifice can simulate a thermostatic expansion valve (TXV) that is held at a constant position during full loading. The last component to be determined for the baseline model was the compressors. The Department chose Copeland

Scroll compressors that had a capacity of 41.2 kBtu/h and an EER of 10.3. These compressors were very similar in capacity to the compressors used in the 7.5-ton R-22 unit, and they fulfilled the 90,000 Btu/h capacity requirement for the unit. The results of the baseline nearly replicated those of the R-22 unit, as the capacity was 90,350 Btu/h (91,040 Btu/h, R-22) and the EER 10.32 (10.33, R-22).

Design Option Procedure

Reaching the 12.0 EER target efficiency for this unit was not as simple as with the R-22 unit, primarily because the baseline EER was 10.1 rather than 11.0. In order to make up the large gap between baseline and target, DOE implemented a three-step process. The first of these steps was enhancing the condenser coil, taking advantage of the lower allowable condensing temperature in the process. From this unit, DOE added an increase in evaporator depth. The final step was an option that was used only for the 410A design options, which involved the use of more efficient fan and blower motors.

The enhancement of the condenser coil involved increasing the coil face area to 20 ft^2, and adding one row in depth. The resulting ΔP_{AIR} was nearly equal to that of the baseline model, since the effects of the area increase and added depth nearly canceled each other out, and the condenser airflow was 6300 cfm (6315 cfm, baseline) and the fan power 610 W (620 W, baseline). The condensing temperature for this modified unit was 116.1˚F, at the threshold of the allowable limit, and the condenser exit temperature was 108.2˚F. The result was a unit with an 11.21 EER, almost 0.9 EER higher than the baseline.

The next model built off the results of the previous step, and involved adding one row to the evaporator coil to make a four-row coil. The effect of this change was a 50 W increase in evaporator blower power, due to the higher ΔP_{AIR}, and increased capacity. The Department subsequently scaled down the compressors by 7 percent from the baseline, to 38.3 kBtu/h, and maintained the capacity. The evaporating temperature increased to 50.8˚F, and the SHR to 0.743 due to the changes. The result was a unit with an 11.76 EER, more than 1.4 EER higher than the baseline.

The final step in this design option process was to introduce electronically commutated motors (ECM) for both the condenser fans and evaporator blower. These motors are more efficient than the ones that are typically used, and provided a gain of 12 percent in condenser fan motor efficiency (82 percent efficiency versus 70 percent) and 5 percent in evaporator blower motor efficiency (90 percent versus 85 percent). This change decreased the power required for these motors, lessened the waste heat that they added to their respective airstreams, and increased unit EER. As this alone was not quite enough to reach the 12.0 EER target, DOE made a small increase to the face area of the evaporator coil. The Department increased the coil to 10.0 ft^2 (8.9 ft^2, baseline), which decreased the blower power to 870 W (910 W, baseline). The condenser fan power was 520 W (620 W, baseline). These changes in turn required another reduction in compressor capacity to maintain system capacity, and DOE scaled down the compressors from the baseline size by 9 percent, to 37.5 kBtu/h. The result was a unit with a

12.24 EER, nearly 2.0 EER higher than the baseline. Figure C.4.1. summarizes the results described above.

Approach	Baseline		Condenser Only 20 ft^2 Area, +1 Row	Cond. & Evap. Prev. step, +1 Row evap	Cond-Evap-ECM Prev. step+ECM+12%Ev.
Product Specs	10.10-EER		+0.89-EER	+1.44-EER	+1.91-EER
Physical Characteristics			**condenser:** +21%area/+1 row **compressor:** 2 x 41.2 kBtu/hr	**condenser:** +21%area/+1 row **evaporator:** +1 row **compressor:** 2 x 41.2 kBtu/hr	**condenser:** +21%area/+1 row **evaporator:** +1 row **condenser fans:** ECM motors **evaporator blower:** **ECM motor** **compressor:** 2 x 41.2 kBtu/hr
Performance Characteristics	**Tc** = 122.8 F **Te** = 48.3 F **Subcool** = 12.9 F **Superheat** = 20 F **Compressor:** 7222 W **Fan:** 620 W **Blower:** 910W SHR = 0.728		**Tc** = 116.1 F **Te** = 48.2 F **Subcool** = 5.8 F **Superheat** = 20 F **Compressor:** 6580 W **Fan:** 610 W -15 CFM **Blower:** 910W SHR = 0.727	**Tc** = 116.6 F **Te** = 50.8 F **Subcool** = 10.6 F **Superheat** = 20 F **Compressor:** 6149 W **Fan:** 610 W/ -15 CFM **Blower:** 960W SHR = 0.743	**Tc** = 116.3 F **Te** = 51.6 F **Subcool** = 10.2 F **Superheat** = 20 F **Compressor:** 5989 W **Fan:** 520 W/ -15 CFM **Blower:** 870W SHR = 0.748

