Volume 5, Issue 1

CARRIER® eDESIGN SUITE NEWS

EXchange

Interpreting High (Low) Peak Design Airflow Sizing Results for HVAC Equipment Selection

A design challenge sometimes occurs when computing design loads using software such as Carrier's Hourly Analysis Program (HAP) or Block Load, whereby under certain design conditions the peak design load airflow (CFM) may seem to be unrealistic if it falls outside the "typical" heating, ventilating and air-conditioning (HVAC) equipment selection range of 300-500 CFM/ton for packaged equipment.

This is a common point of confusion for many users. They mistakenly think HAP is supposed to automatically yield a design airflow quantity that allows them to select a particular type of HVAC equipment. So even though you may select a constant air volume (CAV) roof top unit (RTU) or other type of equipment as the Equipment or System Type, HAP does not automatically tailor the cooling/heating load calculation results to comply with any particular set of operational constraints imposed by any specific type of HVAC equipment. Rather, HAP computes theoretical heat transfer and psychrometric results based on industry-standard methods using ASHRAE® procedures (see Help system in software for detailed explanation of the Transfer Function Method (TFM) load calculation methodology used). The equipment or system type selected simply tells HAP to display specific sizing results required to select equipment of that general type or configuration.

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Interpreting High (Low) Peak Design Airflow Sizing Results for HVAC Equipment Selection

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For example, if you select a chilled water variable air volume (VAV) air handling unit (AHU) with reheat boxes HAP will show you the central VAV unit capacity as well as the zone terminal box sizing requirements. For a single-zone constant air volume (SZ CAV) RTU there are no zone terminal boxes so HAP configures the output reports to show only central component sizing results pertaining to that type of equipment.

If the sizing results from HAP do not match a particular type of equipment's allowable operating airflow range, typically 300-500 CFM/ton for most packaged RTUs, then you must analyze the resulting sensible and latent loads, temperatures and zone humidity level then formulate a strategy for controlling them.

To illustrate this process Figure 1 (below) indicates a design cooling load for a VAV system.

Central Cooling Coil Sizing Data

Total coil load	**27.7**	Tons	Load occurs at	**Jul 1600**	
Total coil load	**332.4**	MBH	OA DB / WB	**95.0 / 76.0**	°F
Sensible coil load	**242.5**	MBH	Entering DB / WB	**81.5 / 65.6**	°F
Coil CFM at Jul 1600	**7689**	CFM	Leaving DB / WB	**51.7 / 50.4**	°F
Max block CFM at Jul 1600	**7950**	CFM	Coil ADP	**48.4**	°F
Sum of peak zone CFM	**8073**	CFM	Bypass Factor	**0.100**	
Sensible heat ratio	**0.730**		Resulting RH	**41**	%
ft²/Ton	**346.6**		Design supply temp.	**55.0**	°F
BTU/(hr·ft²)	**34.6**		Zone T-stat Check	**8 of 9**	OK
Water flow @ 10.0 °F rise	**N/A**		Max zone temperature deviation	**0.1**	°F

Figure 1: Design Cooling Load

The design supply airflow (7,689 CFM) results in a relatively low airflow for the total coil load (27.7 Tons), computed as 7,689 / 27.7 = **278 CFM/Ton**. It will be difficult to find a packaged system that can deliver that low of supply airflow for the nominal unit capacity, likely a 30-Ton unit.

Prior to understanding why calculated airflow may be too high or too low for the nominal unit tonnage it is important that we briefly review some fundamental concepts of load estimation and explore the various input parameters that affect sizing results. Then we will illustrate how to interpret the calculation results and how to modify the inputs to the load estimating software to yield results that allow a reasonable HVAC unit selection that meet the design requirements.

Sensible Heat Ratio

The **Sensible Heat Ratio (SHR)** is the ratio of the central cooling coil's Sensible Load to its Total (Sensible + Latent) Load. Note in Figure 1 that the computed SHR = 0.730. The SHR for most manufacturers' packaged direct expansion (DX) RTUs is typically between 0.7-0.8 with 0.75 being a general rule-of-thumb. The lower the calculated SHR the more difficult it is to find packaged equipment that is capable of meeting the Latent (Total - Sensible) Load and the lower the required supply airflow (CFM/ton). As the SHR increases the Latent Load decreases (Sensible Load increases) and the resulting supply CFM/ton also increases. If the calculated SHR is between 0.70-0.80 the majority of the time the packaged equipment will be selectable within its allowable airflow operating range, again typically 300-500 CFM/ton.

How Does HAP Calculate the Required System Supply Air Quantity (CFM)?

