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BUILDING TECHNOLOGIES PROGRAM

Strategy Guideline: HVAC Equipment Sizing

Arlan Burdick
IBACOS, Inc.

February 2012

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HVAC Equipment Sizing

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** Unless otherwise noted, all figures were created by IBACOS.*

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Definitions

ACCA	Air Conditioning Contractors of America
AFUE	Annual Fuel Utilization Efficiency
AHRI	Air-Conditioning, Heating, and Refrigeration Institute
ARI	Air-Conditioning and Refrigeration Institute
ASHRAE	American Society of Heating, Refrigerating, and Air Conditioning Engineers
BEopt	Building Energy Optimization (software)
cfm	Cubic Feet per Minute
EDB	Entering Dry-Bulb
EWB	Entering Wet-Bulb
HSPF	Heating Seasonal Performance Factor
HVAC	Heating, Ventilation, and Air Conditioning
IECC	International Energy Conservation Code
LAT	Temperature of Air Leaving the Cooling Coil
OAT	Outdoor Air Temperature
SEER	Seasonal Energy Efficiency Ratio
SHR	Sensible Heat Ratio
TD	Room-Air to Supply-Air Temperature Difference

Executive Summary

Right-sizing of a residential heating, ventilation, and air conditioning (HVAC) system involves the selection of equipment and the design of the air distribution system to meet the accurate predicted heating and cooling loads of a house. This Strategy Guideline follows the Air Conditioning Contractors of America (ACCA) *Manual S—Residential Equipment Selection* (Manual S) (Rutkowski 1995) to describe what information is needed and how to use that information to initially select the equipment for a properly designed HVAC system. Equipment selection and duct design are iterative processes where the capacity of the equipment is balanced against the design of the distribution system.

This Strategy Guide describes the equipment selection of a split system air conditioner and furnace for an example house in Chicago, Illinois, as well as a heat pump system for an example house in Orlando, Florida. The required heating and cooling load information for the two example houses was developed by Burdick (2011).

Selection of the equipment can have a substantial impact on the efficiency and operating costs of the system. For the example houses presented in this Strategy Guide, a 92.5% AFUE (annual fuel utilization efficiency) furnace with a 16 SEER (seasonal energy efficiency ratio) air conditioner was the system chosen for the Chicago House and a 9.2 HSPF (heating seasonal performance factor)/16 SEER heat pump for the Orlando House.

With an accurate estimate of the heating and cooling loads completed, the HVAC system designer can begin the equipment selection procedure. It is the system designer's responsibility to select the heating and cooling equipment and to verify that the selected pieces of equipment meet the following:

- The total cooling load, composed of both the sensible load (temperature) and the latent load (humidity)
- The heating load
- The blower capacity (in cubic feet per minute [cfm]) to meet the volume of air cfm range needed.

The general steps to equipment selection are:

1. Determine accurate heating and cooling loads.
2. Select cooling equipment.
3. Select heating equipment.
4. Design the ducts.

It is worth noting that the design of a space conditioning system is an iterative process. Once a duct system configuration and preliminary design are completed, different equipment may need to be selected to achieve the proper airflow to satisfy the room-by-room heating and cooling loads.

1 Introduction

The heating, ventilation, and air conditioning (HVAC) system is arguably the most complex system installed in a house and is a substantial component of the total house energy use. A right-sized HVAC system will provide the desired occupant comfort and will run efficiently. This Strategy Guideline discusses the information needed to initially select the equipment for a properly designed HVAC system. Right-sizing of an HVAC system involves the selection of equipment and the design of the air distribution system to meet the accurate predicted heating and cooling loads of the house. Right-sizing the HVAC system begins with an accurate understanding of the heating and cooling loads on a space; however, a full HVAC design involves more than just the load estimate calculation—the load calculation is the first step of the iterative HVAC design procedure.

The iterative nature of HVAC design is illustrated in Figure 1, along with the Strategy Guidelines that cover the various aspects of the design process. The heating and cooling loads depend on the building location and construction, whereas the equipment selection and distribution design depend on the loads and each other. After initial equipment selection, several adjustments to duct design and equipment selection may be needed to optimize the design.

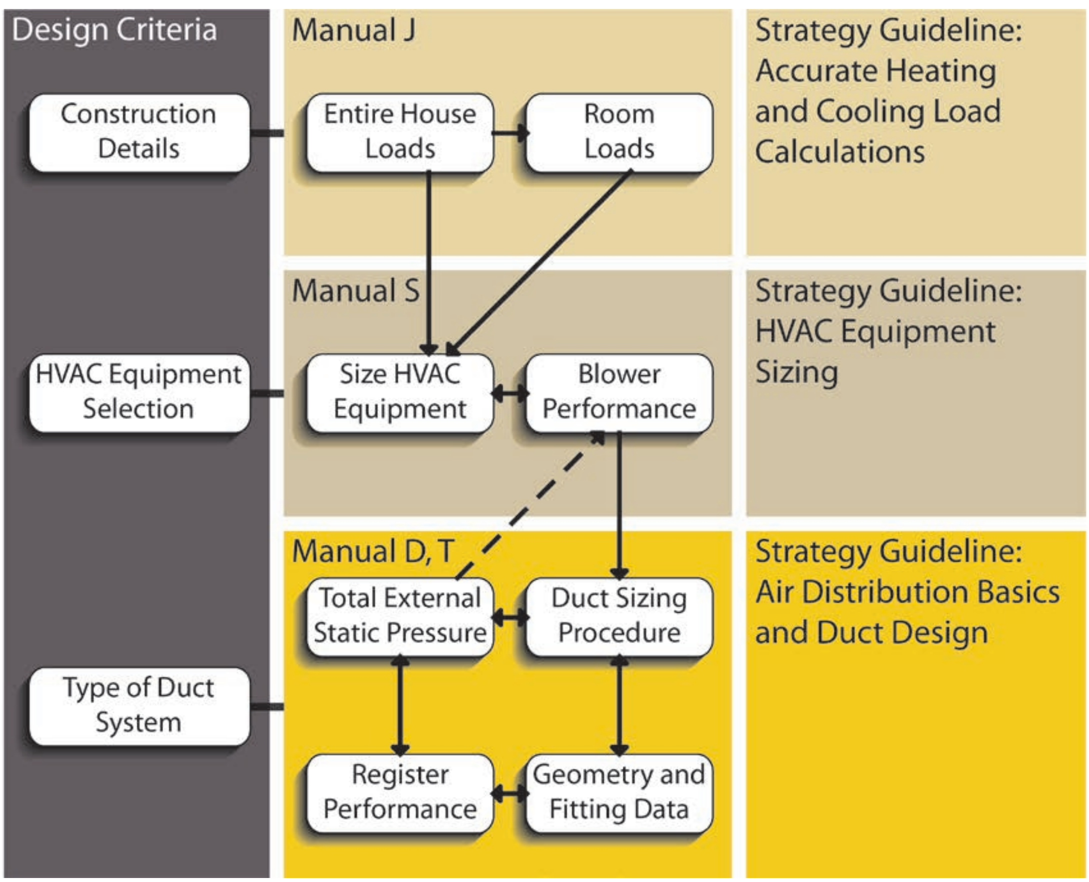


Figure 1. Design information from ACCA Manual J—Residential Load Calculation.

The procedures of residential HVAC design are covered in detail by a series of publications produced by the Air Conditioning Contractors of America (ACCA), which in turn references information provided by the American Society of Heating, Refrigerating and Air-Conditioning Engineers (ASHRAE). The heat loss and gain values are estimated using the procedures from the ACCA *Manual J—Residential Load Calculation* (Manual J) (Rutkowski 2006). Manual J applies only to single-family detached dwellings, low-rise condominiums, and townhouses. Mechanical equipment selection is done with the aid of the ACCA *Manual S—Residential Equipment Selection* (Manual S) (Rutkowski 1995). The ductwork to convey the proper amount of conditioned air to meet the load requirements of the space is designed with the aid of the ACCA *Manual D—Residential Duct System Procedure* (Manual D) (Rutkowski 2009a). Conditioned air delivery to the space is controlled by the type and size of the air outlet. ACCA *Manual T—Air Distribution Basics for Residential and Small Commercial Buildings* (Manual T) (Rutkowski 2009b) provides the guidance on selecting the air outlet size and type. The ACCA procedures have been written into commercial software packages to help the designer work through the iterations required for a good design. Although commercial software is an important tool for design, it should be operated with a solid understanding of the procedures of good design.

2 Overview

For the purposes of this Strategy Guide, an energy-efficient house is defined as one that is designed and built for decreased energy use and improved comfort through higher levels of insulation, more energy-efficient windows, high-efficiency space conditioning and water heating equipment, energy-efficient lighting and appliances, reduced air infiltration, and controlled mechanical ventilation. Specification levels for energy-efficient houses have historically been prescribed by beyond code programs that set a percentage better than code for energy use. For example, ENERGY STAR® requires houses to be 15% more energy efficient than code. Beyond code programs continue to set a percentage better than the improved codes for energy use, raising the bar for whole-house energy efficiency. The 2009 International Energy Conservation Code® (IECC 2009) establishes an estimated 15% improvement in energy efficiency over the previous 2006 IECC requirements. Ongoing code cycles incrementally increase the minimum efficiency of a house. For example, the 2012 IECC achieves approximately 30% savings over the 2006 version. As the new codes are adopted and implemented, a house that was built under a beyond code program in 2010 will likely be the code mandated house in 2015. As the energy efficiency of the house is increased under code or beyond code programs, the peak heating and cooling loads are significantly reduced.

To illustrate the procedures of HVAC equipment selection, two theoretical houses that meet the 2009 IECC prescriptive path were modeled. The first (the "Chicago House") is a one-story, 2,223-ft^2 (above-grade) house that has a full conditioned basement and is located in Chicago, Illinois (IECC Climate Zone 5 - CZ5). The other (the "Orlando House") is a one-story 2,223-ft^2 slab-on-grade house in Orlando, Florida (IECC Climate Zone 2 - CZ2).

In addition to the climatic differences between the two house locations, the Chicago House is a house over a full basement, whereas the Orlando House is a slab-on-grade foundation, allowing for the discussion of different mechanical and ductwork layouts.

Figure 2 shows the Chicago House example split system with an upflow furnace and evaporator coil located in the basement and an outdoor condenser coil. Figure 3 shows the Orlando House example air source heat pump with outdoor coil and horizontal fan coil in the conditioned attic.

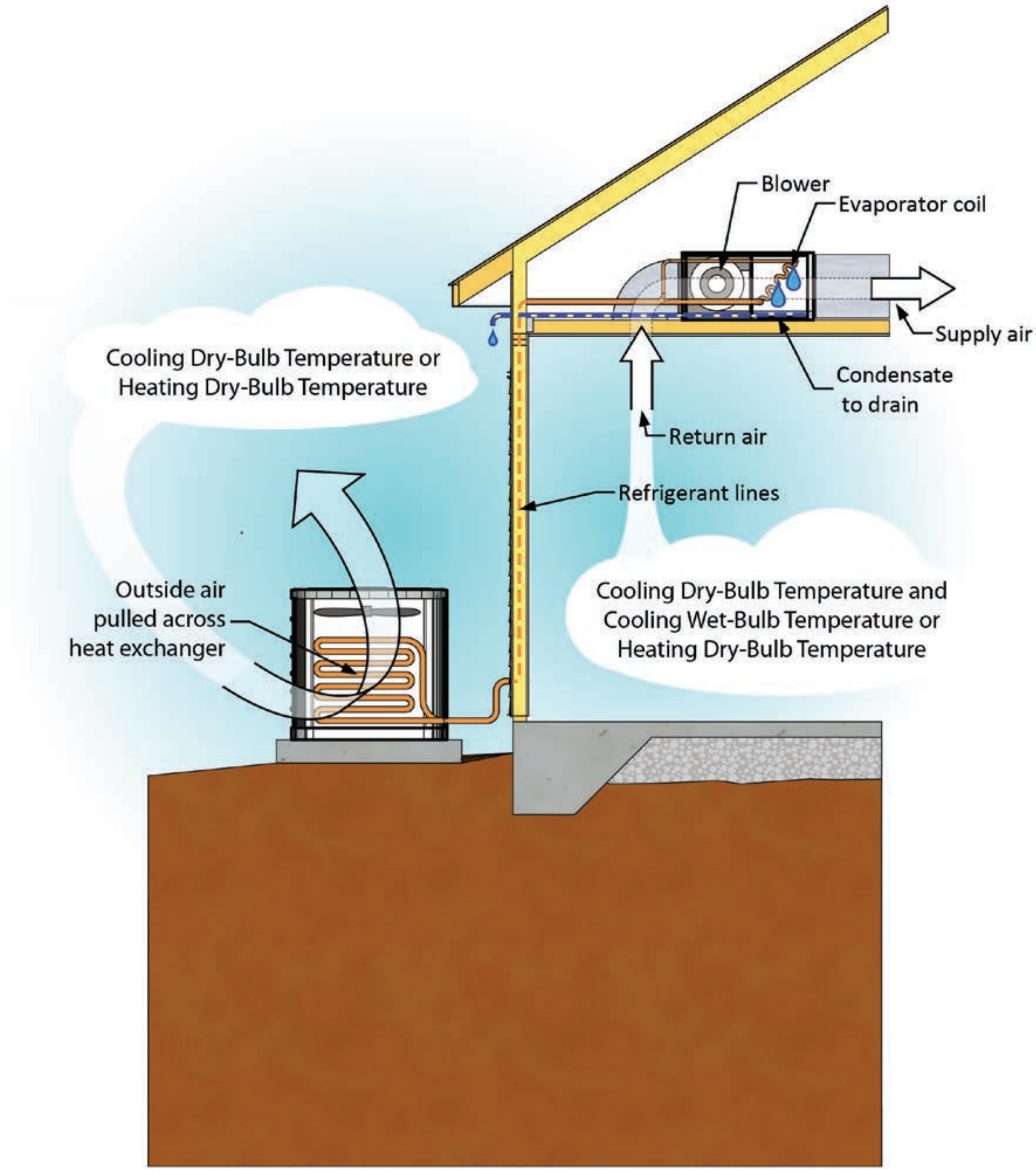


Figure 2. Upflow furnace scenario as in the Chicago House.