Figure C.4.1 7.5-ton R-410a Baseline and Design Options

C.4.2 15-ton R-410A Unit

As with the 7.5-ton R-410A unit, DOE based this model on typical unit details for R-22 units of the same capacity. The Department chose the compressors from Copeland's line of R-410A compressors and, in almost every case, used two equal-size compressors in the unit to attain the 180,000 Btu/h capacity. The Department modeled the baseline unit to match the performance of the 15-ton standard-efficiency unit, and employed design options until the unit neared the 12.0 EER level.

15-ton R-410A Baseline Model

The evaporator coil was 17.5 ft^2 and 4 rows deep, with an 18" evaporator blower. The condenser coil was 28 ft^2 and 3 rows deep, with two 26" condenser fans powered by 0.50 hp motors. Refrigerant-line sizes were identical to those of the R-22 unit, and the flow was controlled with a short tube orifice, which simulated a TXV held at a constant position during full loading. The compressors chosen have a capacity of 57.5 kBtu/h and an EER of 10.6, and DOE used three of these to fulfill the 180,000 Btu/h capacity requirement for the unit.

The results of the baseline nearly replicated those of the R-22 unit, although the capacity more closely mirrored that of the high efficiency R-22 unit. The final baseline values were 182,640 Btu/h (174,090 Btu/h, R-22) and the EER 9.72 (9.70, R-22).

Design Option Procedure

Similar to the 7.5-ton unit, the design options for this unit had to cover a large jump in EER, since DOE modeled the baseline to coincide with a standard efficiency level. However, rather than using a three-step process as it had for the 7.5-ton unit, DOE achieved the 12.0 EER with a major change to only the condenser coil. This was possible since the baseline model had a very high condensing temperature, 126.3˚F, which allowed for significant condenser change before the minimum condensing temperature of 116.0˚F was reached.

The Department increased the condenser face area to 44 ft^2 (28ft^2, baseline), and added one row to make it four rows deep. These changes resulted in a slightly lower ΔP_{AIR}, which yielded a condenser airflow of 13600 cfm (12800 cfm, baseline) and a condenser fan power of 1545 W (1660 W, baseline). The enhanced condenser allowed for a reduction in compressor capacity of 11 percent, to 153.0 kBtu/h, and came very close to the temperature limits: condensing temperature was 116.1˚F, and condenser exit temperature was 102.2˚F. The result was a unit with a 12.12 EER, 2.4 EER higher than the baseline. Figure C.4.2. shows the results described above.

Approach	Baseline		Condenser Only 27% Area, +1 Row	Condenser Max 57% Area, 4 row	Max + ECM 57% Area, 4 row
Product Specs	9.71-EER		+1.91 -EER	+2.41 -EER	+2.67 -EER
Physical Characteristics			**box:** **+47% volume** **condenser:** +27% area/+1 row **compressor:** 1 x 103.0 kBtu/hr 1 x 55.0 kBtu/hr	**box:** **+47% volume** **condenser:** +57% area, +1 row **compressor:** 3 x 50.5 kBtu/hr	**box:** **+47% volume** **condenser:** +57% area, +1 row **condenser fans:** ECM motors **evaporator blower:** ECM motor **compressor:** 3 x 50.5 kBtu/hr
Performance Characteristics	**Tc** = 126.3 F **Te** = 51.0 F **Subcool** = 18.2 F **Superheat** = 10 F **Compressor:** 15890 W **Fan:** 1660 W **Blower:** 1240W SHR = 0.721		**Tc** = 118.2 F **Te** = 51.1 F **Subcool** = 14.1 F **Superheat** = 10 F **Compressor:** 12660 W **Fan:** 1600 W +2,500CFM **Blower: 1240**W SHR = 0.724	**Tc** = 116.1 F **Te** = 51.1 F **Subcool** = 11.7 F **Superheat** = 10 F **Compressor:** 12098 W **Fan:** 1545 W +2,900CFM **Blower: 1240**W SHR = 0.724	**Tc** = 116.1 F **Te** = 51.1 F **Subcool** = 11.7 F **Superheat** = 10 F **Compressor:** 12098 W **Fan:** 1320 W+2,900 CFM **Blower: 1170**W SHR = 0.724

Figure C.4.2   15-ton R-410a Baseline and Design Options