Before system-level airflow can be determined the Sensible and Latent Loads must be determined at the space or zone level along with the required space and/or zone CFM. HAP utilizes the ASHRAE Transfer Function Method to compute design loads. HAP uses an 8-step load calculation process:

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1. Compute sensible and latent loads for all spaces in zones served by the HVAC system.
2. Sum space loads to obtain sensible and latent loads for all zones served by the HVAC system.
3. Determine required zone and space airflow rates.
4. Compute required sizes for zone equipment such as terminal reheat coils, supplemental zone heating units and fan powered mixing boxes, as necessary.
5. Determine required system airflow rates. This includes sizing all fans and outdoor ventilation airflow rates.
6. Simulate HVAC system operation for 289 design load hours (24 hr/design cooling day * 12 months plus 1 design heating hour). Based on the required airflow rates determined in steps 3-5, operation of the HVAC system is mathematically simulated to produce profiles of loads on the central cooling and heating coils.
7. Identify peak coil loads. Cooling and heating coil load profiles from step 6 are inspected to identify maximum loads.
8. Report results.

Required supply airflow rates for system terminals and fans are based on **worst-case** of cooling or heating loads. In most cases the cooling CFM will dominate, unless you are in an extremely cold climate. System airflow rates are then used to simulate system operation for both design cooling and design heating conditions to determine maximum coil loads. Once the system-level airflow (CFM) is calculated HAP considers three other user-inputs that may affect the final calculated supply airflow quantity:

1. Zone Minimum Supply Airflow Override
2. Direct Exhaust Airflow Override
3. Outdoor Ventilation Airflow Override

If any of those three override values exceed the calculated supply air quantity HAP will use the highest of the values found to establish the required supply air quantity.

At the space or zone level HAP uses the standard industry sensible heat equation, which has two unknown values, CFM and d-T:

$$\mathbf{SHC = 1.08 * CFM * d\text{-}T} \qquad \text{(Eq 1)}$$

where

- SHC = sensible heat capacity
- 1.08 is the air constant at sea level (actual value varies depending on site elevation)
- CFM = dehumidified supply air quantity, cu ft/min
- d-T = temp difference between zone (thermostat setpoint) and assumed supply air, deg-F.

HAP requires you to either assume a supply air temperature (SAT) or enter a known airflow quantity (CFM) under the central cooling coil input field. Since most of the time we do not know the CFM (that's what we want to be calculated) we start with a reasonable assumption of the SAT, such as 55-58F for a typical packaged cooling system, as shown in Figure 2.

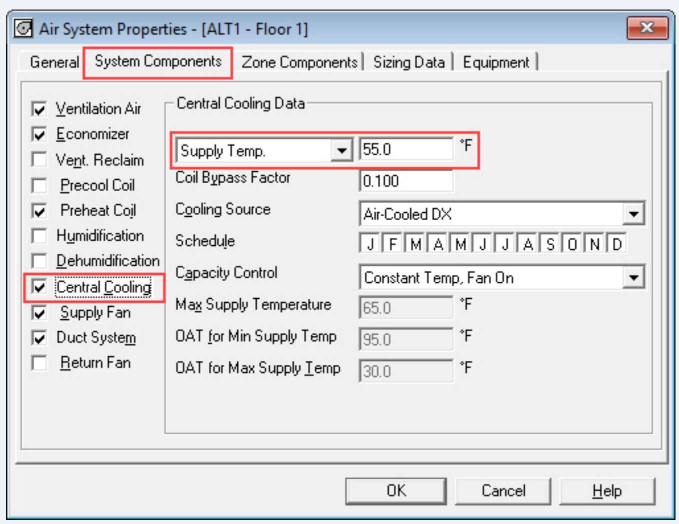

Figure 2: Cooling Coil Sizing Criteria

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It should be noted that SAT is **NOT** the leaving air temperature (LAT) off the cooling coil; rather it is the SAT entering the spaces.

With the assumed SAT, and since HAP also knows the room temperature from the thermostat setpoint you specify, it can then perform the stage 1 calculations. It does this by calculating the sensible and latent loads in the spaces then sums the sensible loads together at the zone level and solves for CFM based on a known d-T by rearranging the SHC equation to solve for CFM, as shown below:

$$\textbf{CFM = SHC / (1.08 * d-T)} \qquad \text{(Eq 2)}$$

For a constant volume system, HAP then determines the system level CFM by summing the zone CFMs.