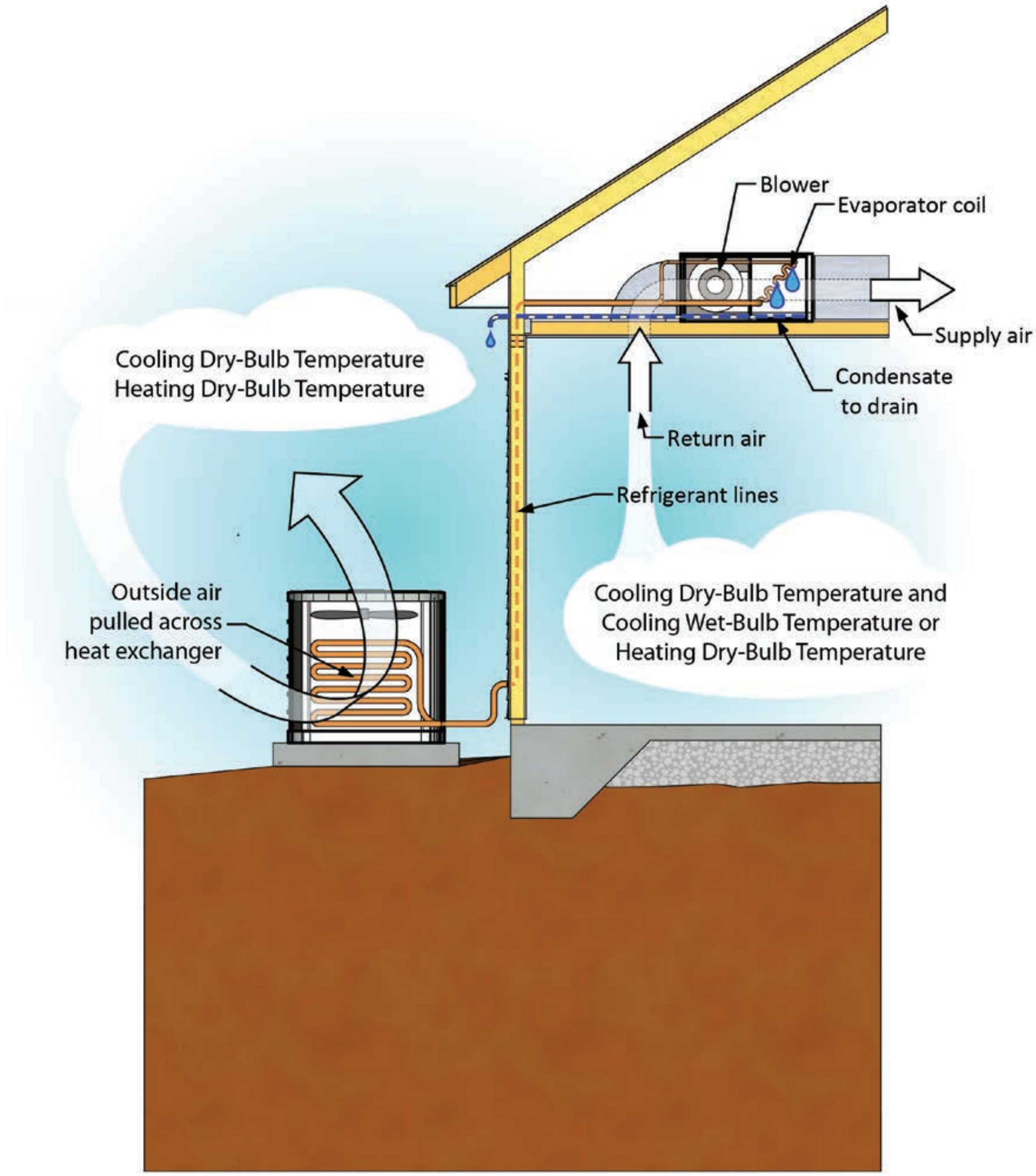


Figure 3. Heat pump scenario as in the Orlando House.

2.1 Design Conditions

The impacts and importance of accurate calculation of heating and cooling loads are covered in depth by Burdick (2011). The equipment selection discussion in this Strategy Guide builds upon the information developed in that heating and cooling loads guide. The Manual J loads from Burdick (2011) are listed in Table 1.

Table 1. ACCA Manual J Heating and Cooling Loads

Chicago House		Orlando House	
Heating Load	41,660 Btu/h	Heating Load	23,640 Btu/h
Sensible Cooling	17,370 Btu/h	Sensible Cooling	16,630 Btu/h
Latent Cooling	3,240 Btu/h	Latent Cooling	4,060 Btu/h
Total Cooling	20,610 Btu/h	Total Cooling	20,690 Btu/h

The general steps to equipment selection are shown in Figure 4, and the design information needed to select equipment is outlined in Figure 5.

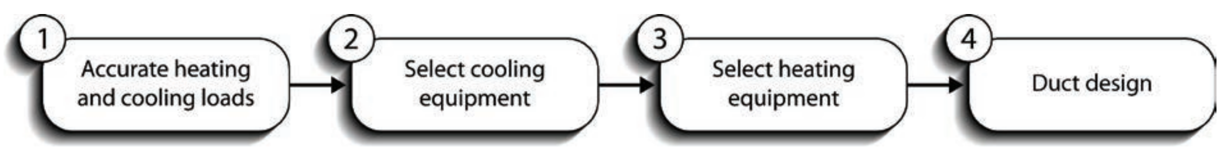


Figure 4. General steps to equipment selection.

Note that the design of a space conditioning system is an iterative process, and once a duct system configuration and preliminary design are completed, different equipment may need to be selected to achieve the proper airflow to satisfy the room-by-room heating and cooling loads.

Manual J	Heating Load	
	Sensible Cooling Load	
	Latent Cooling Load	
Outdoor Conditions	Cooling Dry-bulb Temperature	
	Heating Dry-bulb Temperature	
Indoor Conditions	Cooling Dry-bulb Temperature	
	Cooling Wet-bulb Temperature	
	Heating Dry-bulb Temperature	
Airflow Estimates	Cubic Feet per Minute	

Figure 5. Design information needed.

The design loads along with the indoor and outdoor conditions information needed for equipment selection are all taken from the inputs used in the Manual J procedure. Faithful adherence to the conditions used in Manual J is required for accurate equipment selection.

2.2 Preliminary Equipment Selection

With an accurate estimate of the heating and cooling loads completed, the designer can begin the equipment selection procedure. It is the system designer's responsibility to select the heating and cooling equipment and to verify that the selected pieces of equipment meet the following:

- The total cooling load, composed of both the sensible load (temperature) and the latent load (humidity)
- The heating load
- The blower capacity (in cfm) to meet the volume of air cfm range needed.

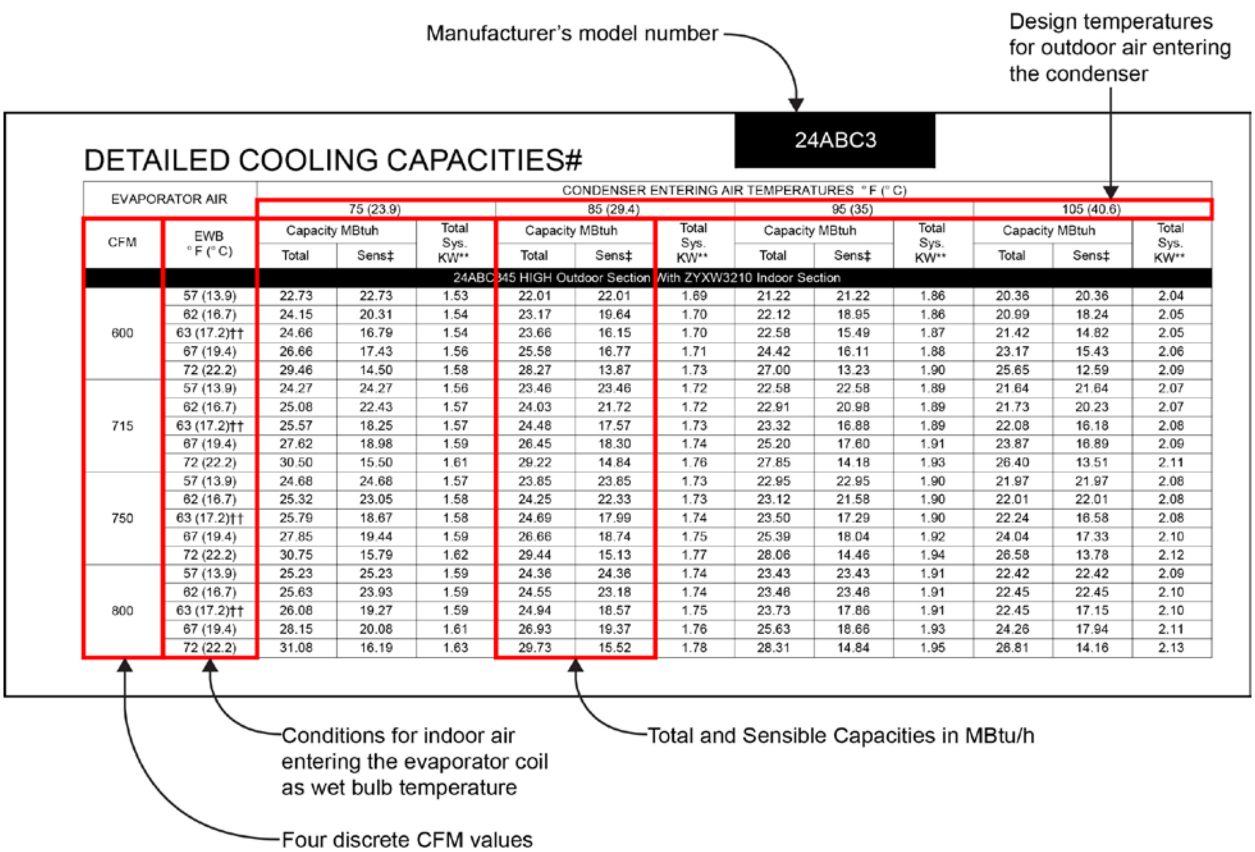
In most areas of the lower 48 United States, there will be both a cooling load and a heating load, where the sizing of the cooling equipment will dominate the equipment selection process. A system designed to satisfy only the heating load will not have enough blower power if cooling is added after installation, because of the greater pressure loss across the cooling coil. The blower must be capable of supplying sufficient cfm airflow across the cooling coil.

Selecting equipment based on the cooling load may lead to a furnace that is oversized for the heating load. Manual S recommends that the furnace size remains within 140% of the peak heating load estimate and that cooling equipment capacity not exceed the total load by more than 15%. Manual S recommends that a heat pump in a cooling dominated climate not exceed the load by more than 15% in either the cooling or the heating mode; however, in a heating dominated climate, a heat pump may exceed the heating load by as much as 25%.

It should be recognized that in energy-efficient houses, the loads may be so small that the standard manufacturer's range of capacities exceeds these percentages. For example, if a house has a 19,000 Btu/h cooling load and manufacturer's equipment is available in only 17,500 Btu/h and 23,400 Btu/h, the larger unit is more than 15% oversized. However, the larger unit should be the equipment selected to ensure the peak load will be met. Designers may want to consider multispeed equipment in these cases, where the system runs more frequently at a lower capacity to more closely meet the load for more times of the year.

2.3 Manufacturer Data Tables

HVAC equipment manufacturers make available tabulated performance data that are used to select the equipment based on the loads and design conditions. A fictitious example of a detailed cooling capacities table is shown in Figure 6.



DETAILED COOLING CAPACITIES#

EVAPORATOR AIR		CONDENSER ENTERING AIR TEMPERATURES °F (°C)											
		75 (23.9)			85 (29.4)			95 (35)			105 (40.6)		
CFM	EWB °F (°C)	Capacity MBtuh Total	Capacity MBtuh Sens‡	Total Sys. KW**	Capacity MBtuh Total	Capacity MBtuh Sens‡	Total Sys. KW**	Capacity MBtuh Total	Capacity MBtuh Sens‡	Total Sys. KW**	Capacity MBtuh Total	Capacity MBtuh Sens‡	Total Sys. KW**
		24ABC345 HIGH Outdoor Section With ZYXW3210 Indoor Section											
600	57 (13.9)	22.73	22.73	1.53	22.01	22.01	1.69	21.22	21.22	1.86	20.36	20.36	2.04
	62 (16.7)	24.15	20.31	1.54	23.17	19.64	1.70	22.12	18.95	1.86	20.99	18.24	2.05
	63 (17.2)††	24.66	16.79	1.54	23.66	16.15	1.70	22.58	15.49	1.87	21.42	14.82	2.05
	67 (19.4)	26.66	17.43	1.56	25.58	16.77	1.71	24.42	16.11	1.88	23.17	15.43	2.06
	72 (22.2)	29.46	14.50	1.58	28.27	13.87	1.73	27.00	13.23	1.90	25.65	12.59	2.09
715	57 (13.9)	24.27	24.27	1.56	23.46	23.46	1.72	22.58	22.58	1.89	21.64	21.64	2.07
	62 (16.7)	25.08	22.43	1.57	24.03	21.72	1.72	22.91	20.98	1.89	21.73	20.23	2.07
	63 (17.2)††	25.57	18.25	1.57	24.48	17.57	1.73	23.32	16.88	1.89	22.08	16.18	2.08
	67 (19.4)	27.62	18.98	1.59	26.45	18.30	1.74	25.20	17.60	1.91	23.87	16.89	2.09
	72 (22.2)	30.50	15.50	1.61	29.22	14.84	1.76	27.85	14.18	1.93	26.40	13.51	2.11
750	57 (13.9)	24.68	24.68	1.57	23.85	23.85	1.73	22.95	22.95	1.90	21.97	21.97	2.08
	62 (16.7)	25.32	23.05	1.58	24.25	22.33	1.73	23.12	21.58	1.90	22.01	22.01	2.08
	63 (17.2)††	25.79	18.67	1.58	24.69	17.99	1.74	23.50	17.29	1.90	22.24	16.58	2.08
	67 (19.4)	27.85	19.44	1.59	26.66	18.74	1.75	25.39	18.04	1.92	24.04	17.33	2.10
	72 (22.2)	30.75	15.79	1.62	29.44	15.13	1.77	28.06	14.46	1.94	26.58	13.78	2.12
800	57 (13.9)	25.23	25.23	1.59	24.36	24.36	1.74	23.43	23.43	1.91	22.42	22.42	2.09
	62 (16.7)	25.63	23.93	1.59	24.55	23.18	1.74	23.46	23.46	1.91	22.45	22.45	2.10
	63 (17.2)††	26.08	19.27	1.59	24.94	18.57	1.75	23.73	17.86	1.91	22.45	17.15	2.10
	67 (19.4)	28.15	20.08	1.61	26.93	19.37	1.76	25.63	18.66	1.93	24.26	17.94	2.11
	72 (22.2)	31.08	16.19	1.63	29.73	15.52	1.78	28.31	14.84	1.95	26.81	14.16	2.13

Figure 6. Fictitious example: detailed cooling capacities table.

It is imperative that actual manufacturer's performance data are used to select equipment rather than data from the Air-Conditioning, Heating, and Refrigeration Institute (AHRI), formerly known as Air Conditioning and Refrigeration Institute (ARI) data. AHRI certification data are tested at one set of conditions to offer an "apples-to-apples" comparison of equipment performance; this is not the same set of conditions as the Manual J indoor design conditions. Also, AHRI certification data list only total capacity and not sensible or latent capacity. Therefore, AHRI does not provide enough information to accurately select a piece of equipment. Details about the limitations of AHRI data are provided in Section 8 of Manual S.

Each manufacturer may tabulate the equipment performance data differently. Taking time to understand how the equipment data are formatted is important to making the correct selection. In the example detailed cooling capacities table shown in Figure 6, the manufacturer lists the total and sensible capacities of the equipment in MBtu/h. Design temperatures for outdoor air entering the condenser are listed in a 10-degree spread, requiring interpolation to find performance at a design temperature between the listed temperatures. Interpolation of data is acceptable to determine the performance standards of specific design temperatures; extrapolation of non-linear data is not an acceptable practice. The conditions for indoor air entering the evaporator coil are listed as the entering wet-bulb (EWB) temperature.

The outside condenser and indoor coil package are selected from the manufacturer's expanded product performance data tables. The outside condenser and indoor coil package will determine the system size and operating airflows. The initial cfm airflow estimate and operating cfm will likely be different values.

2.4 Cooling Equipment Capacity Sensitivities

Capacities of cooling equipment are conditional based on the following operating conditions:

- Cubic feet per minute airflow across the coil (Figure 7)
- Outdoor air temperature (OAT) at the condenser (Figure 8)
- Indoor wet-bulb temperature at the coil (Figure 9)
- Indoor dry-bulb temperature at the coil (Figure 10).

Figures 7 through 10 show the conditional nature of direct expansion cooling equipment and that the available latent capacity is a direct relationship to the total and sensible capacities.

It is helpful to consider these relationships while searching the manufacturer's detailed cooling capacity data. This conditional nature among total, sensible, and latent capacities causes an imbalance between latent load and latent capacity, where latent capacity slightly exceeds the latent load. This can be somewhat self-correcting. Manual S, Section 3-6, explains that with a latent capacity that slightly exceeds the latent load, about half the excess latent capacity will be converted to sensible capacity, aiding in equipment selection.

Figure 7 shows that increasing the airflow across the coil (cfm) will increase the total and sensible capacities of the equipment. However, because the sensible capacity increases at a higher rate than the total capacity, the latent capacity available is reduced as the cfm increases. This relationship between cfm and latent capacity of the equipment is an important consideration when selecting equipment to meet the latent load.

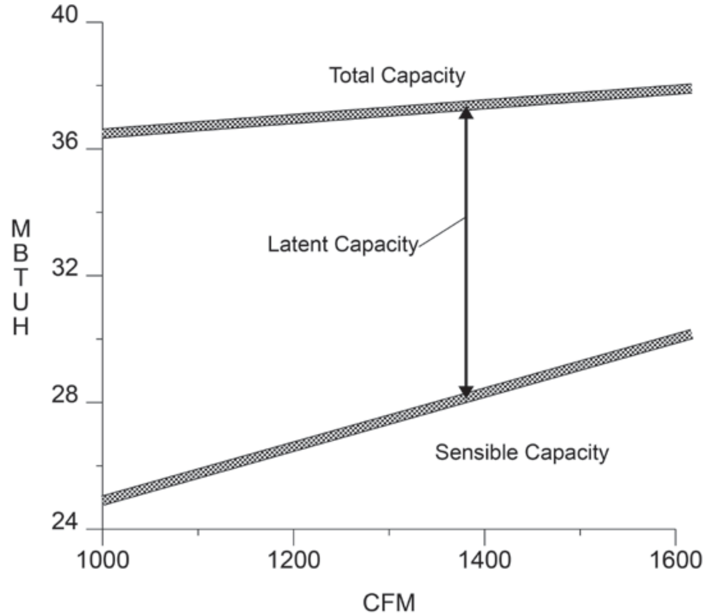


Figure 7. Airflow sensitivity graph (from Manual S, Figure 3-1).
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Figure 8 shows that as the OAT at the condenser increases, the total sensible and latent capacities decrease. Because the total capacity decreases at a faster rate than the sensible capacity, the available latent capacity of the equipment is lower. Because of this effect of OAT on equipment capacity, it is critical that the same temperature used for load calculations is used for equipment selection.

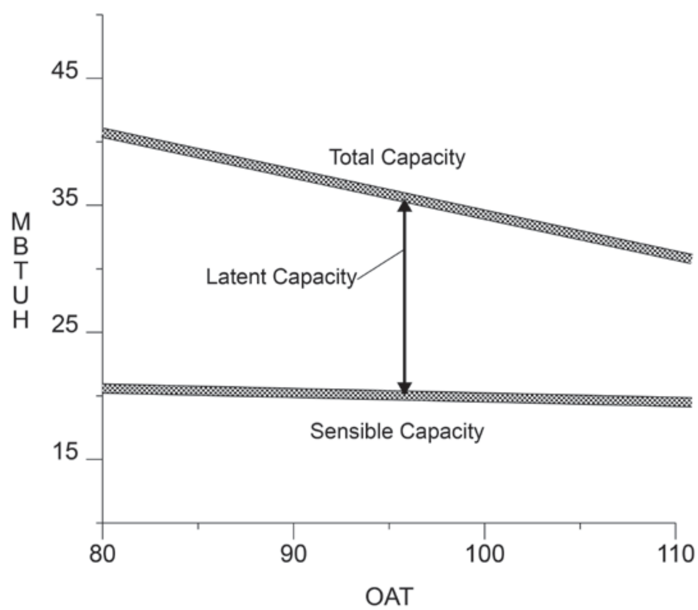

Figure 8. OAT sensitivity graph (from Manual S, Figure 3-1).
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As shown in Figure 9, higher EWB temperatures at the indoor coil will result in a higher total capacity but a lower sensible capacity. This will result in a greater available latent capacity as the difference between the total and sensible capacities increases.

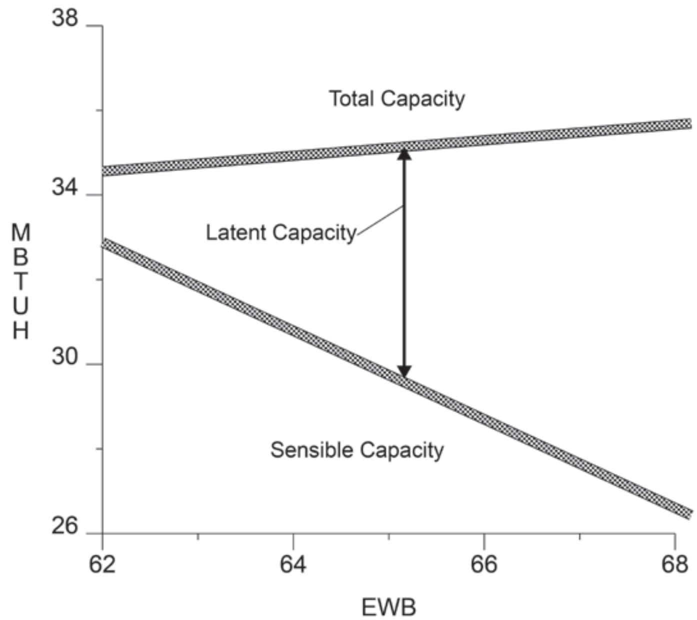

Figure 9. Indoor EWB sensitivity graph (from Manual S, Figure 3-1).
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Figure 10 shows that a higher entering dry-bulb (EDB) temperature at the indoor coil will have relatively little effect on the total capacity but will increase the sensible capacity of the equipment, resulting in a lower available latent capacity.

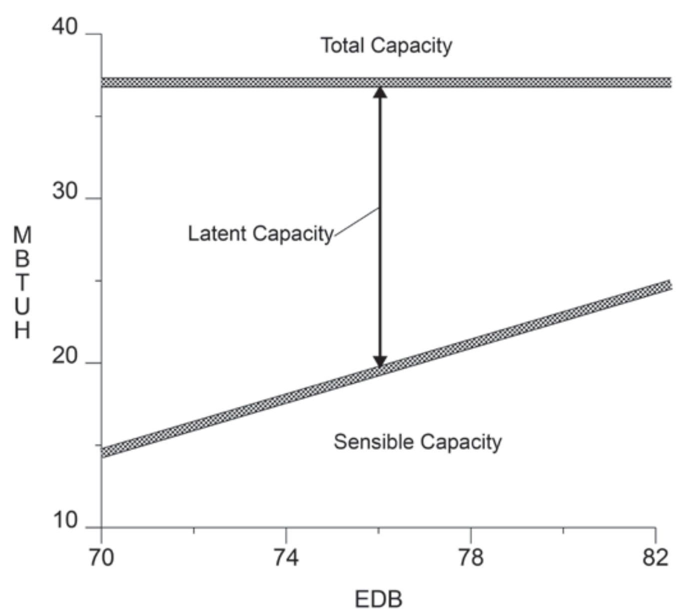

Figure 10. Indoor EDB temperature sensitivity graph (from Manual S, Figure 3-1).
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2.5 Estimated Target Cubic Feet per Minute

To search the manufacturer's expanded performance data, an initial estimate of the required cfm airflow is needed. The estimated cfm airflow is used only to narrow down the equipment choices from the manufacturer's data and will likely be higher than the actual cfm used for the design.

The detailed procedure to estimate a target cfm is described in Manual S. Section 1.3 of Manual S instructs the designer to use the sensible heat ratio (SHR) value to determine the desired room-air to supply-air temperature difference (TD) in degrees Fahrenheit. The SHR is the ratio of the sensible load determined from the Manual J procedures to the Manual J total load shown in Equation 1.

Sensible Heat Ratio (SHR) – The sensible load divided by the total load.

Temperature Difference (TD) – The difference between the room air temperature and the supply air temperature.

$$\text{SHR} = \frac{\text{Sensible Load}}{\text{Total Load}} \tag{1}$$

Figure 11, taken from Table 1-4 in Manual S, shows the relationship of an SHR range to TD value that is compatible with the sensible and latent loads. The column "LAT" is the temperature of air leaving the coil, and the column "Room db" is the indoor conditions dry-bulb temperature used in the Manual J calculations. The difference between the two is the TD value.

Sensible Heat Ratio Versus TD Value			
SHR	**LAT**	**Room db**	**TD**
Below 0.80	54	75	21
0.80 to 0.85	56	75	19
Above 0.85	58	75	17

Figure 11. TD value table (from Manual S, Table 1-4).
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Determining the SHR early is helpful because the SHR can also be used as a double check on the accuracy of the Manual J loads. A relatively high SHR (e.g., above 0.85) indicates a relatively small latent load compared to the sensible load. Conversely, a relatively low SHR (e.g., below 0.75) indicates the latent load is relatively large compared to the sensible load.

Once the TD value is known, the estimated cfm target can be calculated using Equation 2.

$$\text{cfm Target} = \frac{\text{Manual J Sensible Load}}{1.1 \times \text{TD}} \tag{2}$$

The number 1.1 in the denominator of Equation 2 represents an air properties constant associated with elevations lower than 1,500 feet and is appropriate for both the Chicago House and the

Orlando House examples. The value of the air properties constant will decrease with higher elevations. Correction factor information can be found in Appendix 6 of Manual S.

2.6 Efficiency Considerations

The selection of the efficiency rating (AFUE or SEER) of the equipment will affect the long-term energy consumption and system operating costs. Although the decision-making process for selecting an efficiency level is a cost/benefit analysis of installed first cost versus the operational savings of the more efficient equipment, it is anticipated that future legislation will tighten the efficiency requirements of HVAC equipment, making more efficient equipment mandatory. To illustrate the impact of equipment efficiency on operating energy use and costs, two performance levels were compared using Building Energy Optimization software, version 1.1 (BEopt 1.1).

For the Chicago House example, an 80% AFUE furnace with a 13 SEER air conditioner was compared to a 92.5% AFUE furnace with a 16 SEER air conditioner. The results show a total annual source energy use reduction of 37.7 MBtu/yr or 11%. The bulk of the reduction, 31.6 MBtu/yr, was in the heating mode with 1.7 MBtu/yr in cooling and the remaining 4.4 MBtu/yr reduction coming from fan energy savings. Figure 12 shows that, on an annualized utility bill basis (dollars/year), the more efficient equipment in the Chicago House resulted in an estimated $273 reduction.

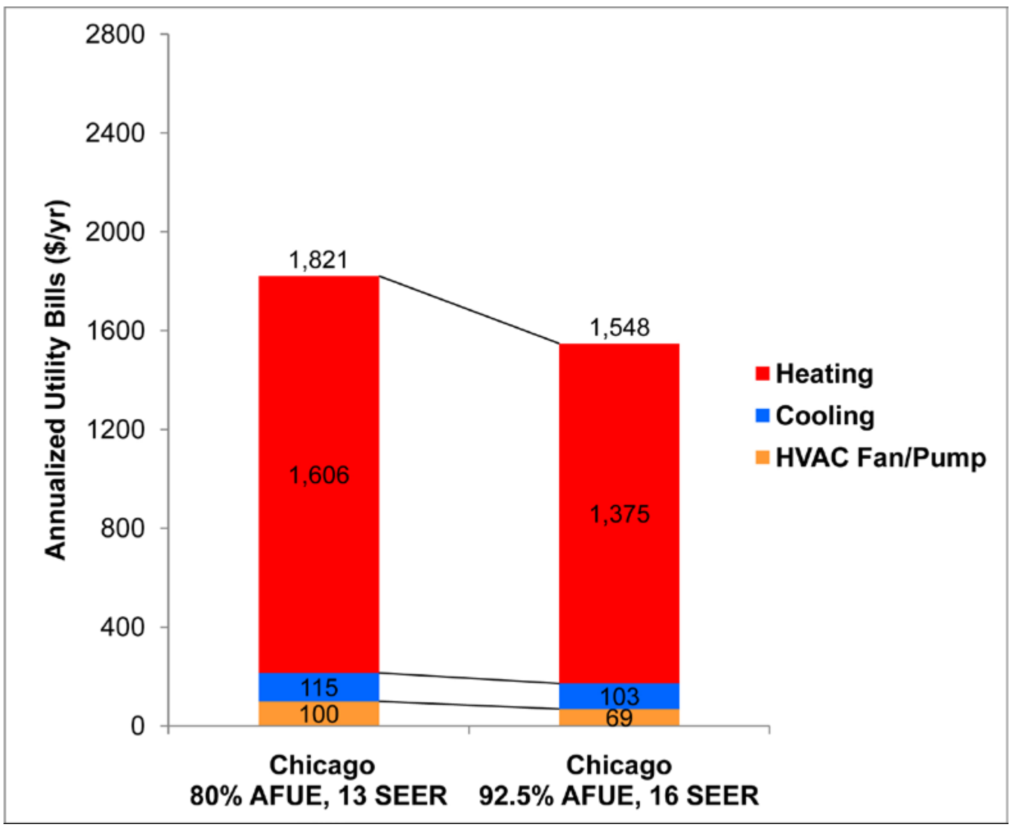

Figure 12. Chicago House BEopt 1.1 output.

For the Orlando House, an 8.0 HSPF/13 SEER heat pump was compared with a 9.2 HSPF/16 SEER heat pump. The results show a total annual source energy use reduction of 9.3 MBtu/yr or 13%. The bulk of the reduction, 7.5 MBtu/yr, was in the cooling mode with 0.7 MBtu/yr in heating and the remaining 1.1 MBtu/yr reduction coming from fan energy savings. Figure 13 shows that, on an annualized utility bill basis (dollars/year), the more efficient equipment in the Orlando House resulted in an estimated $65 reduction.

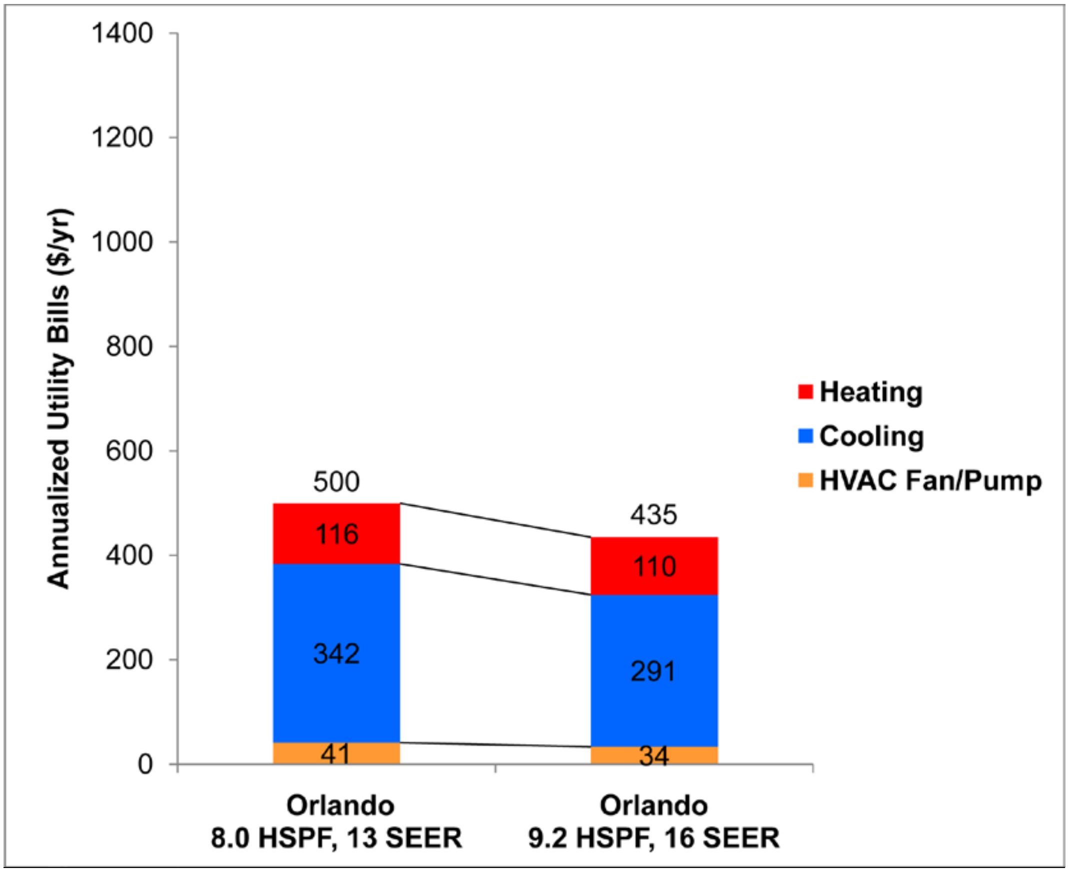

Figure 13. Orlando House BEopt 1.1 output.

For the example houses in this Strategy Guide, a 92.5% AFUE furnace with a 16 SEER air conditioner was the system chosen for the Chicago House and a 9.2 HSPF/16 SEER heat pump for the Orlando House.

3 Chicago House Example

The following is an example selection process of a split system for the Chicago House with an upflow gas furnace located in the basement and an air conditioner. The Chicago House has both a cooling load and a heating load; therefore, the cooling equipment will be selected first.

3.1 Cooling Equipment Selection

Split system cooling equipment consisting of an outdoor condensing unit and an indoor evaporator coil needs the two pieces to be matched for efficiency and performance. The example here shows a manufacturer matched equipment package. Although it is possible to select the two pieces separately, it is typically not necessary. The manufacturer matched package greatly simplifies the cooling equipment selection.

The first step highlighted in Figure 14 instructs the designer to define the cooling design parameters from the cooling loads of the Manual J procedure. The cooling design information developed by Burdick (2011) from the Manual J procedures is shown in Figure 15.

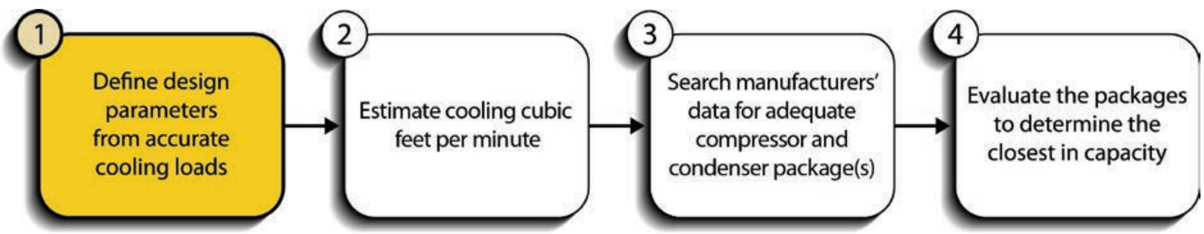

Figure 14. Cooling equipment selection step 1.

Manual J	Heating Load	41,660 Btu/h
	Sensible Cooling Load	17,370 Btu/h
	Latent Cooling Load	3,240 Btu/h
Outdoor Conditions	Cooling Dry-bulb Temperature	89 °F
	Heating Dry-bulb Temperature	2 °F
Indoor Conditions	Cooling Dry-bulb Temperature	75 °F
	Cooling Wet-bulb Temperature	63 °F
	Heating Dry-bulb Temperature	70 °F
Airflow Estimates	Cubic Feet per Minute	831 cfm

Figure 15. Chicago House design information.

Step 2, highlighted in Figure 16, instructs the designer to estimate the cooling cfm. Utilizing the methods outlined in Section 1.3 of Manual S, the initial cfm airflow is estimated.

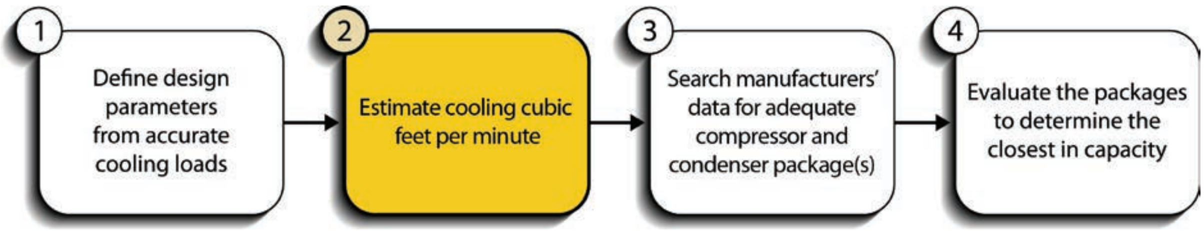

Figure 16. Cooling equipment selection step 2.

Equation 3 applies the cooling design parameters from Figure 15 to Equation 1, resulting in an SHR of 0.84.

$$\text{SHR} = \frac{\text{Sensible Load}}{\text{Total Load}} = \frac{17.37\,\dfrac{\text{MBtu}}{\text{h}}}{20.61\,\dfrac{\text{MBtu}}{\text{h}}} = 0.84 \qquad (3)$$

From the TD value table (Figure 11), the TD value for SHR = 0.84 is 19. These values can now be applied to the sensible heat equation (Equation 2) as shown in Equation 4.

$$\text{cfm Target} = \frac{\text{Manual J Sensible Load}}{1.1 \times \text{TD}} = \frac{17{,}370}{1.1 \times 19} = 831 \qquad (4)$$

Step 3, highlighted in Figure 17, is to search the manufacturer's data for an adequate cooling equipment package.

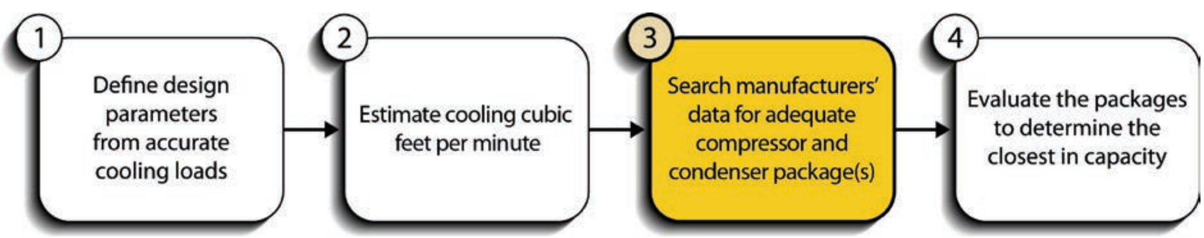

Figure 17. Cooling equipment selection step 3.

To search the fictitious example of a cooling equipment package, detailed cooling capacities, as shown in Figure 6, a nominal size must be determined. Considering that 12.00 MBtu/h is equivalent to 1 nominal ton of cooling, the 20,610 Btu/h total cooling load for the Chicago House leads to a 2-ton nominal size cooling system as the target size to begin the equipment search.

The EWB temperature for the indoor coil can be determined on the sea-level psychometric chart from the Manual J conditions of 75°F indoor design dry-bulb temperature and 50% relative humidity, as shown in Figure 18.

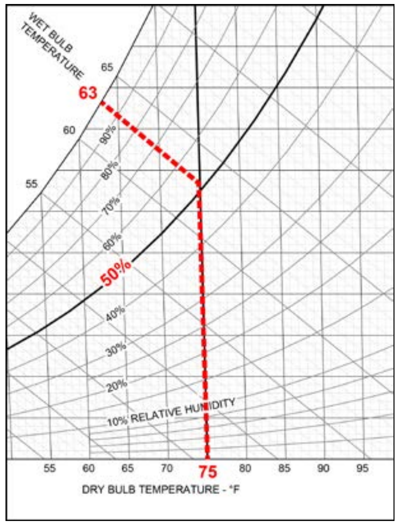

Figure 18. Wet-bulb temperature on sea-level psychometric chart.

The detailed cooling capacities tables give the data required for step 4 of Figure 19.

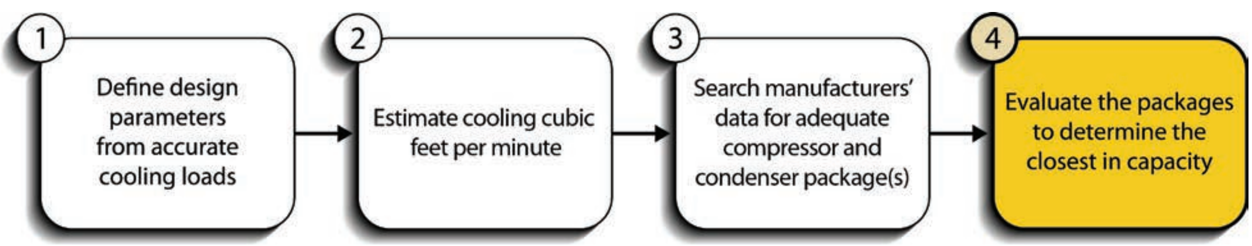

Figure 19. Cooling equipment selection step 4.

The outside design temperature for the Chicago House used in the Manual J load calculation was 89°F. The 89°F outdoor design temperature will require interpolation of the manufacturer's published data between the 85°F and 95°F condenser entering air temperature points.

The fictitious example cooling equipment package detailed cooling capacities table is searched at the closest published cfm data to the initial cfm airflow estimate of 831 cfm. Figure 20 highlights the 800 cfm and 63°F EWT temperature row, depicting a total capacity of 24.94 MBtu/h at 85°F and 23.73 MBtu/h at 95°F.

DETAILED COOLING CAPACITIES#

24ABC3

EVAPORATOR AIR		CONDENSER ENTERING AIR TEMPERATURES °F (°C)											
		75 (23.9)			85 (29.4)			95 (35)			105 (40.6)		
CFM	EWB °F (°C)	Capacity MBtuh		Total Sys. KW**	Capacity MBtuh		Total Sys. KW**	Capacity MBtuh		Total Sys. KW**	Capacity MBtuh		Total Sys. KW**
		Total	Sens‡		Total	Sens‡		Total	Sens‡		Total	Sens‡	
24ABC345 HIGH Outdoor Section With ZYXW3210 Indoor Section													
600	57 (13.9)	22.73	22.73	1.53	22.01	22.01	1.69	21.22	21.22	1.86	20.36	20.36	2.04
	62 (16.7)	24.15	20.31	1.54	23.17	19.64	1.70	22.12	18.95	1.86	20.99	18.24	2.05
	63 (17.2)††	24.66	16.79	1.54	23.66	16.15	1.70	22.58	15.49	1.87	21.42	14.82	2.05
	67 (19.4)	26.66	17.43	1.56	25.58	16.77	1.71	24.42	16.11	1.88	23.17	15.43	2.06
	72 (22.2)	29.46	14.50	1.58	28.27	13.87	1.73	27.00	13.23	1.90	25.65	12.59	2.09
715	57 (13.9)	24.27	24.27	1.56	23.46	23.46	1.72	22.58	22.58	1.89	21.64	21.64	2.07
	62 (16.7)	25.08	22.43	1.57	24.03	21.72	1.72	22.91	20.98	1.89	21.73	20.23	2.07
	63 (17.2)††	25.57	18.25	1.57	24.48	17.57	1.73	23.32	16.88	1.89	22.08	16.18	2.08
	67 (19.4)	27.62	18.98	1.59	26.45	18.30	1.74	25.20	17.60	1.91	23.87	16.89	2.09
	72 (22.2)	30.50	15.50	1.61	29.22	14.84	1.76	27.85	14.18	1.93	26.40	13.51	2.11
750	57 (13.9)	24.68	24.68	1.57	23.85	23.85	1.73	22.95	22.95	1.90	21.97	21.97	2.08
	62 (16.7)	25.32	23.05	1.58	24.25	22.33	1.73	23.12	21.58	1.90	22.01	22.01	2.08
	63 (17.2)††	25.79	18.67	1.58	24.69	17.99	1.74	23.50	17.29	1.90	22.24	16.58	2.08
	67 (19.4)	27.85	19.44	1.59	26.66	18.74	1.75	25.39	18.04	1.92	24.04	17.33	2.10
	72 (22.2)	30.75	15.79	1.62	29.44	15.13	1.77	28.06	14.46	1.94	26.58	13.78	2.12
800	57 (13.9)	25.23	25.23	1.59	24.36	24.36	1.74	23.43	23.43	1.91	22.42	22.42	2.09
	62 (16.7)	25.63	23.93	1.59	24.55	23.18	1.74	23.48	23.48	1.91	22.45	22.45	2.10
	63 (17.2)††	26.08	19.27	1.59	24.94	18.57	1.75	23.73	17.86	1.91	22.45	17.15	2.10
	67 (19.4)	28.15	20.08	1.61	26.93	19.37	1.76	25.63	18.66	1.93	24.26	17.94	2.11
	72 (22.2)	31.08	16.19	1.63	29.73	15.52	1.78	28.31	14.84	1.95	26.81	14.16	2.13

Figure 20. Fictitious example: detailed cooling capacities table, 800 cfm.

It is clear without going through the interpolation exercise that this cooling equipment total capacity at 800 cfm exceeds the sizing limits for the total Manual J cooling load of 15% larger than the peak total load (see Equation 5).

$$20.61\,\text{MBtu/h} \times 1.15 = 23.70\,\text{MBtu/h} \tag{5}$$

This is not an unexpected result because the initial 831 cfm airflow estimate is in alignment with the 400 cfm per 12.00 MBtu/h conventional methodologies of lower performing equipment and less efficient houses. It is not uncommon to find the target cfm range for a higher efficiency house with higher efficiency equipment to fall closer to 350 cfm per 12.00 MBtu/h.

Evaluating the same piece of equipment at lower cfm is the appropriate next step. The 715 cfm data row in Figure 21 shows the total capacity to be closer to the total load requirement. However, the sensible capacity values of 17.57 MBtu/h at 85°F and 16.88 MBtu/h at 95°F show, without interpolating between the values, that the package will likely not meet the sensible portion of the load (17.37 MBtu/h) at 89°F.

DETAILED COOLING CAPACITIES#

24ABC3

EVAPORATOR AIR		CONDENSER ENTERING AIR TEMPERATURES °F (°C)											
		75 (23.9)			85 (29.4)			95 (35)			105 (40.6)		
CFM	EWB °F (°C)	Capacity MBtuh Total	Capacity MBtuh Sens‡	Total Sys. KW**	Capacity MBtuh Total	Capacity MBtuh Sens‡	Total Sys. KW**	Capacity MBtuh Total	Capacity MBtuh Sens‡	Total Sys. KW**	Capacity MBtuh Total	Capacity MBtuh Sens‡	Total Sys. KW**
24ABC345 HIGH Outdoor Section With ZYXW3210 Indoor Section													
600	57 (13.9)	22.73	22.73	1.53	22.01	22.01	1.69	21.22	21.22	1.86	20.36	20.36	2.04
	62 (16.7)	24.15	20.31	1.54	23.17	19.64	1.70	22.12	18.95	1.86	20.99	18.24	2.05
	63 (17.2)††	24.66	16.79	1.54	23.66	16.15	1.70	22.58	15.49	1.87	21.42	14.82	2.05
	67 (19.4)	26.66	17.43	1.56	25.58	16.77	1.71	24.42	16.11	1.88	23.17	15.43	2.06
	72 (22.2)	29.46	14.50	1.58	28.27	13.87	1.73	27.00	13.23	1.90	25.65	12.59	2.09
715	57 (13.9)	24.27	24.27	1.56	23.46	23.46	1.72	22.58	22.58	1.89	21.64	21.64	2.07
	62 (16.7)	25.08	22.43	1.57	24.03	21.72	1.72	22.91	20.98	1.89	21.73	20.23	2.07
	63 (17.2)††	25.57	18.25	1.57	**24.48**	**17.57**	1.73	**23.32**	**16.88**	1.89	22.08	16.18	2.08
	67 (19.4)	27.62	18.98	1.59	26.45	18.30	1.74	25.20	17.60	1.91	23.87	16.89	2.09
	72 (22.2)	30.50	15.50	1.61	29.22	14.84	1.76	27.85	14.18	1.93	26.40	13.51	2.11
750	57 (13.9)	24.68	24.68	1.57	23.85	23.85	1.73	22.95	22.95	1.90	21.97	21.97	2.08
	62 (16.7)	25.32	23.05	1.58	24.25	22.33	1.73	23.12	21.58	1.90	22.01	22.01	2.08
	63 (17.2)††	25.79	18.67	1.58	**24.69**	**17.99**	1.74	**23.50**	**17.29**	1.90	22.24	16.58	2.08
	67 (19.4)	27.85	19.44	1.59	26.66	18.74	1.75	25.39	18.04	1.92	24.04	17.33	2.10
	72 (22.2)	30.75	15.79	1.62	29.44	15.13	1.77	28.06	14.46	1.94	26.58	13.78	2.12
800	57 (13.9)	25.23	25.23	1.59	24.36	24.36	1.74	23.43	23.43	1.91	22.42	22.42	2.09
	62 (16.7)	25.63	23.93	1.59	24.55	23.18	1.74	23.46	23.46	1.91	22.45	22.45	2.10
	63 (17.2)††	26.08	19.27	1.59	24.94	18.57	1.75	23.73	17.86	1.91	22.45	17.15	2.10
	67 (19.4)	28.15	20.08	1.61	26.93	19.37	1.76	25.63	18.66	1.93	24.26	17.94	2.11
	72 (22.2)	31.08	16.19	1.63	29.73	15.52	1.78	28.31	14.84	1.95	26.81	14.16	2.13

Figure 21. Fictitious example: detailed cooling capacities table, 715 cfm and 750 cfm.

The published 750 cfm data in Figure 21 for both total and sensible capacities appear to be within the range of the load requirements. Figure 22 shows the linear interpolation of the values to determine the capacity at 89°F. This analysis shows that the capacity of this package at 89°F will meet the total, sensible, and latent loads at 750 cfm. Utilizing a spreadsheet to handle the multiple interpolation calculations makes the procedure more manageable (see Figure 23).

City: Chicago, IL	Cooling Outdoor Design Temperature = 89 °F										
	Total Cooling Design Load = 20.6 MBtu/h										
	Sensible Cooling Design Load = 17.4 MBtu/h										
	Condenser Entering Air Temperatures °F										
	85	**86**	**87**	**88**	**89**	**90**	**91**	**92**	**93**	**94**	**95**
Total Cooling (CFM)	**Total Capacity (MBtu/h)**										
600	23.66	23.55	23.44	23.34	23.23	23.12	23.01	22.90	22.80	22.69	22.58
715	24.45	24.33	24.22	24.10	23.98	23.87	23.75	23.63	23.52	23.40	23.29
750	24.69	24.57	24.45	24.33	24.21	24.10	23.98	23.86	23.74	23.62	23.50
	Sensible Capacity (MBtu/h)										
750	17.99	17.92	17.85	17.78	17.71	17.64	17.57	17.50	17.43	17.36	17.29

Figure 22. Interpolation table for capacities at 89°F outdoor design temperature.

The fictitious example cooling equipment package will exceed our cooling load requirements by 17% at 750 cfm. As previously stated, in energy efficient houses the loads may be so small that the standard manufacturer's range of capacities could exceed the 15% sizing limits. Equipment must be selected to meet the peak load without grossly oversizing the equipment, and the selection should be based on the selection procedures backed up by the data.

The target 750 cfm value will be carried through the remainder of the system design procedures; however, this cfm value may be revisited and changed when the duct design is done. A furnace for the Chicago House can now be selected based on the cooling equipment selection and the Manual J heating loads.

3.2 Furnace Selection

With the cooling equipment package selected, a furnace must be selected that meets the heating load. The furnace capacity must meet the Manual J heating load while not exceeding the heating load by more than 40%. From Manual S, Section 2-5, the steps to selecting a furnace are shown in Figure 23.

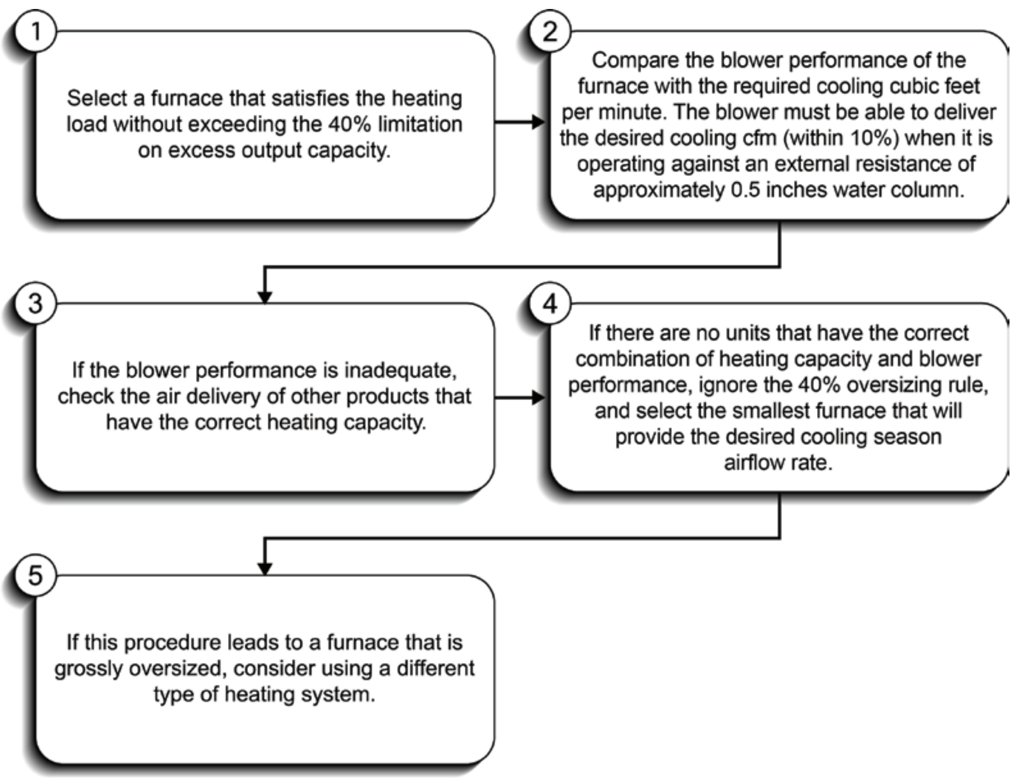

Figure 23. Furnace selection steps.

Figure 24 shows that furnace capacities span a much larger range than cooling equipment capacities. A particular manufacturer may size a furnace line from 60,000 Btu/h to 120,000 Btu/h with 80,000 Btu/h as the intermediate size. This can be alleviated somewhat with multiple-stage equipment. A two-stage furnace in high stage may have a rating of 60,000 Btu/h but can be 39,000 Btu/h in low stage. Furnaces are rated as the input Btu/h, but the capacity is derated by the efficiency level. For example, a 60,000 Btu/h furnace with a 92.5% efficiency rating will yield around 55,500 Btu/h output capacity.

Speed Setting
Orientation
Capacity Column

SPECIFICATIONS

UNIT SIZE			060–12	080–12	080–16	100–20	120–20
CAPACITY* Direct Vent (2 – pipe) (Shaded capacities are specified on the rating plate)	Low	Upflow	37000	49000	49000	61000	73000
		Downflow	36000	49000	49000	61000	73000
		Horizontal	36000	49000	49000	61000	73000
	High	Upflow	56000	75000	75000	93000	113000
		Downflow	56000	75000	75000	93000	112000
		Horizontal	56000	74000	74000	93000	112000
CAPACITY* Non–Direct vent (1–pipe)	Low	Upflow	36000	49000	49000	61000	73000
		Downflow	36000	49000	49000	61000	73000
		Horizontal	36000	48000	49000	61000	73000
	High	Upflow	56000	75000	75000	94000	113000
		Downflow	56000	75000	75000	93000	112000
		Horizontal	56000	74000	74000	93000	112000
AFUE* Direct Vent (2–pipe) Nonweatherized ICS		Upflow	92.5				
		Downflow	91				
		Horizontal	92.1				
AFUE* Non–Direct Vent (1–pipe) Nonweatherized ICS		Upflow	92.1				
		Downflow	90.7				
		Horizontal	91.7				
Input Btuh {	Low		39000	52000	52000	65000	78000
	High		60000	80000	80000	100000	120000
CERTIFIED TEMPERATURE RANGE(F)	High		30–60 16.5–33)	40–70 (22–38.5)	30–60 16.5–33)	30–60 16.5–33)	40–70 (22–38.5)
	Low		20–50 (11–27.5)	30–60 16.5–33)	30–60 16.5–33)	30–60 16.5–33)	30–60 16.5–33)
CERTIFIED EX STATIC PRESSURE (IN. W.C.)		Heating	0.12	0.15	0.15	0.20	0.20
		Cooling	0.50	0.50	0.50	0.50	0.50
AIRFLOW (CFM}		High Heat	1040	1180	1450	1890	2065
		Low Heat	635	835	910	935	1425
		Cooling	1250	1240	1620	1635	1925

98JKL

Figure 24. Furnace specification chart.

Each manufacturer may tabulate the furnace performance data differently. The furnace must also dimensionally match the size of the indoor coil. The example equipment manufacturer furnace is matched to the indoor cooling coil size and type, but not all manufacturers treat coil matching the same way. Some may require that the designers verify the furnace and coil are dimensionally compatible. Taking time to understand how the furnace data are formatted is important to making the correct selection.

Figure 25 is an example of an air delivery performance table. The blower performance of the furnace and the required cooling cfm must be compared. While comparing the blower performance of the furnace and the required cooling cfm, the blower must be able to meet the cooling cfm within 10%. The assumption at the start of the equipment selection process is 0.50 inch water column external static pressure for the system. The equipment should have the capability of field setting the blower speed to match the cooling cfm per the manufacturer's instructions.

AIR DELIVERY - CFM (WITH FILTER)

UNIT SIZE	RETURN–AIR SUPPLY	SPEED	EXTERNAL STATIC PRESSURE (IN. W.C.)									
			0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1
060–12	SIDE/BOTTOM	5(Gry)	1430	1390	1345	1300	1250	1205	1150	1080	995	845
		3(Blu)	1240	1200	1145	1100	1040	975	915	860	790	730
		4(Yel)	1090	1030	980	935	850	800	730	665	590	525
		1(Red)	900	835	780	705	635	565	490	410	335	200
		2(Org)	805	620	440	380	300	–	–	–	–	–
		x(xxx)	xxxx	xxxx	xxxx	xxxx	xxxx	xxxx	xxxx	xxxx	xxxx	xxxx

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Equipment is field adjustable to desired CFM

Design assumption is 0.50 iwc external static pressure

Figure 25. Air delivery performance table.

To select a furnace for the Chicago House, the heating load estimate from the Manual J procedure of 41,660 Btu/h must be satisfied. The two parameters that must be considered for selection are the heating capacity and the cfm airflow. Equation 6 sets the maximum target capacity of a furnace for the Chicago House.

$$41{,}660\text{ Btu/h} \times 1.4 = 58{,}324\text{ Btu/h} \tag{6}$$

The furnace with specifications as shown in Figure 26 will meet the required load without exceeding the maximum target capacity in the upflow configuration at the high setting. The cooling airflow of 1250 cfm almost doubles the 750 cfm design airflow determined in the cooling equipment selection; however, Figure 27 shows that the equipment is field adjustable to within the airflow margin constraint of 0.5 inch water column external static pressure.

SPECIFICATIONS

UNIT SIZE			060–12	080–12	080–16	100–20	120–20
CAPACITY* Direct Vent (2 – pipe) (Shaded capacities are specified on the rating plate)	Low	Upflow	37000	49000	49000	61000	73000
		Downflow	36000	49000	49000	61000	73000
		Horizontal	36000	49000	49000	61000	73000
	High	Upflow	56000	75000	75000	93000	113000
		Downflow	56000	75000	75000	93000	112000
		Horizontal	56000	74000	74000	93000	112000
CAPACITY* Non–Direct vent (1–pipe)	Low	Upflow	36000	49000	49000	61000	73000
		Downflow	36000	49000	49000	61000	73000
		Horizontal	36000	48000	49000	61000	73000
	High	Upflow	56000	75000	75000	94000	113000
		Downflow	56000	75000	75000	93000	112000
		Horizontal	56000	74000	74000	93000	112000
AFUE* Direct Vent (2–pipe) Nonweatherized ICS		Upflow	92.5				
		Downflow	91				
		Horizontal	92.1				
AFUE* Non–Direct Vent (1–pipe) Nonweatherized ICS		Upflow	92.1				
		Downflow	90.7				
		Horizontal	91.7				
Input Btuh {	Low		39000	52000	52000	65000	78000
	High		60000	80000	80000	100000	120000
CERTIFIED TEMPERATURE RANGE(F)	High		30–60 16.5–33)	40–70 (22–38.5)	30–60 16.5–33)	30–60 16.5–33)	40–70 (22–38.5)
	Low		20–50 (11–27.5)	30–60 16.5–33)	30–60 16.5–33)	30–60 16.5–33)	30–60 16.5–33)
CERTIFIED EX STATIC PRESSURE (IN. W.C.)		Heating	0.12	0.15	0.15	0.20	0.20
		Cooling	0.50	0.50	0.50	0.50	0.50
AIRFLOW (CFM}		High Heat	1040	1180	1450	1890	2065
		Low Heat	635	835	910	935	1425
		Cooling	1250	1240	1620	1635	1925

98JKL

Figure 26. Fictitious furnace performance example.

AIR DELIVERY - CFM (WITH FILTER)

UNIT SIZE	RETURN–AIR SUPPLY	SPEED	EXTERNAL STATIC PRESSURE (IN. W.C.)									
			0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1
060–12	SIDE/BOTTOM	5(Gry)	1430	1390	1345	1300	1250	1205	1150	1080	995	845
		3(Blu)	1240	1200	1145	1100	1040	975	915	860	790	730
		4(Yel)	1090	1030	980	935	850	800	730	665	590	525
		1(Red)	900	835	780	705	635	565	490	410	335	200
		2(Org)	805	620	440	380	300	–	–	–	–	–
		x(xxx)	xxxx	xxxx	xxxx	xxxx	xxxx	xxxx	xxxx	xxxx	xxxx	xxxx

98JKL

Figure 27. Air delivery performance table for the Chicago House.

It is possible that conditions will arise from the duct design that may change the external static pressure and may require a different piece of equipment be chosen. The iterative nature of the equipment selection process is in balancing the capacity of the equipment to meet the load with the blower performance.

4 Orlando House Example

The Orlando House example equipment choice is a 9.2 HSPF/16 SEER air source heat pump. The selection of an air source heat pump is driven by climatic conditions, the size of the cooling load compared to the heating load, and the economic factors of gas prices compared to the cost of electricity. Air source heat pump systems have not traditionally been a common choice in cold climates with a large heating load; however, an air source heat pump is a likely choice for climates where the heating load is much smaller than the cooling load. Dual fuel options can make an air source heat pump a viable choice for a cold climate by providing an economical solution of a gas-fired burner to meet the larger heating load in cold climates. This evaluation will require an in-depth cost comparison. Details of the economic balance-point evaluation for a dual fuel system are covered in Manual S, Section 6.

Initially, heat pump equipment selection is based on the cooling load. The selection of heat pump equipment to meet the cooling capacity follows the selection of cooling equipment in the air conditioner selection in Section 3 of Manual S and is sensitive to the same four operating parameters as the split system cooling equipment. In warm climates, the sizing limitations of an air source heat pump are 15% greater than the Manual J loads for heating and total cooling loads, whereas in cold climates, it is permissible to exceed the Manual J heating load by 25%.

Because no furnace is used in this particular heat pump system, an indoor fan coil air handler with electric resistance heat coils will be selected to match the performance requirements and design parameters for airflow and heating load. The additional information required for heat pump selection is the amount of supplemental heat and the emergency heat capacity.

Based on Section 5 of Manual S, the steps to selecting a heat pump are shown in Figure 28.

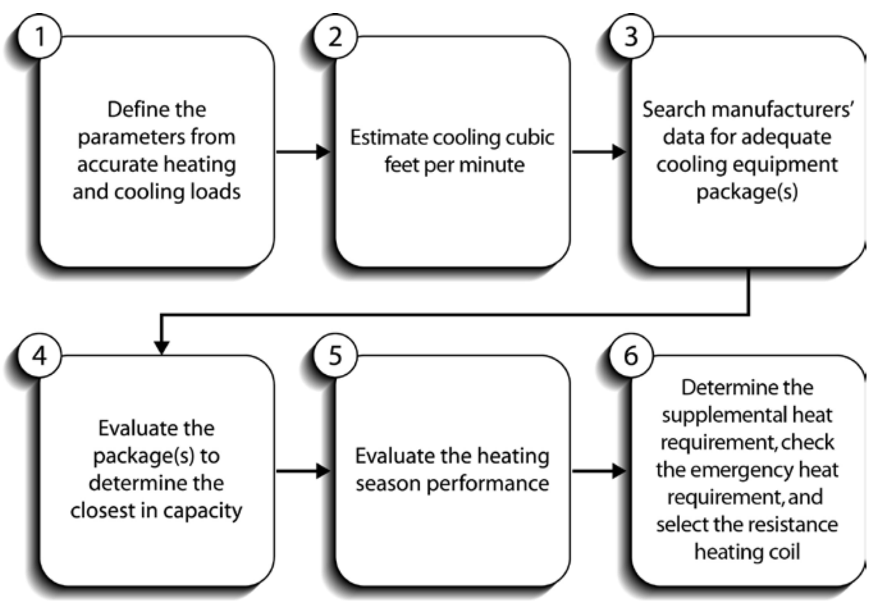

Figure 28. Air source heat pump selection steps.

Once the cooling equipment selection has been made, a balance-point diagram can be created from the manufacturer's published performance data and the heating season outdoor design temperatures. The balance-point diagram is used to determine the amount of supplemental and emergency heat required. Figure 29 shows the development of the balance-point diagram. The capacity line (A) is provided by the manufacturer in the detailed equipment specifications. The load line (B) is created by drawing a straight line from the design heating load at the outdoor design temperature to 0 Btu/h load at 65°F outdoor temperature. Emergency heat (C) is the total design heating load. Supplemental heat (D) is the difference between the capacity line and the design heating load.

Supplemental Heat –
The amount of electric resistance heat beyond the heat pump capacity required to meet the peak heating load.

Emergency Heat –
The amount of electric resistance heat required to meet the peak heating load should the heat pump fail.

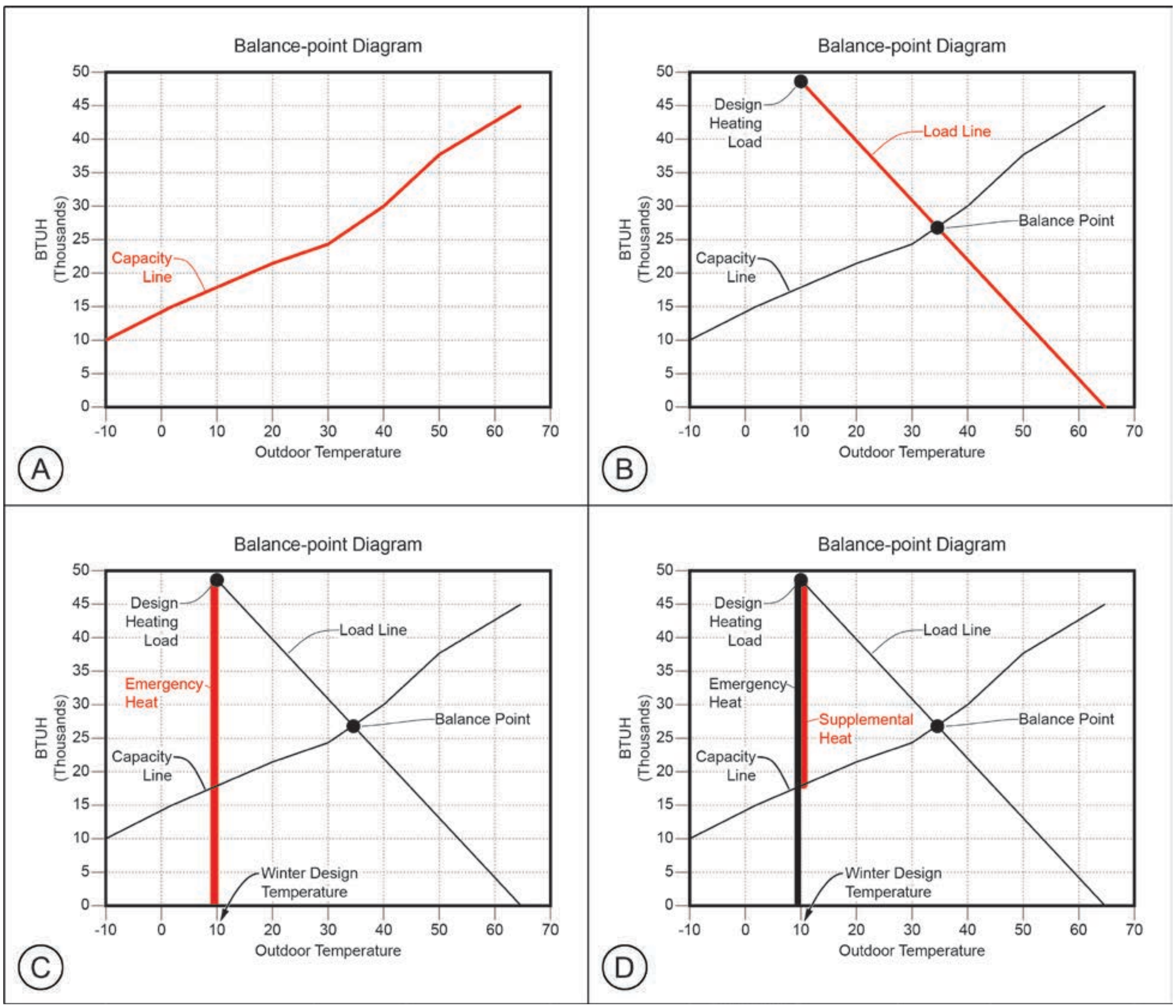

Figure 29. Development of balance-point diagram.

4.1 Air Source Heat Pump Selection

The first step highlighted in Figure 30 instructs the designer to define the cooling design parameters from the cooling loads of the Manual J procedure. The cooling design information developed by Burdick (2011) from the Manual J procedures is shown in Figure 31.

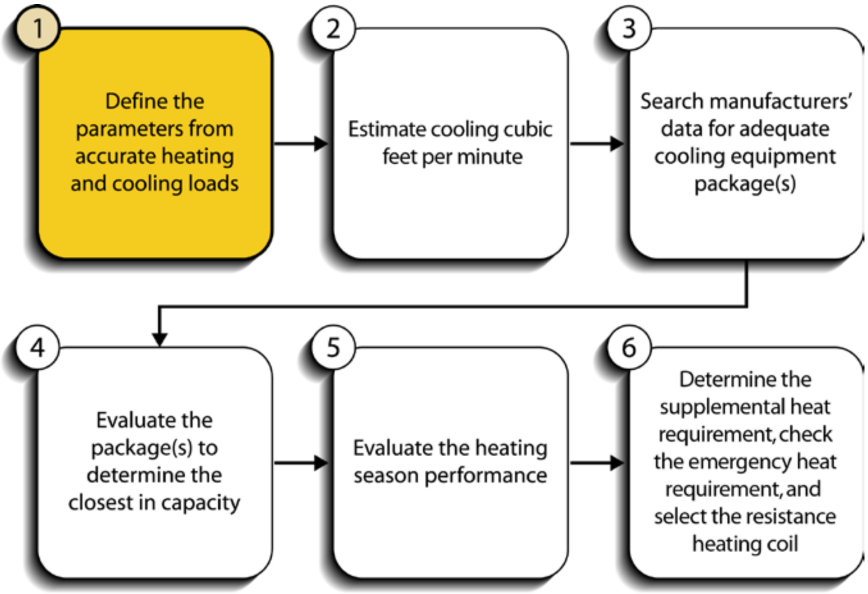

Figure 30. Air source heat pump selection step 1.

Manual J	Heating Load	23,640 Btu/h
	Sensible Cooling Load	16,630 Btu/h
	Latent Cooling Load	4,060 Btu/h
Outdoor Conditions	Cooling Dry-bulb Temperature	92 °F
	Heating Dry-bulb Temperature	38 °F
Indoor Conditions	Cooling Dry-bulb Temperature	75 °F
	Cooling Wet-bulb Temperature	63 °F
	Heating Dry-bulb Temperature	70 °F
Airflow Estimates	Cubic Feet per Minute	719 cfm

Figure 31. Orlando House design information.

4.2 Heat Pump Cooling Capacity Selection

Step 2 of the selection process, highlighted in Figure 32, instructs the designer to estimate the cooling cfm. The methods outlined in Section 1-3 of Manual S are used to estimate the initial cfm airflow.

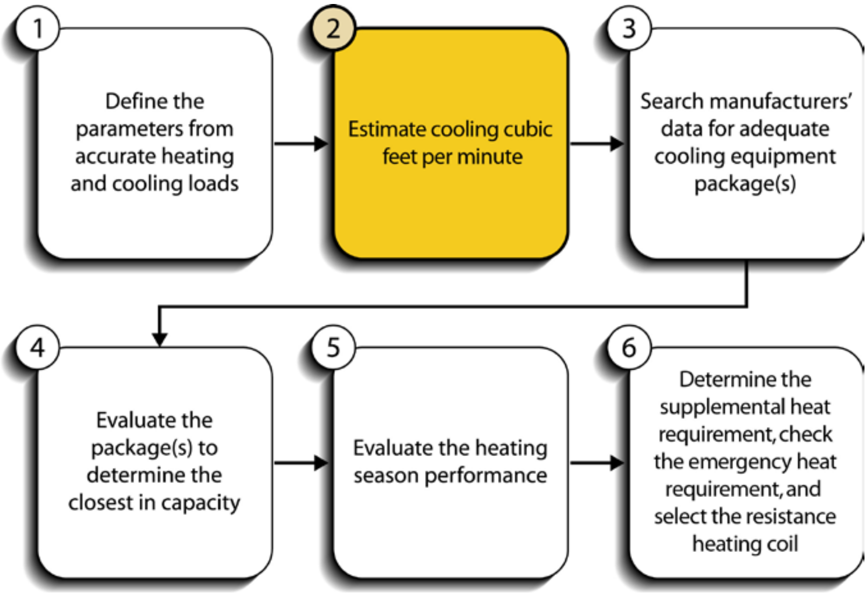

Figure 32. Air source heat pump selection step 2.

Equation 7 applies the cooling design parameters from Figure 31 to Equation 1, resulting in an SHR of 0.80.

$$\text{SHR} = \frac{\text{Sensible Load}}{\text{Total Load}} = \frac{16.63\,\frac{\text{MBtu}}{\text{h}}}{20.69\,\frac{\text{MBtu}}{\text{h}}} = 0.80 \tag{7}$$

From the TD value table (Figure 11), the SHR value of 0.80 is the break point between the TD values of 19 and 21. Note that a higher TD value will lead to a lower target cfm value, and a lower cfm value is desirable for a humid climate. Therefore, a TD of 21 was selected for this Orlando House example in Florida. This is the type of design consideration that can be applied to the equipment selection procedure.

With the TD value for SHR = 0.80 of 21 selected, these values can now be applied to the sensible heat equation (Equation 2) as shown in Equation 8.

$$\text{cfm Target} = \frac{\text{Manual J Sensible Load}}{1.1 \times \text{TD}} = \frac{16{,}630}{1.1 \times 21} = 719 \tag{8}$$

Step 3, highlighted in Figure 33, is to search the manufacturer's data for an adequate cooling equipment package. To search the fictitious example of a cooling equipment package detailed cooling capacities, a nominal size must be determined. Considering that 12.00 MBtu/h is equivalent to 1 nominal ton of cooling, the 20,690 Btu/h total cooling load for the Orlando House leads to a 2-ton nominal cooling system as the target size to begin the equipment search.

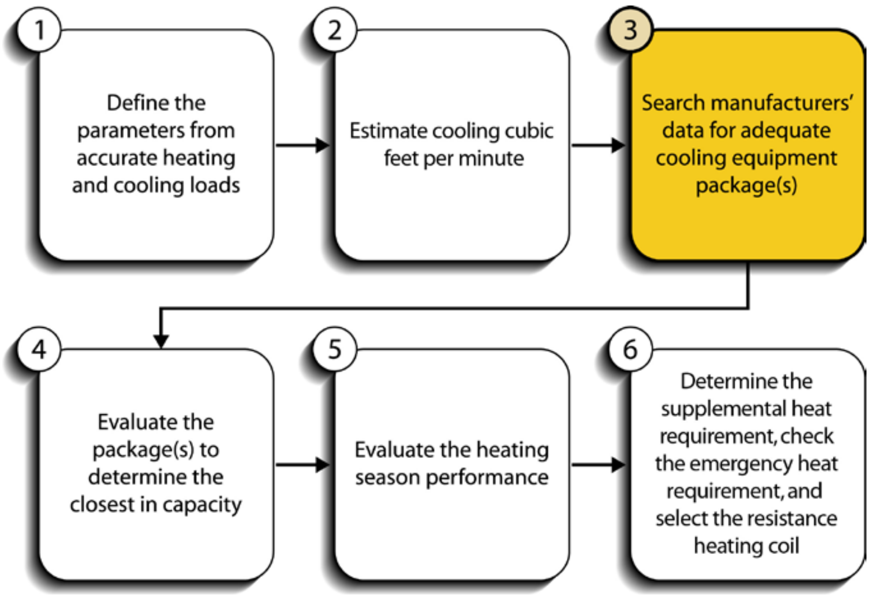

Figure 33. Air source heat pump selection step 3.

The EWB temperature for the indoor coil can be determined on the psychometric chart from the Manual J conditions of 75°F indoor design dry-bulb temperature and 50% relative humidity, as shown in Figure 34.

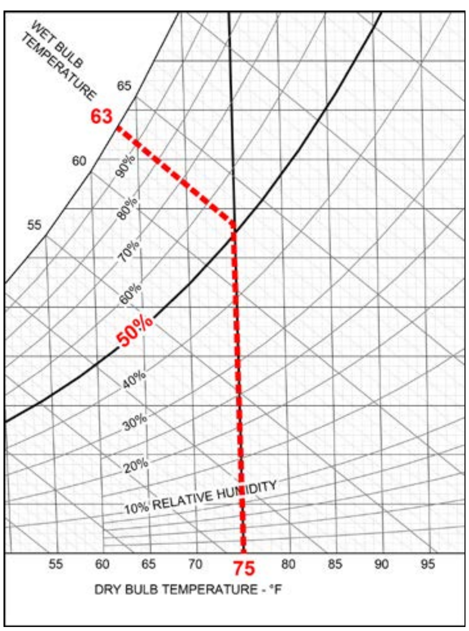

Figure 34. Wet-bulb temperature on sea-level psychometric chart.

The detailed cooling capacities tables give the data required for step 4 (see Figure 35). The outside design temperature for the Orlando House used in the Manual J load calculation was 92°F. The 92°F outdoor design temperature will require interpolation of the manufacturer's published data between the 85°F and 95°F condenser entering air temperature points.

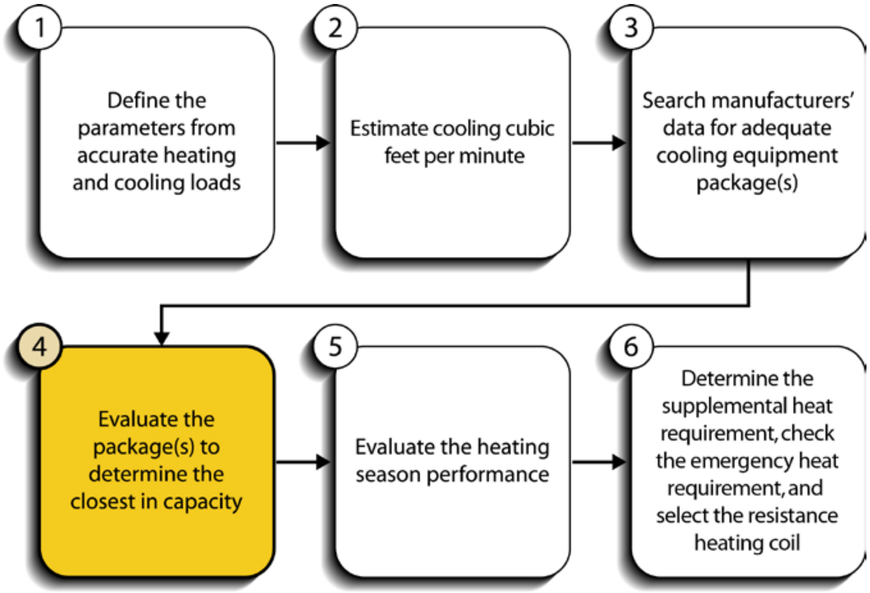

Figure 35. Cooling equipment selection step 4.

The fictitious example heat pump detailed cooling capacities table (Figure 36) is searched at the closest published cfm data to the initial cfm airflow estimate of 719 cfm. Figure 36 highlights the 700 cfm and 63°F entering evaporator wet-bulb temperature row. The published 700 cfm data in Figure 36 for both total and sensible capacities appear to be within the range of the load requirements. Figure 37 shows the linear interpolation of the values to determine the capacity at 92°F. This analysis shows that the capacity of this package at 92°F will meet the total, sensible, and latent loads at 700 cfm. Utilizing a spreadsheet to handle the multiple interpolation calculations makes the procedure more manageable, as shown in Figure 37.

24XYZ

DETAILED COOLING CAPACITIES#

EVAPORATOR AIR		CONDENSER ENTERING AIR TEMPERATURES °F (°C)											
		75 (23.9)			85 (29.4)			95 (35)			105 (40.6)		
CFM	EWB °F (°C)	Capacity MBtuh Total	Capacity MBtuh Sens‡	Total Sys. KW**	Capacity MBtuh Total	Capacity MBtuh Sens‡	Total Sys. KW**	Capacity MBtuh Total	Capacity MBtuh Sens‡	Total Sys. KW**	Capacity MBtuh Total	Capacity MBtuh Sens‡	Total Sys. KW**
24XYZ01234 Outdoor Section With AB3CDE456 Indoor Section													
600	72 (22.2)	27.16	14.15	1.44	25.86	13.63	1.63	24.49	13.06	1.82	23.07	12.47	2.05
	67 (19.4)	24.67	16.90	1.42	23.49	16.34	1.60	22.25	15.75	1.80	20.95	15.14	2.02
	63 (17.2)††	22.75	16.19	1.41	21.66	15.63	1.59	20.50	15.05	1.78	19.29	14.42	2.00
	62 (16.7)	22.41	19.59	1.40	21.34	19.02	1.58	20.21	18.42	1.78	19.02	17.76	2.00
	57 (13.9)	21.04	21.04	1.39	20.22	20.22	1.57	19.33	19.33	1.77	18.39	18.39	1.99
700	72 (22.2)	28.13	15.04	1.47	26.75	14.46	1.65	25.30	13.86	1.85	23.79	13.24	2.07
	67 (19.4)	25.58	18.21	1.45	24.32	17.62	1.63	23.00	17.00	1.83	21.61	16.36	2.05
	63 (17.2)††	23.61	17.41	1.43	**22.44**	**16.82**	1.61	**21.20**	**16.20**	1.81	19.91	15.56	2.03
	62 (16.7)	23.27	21.34	1.43	22.13	20.72	1.61	20.94	20.05	1.80	19.70	19.32	2.02
	57 (13.9)	22.33	22.33	1.42	21.43	21.43	1.60	20.46	20.46	1.80	19.43	19.43	2.02
735	72 (22.2)	28.43	15.32	1.48	27.01	14.74	1.66	25.54	14.13	1.86	24.00	13.50	2.08
	67 (19.4)	25.86	18.65	1.46	24.57	18.05	1.64	23.22	17.42	1.83	21.81	16.77	2.05
	63 (17.2)††	23.86	17.82	1.44	22.67	17.22	1.62	21.41	16.58	1.82	20.10	15.93	2.03
	62 (16.7)	23.53	21.93	1.44	22.38	21.28	1.62	21.17	20.58	1.81	19.93	19.81	2.03
	57 (13.9)	22.74	22.74	1.43	21.80	21.80	1.61	20.81	20.81	1.81	19.76	19.76	2.03
800	72 (22.2)	28.91	15.83	1.49	27.45	15.23	1.67	25.93	14.61	1.87	24.35	13.97	2.09
	67 (19.4)	26.31	19.45	1.47	24.98	18.83	1.65	23.59	18.19	1.85	22.13	17.52	2.07
	63 (17.2)††	24.28	18.56	1.45	23.05	17.94	1.63	21.75	17.30	1.83	20.40	16.63	2.05
	62 (16.7)	23.98	22.97	1.45	22.80	22.27	1.63	21.58	21.51	1.83	20.35	20.35	2.05
	57 (13.9)	23.44	23.44	1.44	22.46	22.46	1.63	21.42	21.42	1.83	20.31	20.31	2.05

Figure 36. Fictitious example: detailed heat pump cooling capacities table, 700 cfm.

City: Orlando, FL	Cooling Outdoor Design Temperature = 92 °F										
	Total Cooling Design Load = 20.7 MBtu/h										
	Sensible Cooling Design Load = 17.4 MBtu/h										
	Condenser Entering Air Temperatures °F										
	85	**86**	**87**	**88**	**89**	**90**	**91**	**92**	**93**	**94**	**95**
Total Cooling (CFM)	**Total Capacity (MBtu/h)**										
700	22.44	22.32	22.19	22.07	21.82	21.82	21.70	**21.57**	21.45	21.32	21.20
735	22.65	22.53	22.40	22.28	22.15	22.02	21.90	21.77	21.64	21.52	21.39
800	23.05	22.92	22.79	22.66	22.53	22.40	22.27	22.14	22.01	21.88	21.80
	Sensible Capacity (MBtu/h)										
700	16.82	16.76	16.70	16.63	16.57	16.51	16.45	**16.39**	16.32	16.26	16.20

Figure 37. Interpolation table for cooling capacities at 92°F outdoor design temperature.

4.3 Heat Pump Heating Capacity Selection

To complete step 5 highlighted in Figure 38 and to evaluate the heating season performance of the heat pump, the heating capacity must be determined from the heat pump heating performance table as shown in Figure 39, and interpolation to the 38°F heating design temperature will be required. The indoor heating design temperature from Manual J is 70°F, and the design cfm determined in the cooling selection is 700 cfm.

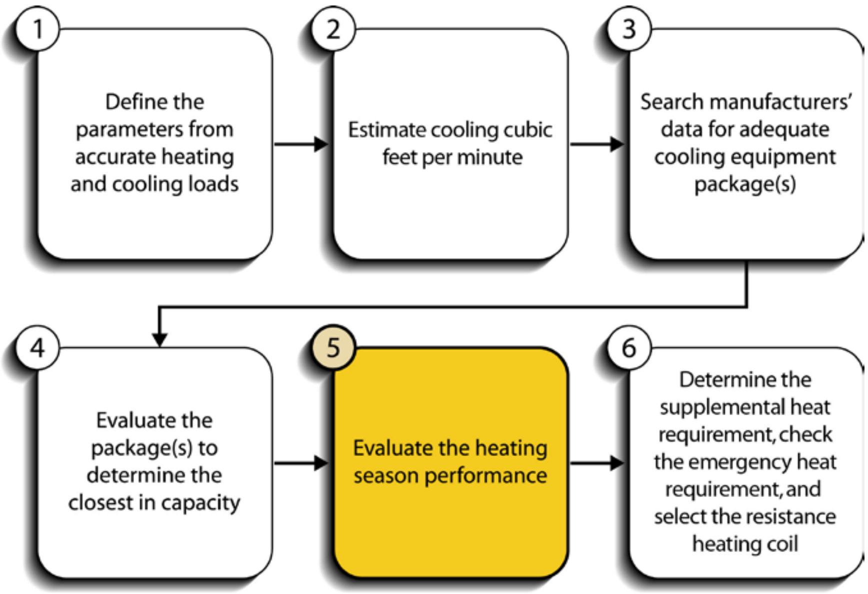

Figure 38. Cooling equipment selection step 5.

HEAT PUMP HEATING PERFORMANCE

24XYZ

INDOOR AIR		OUTDOOR COIL ENTERING AIR TEMPERATURES °F (°C)														
		−3 (−19.4)			7 (−13.9)			17 (−8.3)			27 (−2.8)			37 (2.8)			47 (8.3)		
EDB °F (°C)	CFM	Capacity MBtuh		Total Sys. KW†	Capacity MBtuh		Total Sys. KW†	Capacity MBtuh		Total Sys. KW†	Capacity MBtuh		Total Sys. KW†	Capacity MBtuh		Total Sys. KW†	Capacity MBtuh		Total System KW†
		Total	Integ*		Total	Integ*		Total	Integ*		Total	Integ*		Total	Integ*		Total	Integ*	
24XYZ01234 Outdoor Section With AB3CDE456 Indoor Section																			
65 (18.3)	600	9.85	9.06	1.48	12.27	11.28	1.58	15.08	13.75	1.70	18.30	16.25	1.83	21.96	19.98	1.99	26.11	26.11	2.19
	700	9.98	9.18	1.47	12.45	11.44	1.56	15.31	13.96	1.67	18.59	16.51	1.79	22.31	20.30	1.95	26.46	26.46	2.15
	735	10.01	9.21	1.47	12.50	11.49	1.56	15.38	14.02	1.66	18.68	16.59	1.78	22.40	20.39	1.94	26.55	26.55	2.15
	800	10.08	9.28	1.46	12.59	11.57	1.55	15.49	14.12	1.65	18.80	16.70	1.77	22.54	20.51	1.93	26.67	26.67	2.14
70 (21.1)	600	9.62	8.85	1.56	12.01	11.04	1.67	14.77	13.47	1.78	17.93	15.93	1.92	21.54	19.60	2.08	25.64	25.64	2.29
	700	9.77	8.99	1.55	12.19	11.20	1.65	15.00	13.67	1.75	18.22	16.18	1.88	21.89	19.92	2.03	26.00	26.00	2.24
	735	9.81	9.03	1.55	12.24	11.25	1.64	15.06	13.73	1.75	18.31	16.26	1.87	21.98	20.00	2.02	26.09	26.09	2.23
	800	9.88	9.09	1.55	12.33	11.33	1.64	15.18	13.84	1.74	18.44	16.38	1.86	22.13	20.14	2.01	26.22	26.22	2.23
75 (23.9)	600	9.20	8.47	1.64	11.75	10.80	1.75	14.46	13.18	1.87	17.57	15.60	2.01	21.12	19.22	2.17	25.15	25.15	2.38
	700	9.51	8.75	1.63	11.92	10.95	1.73	14.68	13.38	1.84	17.85	15.85	1.97	21.46	19.53	2.12	25.52	25.52	2.33
	735	9.55	8.79	1.63	11.98	11.01	1.73	14.75	13.45	1.83	17.93	15.93	1.96	21.57	19.63	2.11	25.62	25.62	2.32
	800	9.63	8.86	1.63	12.07	11.09	1.72	14.86	13.55	1.82	18.07	16.05	1.94	21.72	19.76	2.10	25.75	25.75	2.31

Figure 39. Heat pump heating performance table at 700 cfm.

Interpolation of the capacity at 38°F is shown in Figure 40. This equipment's capacity of 22.73 MBtu/h closely matches the heating load of 23.64 MBtu/h, and the balance will be made up with supplemental heat as determined in step 6 (Figure 41). Step 6 is to determine the supplemental and emergency heat requirements from the balance-point diagram.

	Heating Outdoor Design Temperature = 38 °F										
	Heating Design Load = 23.6 MBtu/h										
	Condenser Entering Air Temperatures °F										
	37	**38**	**39**	**40**	**41**	**42**	**43**	**44**	**45**	**46**	**47**
Total Heating (CFM)	**Total Capacity (MBtu/h)**										
700	22.31	**22.73**	23.14	23.56	23.97	24.39	24.80	25.22	25.63	26.05	26.46
735	22.39	22.80	23.22	23.63	24.05	24.46	24.88	25.29	25.70	26.12	26.53
800	22.54	22.95	23.37	23.78	24.19	24.61	25.02	25.43	25.84	26.26	26.70

Figure 40. Interpolation table for heating capacities at 38°F outdoor design temperature.

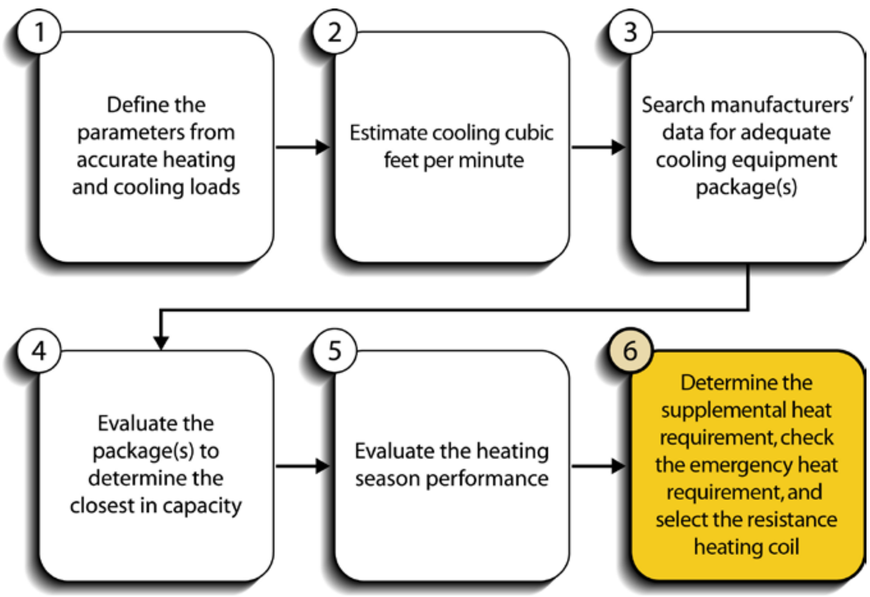

Figure 41. Cooling equipment selection step 6.

The balance-point diagram, Figure 42, for the selected outdoor heat exchanger and fan coil shows a small 3,000 Btu/h supplemental heat requirement.

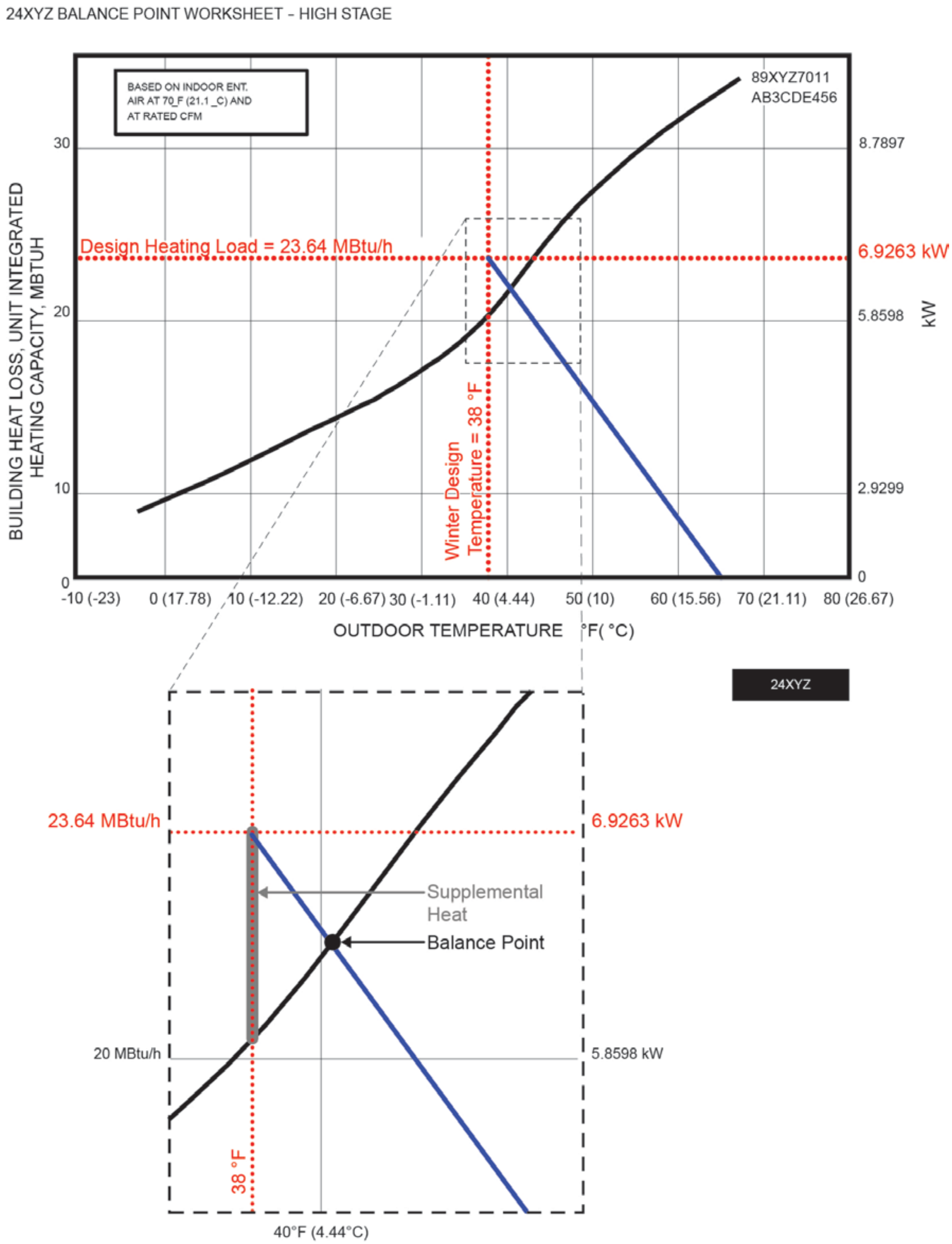

Figure 42. Balance-point diagram for the Orlando House.

Supplemental and emergency heat are supplied by electric resistance heat and are sized in kilowatts, requiring a conversion from Btu/h. The 23.64 MBtu/h total load requirement converts to 6.9 kW emergency heat, and the supplemental heat requirement is less than 1 kW. This supplemental heating capacity will be specified in the sizing of the fan coil.

The electric resistance coils will be specified with the fan coil sizing shown in Figure 43. The available coils are 5, 8, and 10 kW. The 8 kW is larger than the required 6.9 kW, but this is resolved by staged resistance heat providing resistance heat in eight individual 1-kW increments.

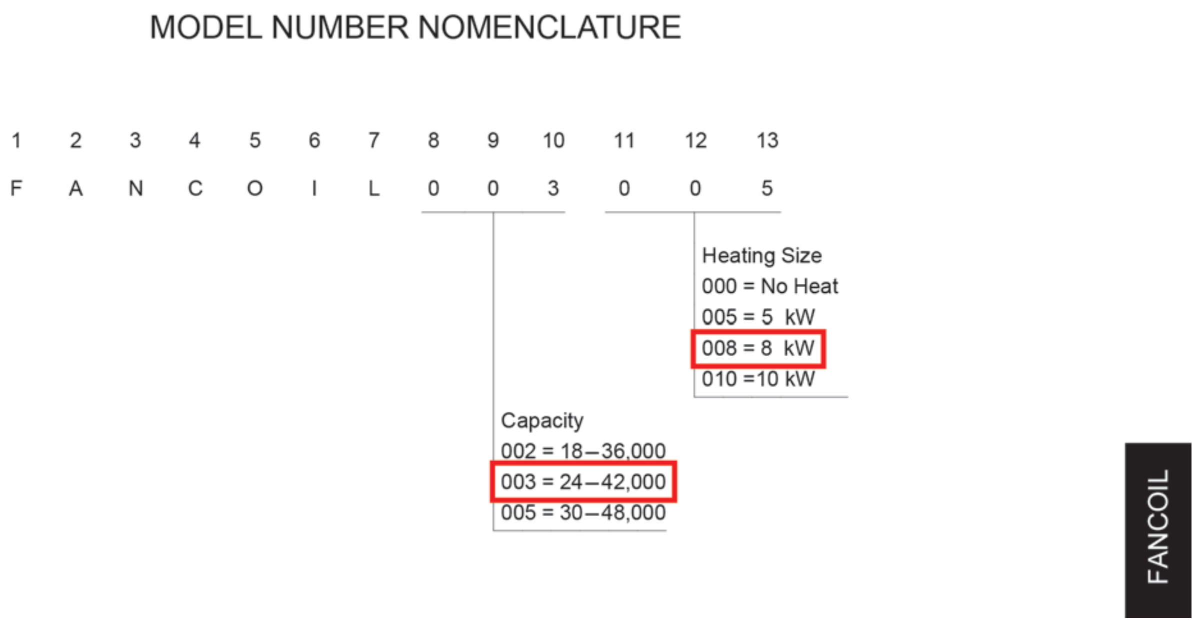

Figure 43. Fan coil nomenclature.

We now have a heat pump and fan coil package that will meet the heating and cooling load needs of the Orlando House. The next step in right-sizing the HVAC system is to use the blower performance data of the fan coil to create the air delivery duct design.

5 Conclusion

Right-sizing of an HVAC system is the selection of equipment and the design of the air distribution system to meet the accurate predicted heating and cooling loads of the house. The equipment effective capacity should match the loads generated in the load calculation procedure within the sizing guidelines. Selection of the equipment can have a substantial impact on the overall efficiency and operating costs of the installed system.

Note that the design of a space conditioning system is an iterative process, and once a duct system configuration and preliminary design are completed, different equipment may need to be selected to achieve the proper airflow to satisfy the room-by-room heating and cooling loads. The design loads along with the indoor and outdoor conditions information needed for equipment selection are all taken from the inputs used in the Manual J procedure. Faithful adherence to the conditions used in Manual J is required for accurate equipment selection.

The selection of heating and cooling equipment is dependent on accurate heating and cooling load calculations. From this point onward, in the design process, the actual performance data of the equipment selections and room-by-room loads from the Manual J calculations should be used to design the duct system.

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