Central Cooling Coil Sizing Data

Total coil load	27.7	Tons
Total coil load	332.4	MBH
Sensible coil load	242.5	MBH
Coil CFM at Jul 1600	7689	CFM
Max block CFM at Jul 1600	7950	CFM
Sum of peak zone CFM	8073	CFM
Sensible heat ratio	0.730	
ft²/Ton	346.6	
BTU/(hr-ft²)	34.6	
Water flow @ 10.0 °F rise	N/A	

Load occurs at	Jul 1600	
OA DB / WB	95.0 / 76.0	°F
Entering DB / WB	81.5 / 65.6	°F
Leaving DB / WB	51.7 / 50.4	°F
Coil ADP	48.4	°F
Bypass Factor	0.100	
Resulting RH	41	%
Design supply temp.	55.0	°F
Zone T-stat Check	8 of 9	OK
Max zone temperature deviation	0.1	°F

Figure 3: Cooling Coil Sizing Results, 55F SAT

For a variable volume system, HAP determines the peak coincident airflow at the VAV supply fan to determine system level CFM.

Modifying HAP Inputs to Facilitate Equipment Selection

To illustrate how to modify HAP inputs to facilitate equipment selection we will look at two examples; one with low (<300) CFM/ton and one with a high (>500) CFM/ton. We will then look at how you can modify the inputs to yield coil selection criteria that is more reasonable.

Example 1 (Low CFM/ton):

The initial load results from Figure 1 are shown below. As discussed previously the resulting supply airflow is 278 CFM/ton (7,689 / 27.7), well below the normal minimum allowable airflow of 300 CFM/ton.

The sensible heat factor is 0.73, which is within the normal range (0.7-0.8); however look at the resulting room relative humidity (RH) of 41%. This means the cooling coil is dehumidifying the zone below where it really needs to be. ASHRAE research has shown that microbial growth can occur in the building above 60% RH so you should aim for RH of 50%, ± 5%. So a 41% resulting RH means our assumed SAT of 55F is a bit too cold, so we should raise the assumed SAT slightly and recalculate the design load. Raising the SAT input from 55F to 57F:

Central Cooling Coil Sizing Data

Total coil load	27.4	Tons
Total coil load	329.3	MBH
Sensible coil load	245.3	MBH
Coil CFM at Jul 1600	8501	CFM
Max block CFM at Jul 1600	8834	CFM
Sum of peak zone CFM	8970	CFM
Sensible heat ratio	0.745	
ft²/Ton	349.9	
BTU/(hr-ft²)	34.3	
Water flow @ 10.0 °F rise	N/A	

Load occurs at	Jul 1600	
OA DB / WB	95.0 / 76.0	°F
Entering DB / WB	81.0 / 65.8	°F
Leaving DB / WB	53.7 / 52.4	°F
Coil ADP	50.7	°F
Bypass Factor	0.100	
Resulting RH	44	%
Design supply temp.	57.0	°F
Zone T-stat Check	9 of 9	OK
Max zone temperature deviation	0.0	°F

Figure 4: Revised Cooling Coil Sizing Results, 57F SAT

Notice the design supply air CFM increased significantly, from 7,689 to 8,501 CFM, and now yields 310 CFM/ton (8501 / 27.4), within the allowable selection range of most packaged equipment. Notice the Room RH has increased to 44% as well, which is acceptable.

Further increasing the SAT to a higher value, such as 58F, will increase the CFM/ton and zone RH, however we will conclude this example now since we achieved our objective of getting the supply airflow between 300 - 500 CFM/ton.

Example 2 (High CFM/ton):

Figure 5 below shows a design load result with the supply airflow of 510 CFM/ton (17,952 / 35.2).

When the resulting CFM/ton is very high, as in this case, most times this also means the calculated SHR is above 0.8, in this case nearly 0.9, so the resulting supply airflow is very high, higher than a typical package unit's allowable airflow operating range (300-500 CFM/ton). When the SHR is very high this often means the cooling load is dominated by internal heat gains which are mostly sensible load. Going back to our earlier Equation 1, there are two variables that affect the sensible load, CFM and d-T (or SAT), therefore to reduce the calculated CFM you have two options:

1. Reduce the assumed SAT (or)
2. Directly enter a reduced supply CFM and let HAP compute the required SAT.

Central Cooling Coil Sizing Data

Total coil load	35.2	Tons	Load occurs at	Aug 1600	
Total coil load	421.8	MBH	OA DB / WB	95.0 / 76.0	°F
Sensible coil load	378.3	MBH	Entering DB / WB	74.0 / 61.4	°F
Coil CFM at Aug 1600	17952	CFM	Leaving DB / WB	54.1 / 52.9	°F
Max block CFM at Aug 1600	18743	CFM	Coil ADP	51.9	°F
Sum of peak zone CFM	19063	CFM	Bypass Factor	0.100	
Sensible heat ratio	0.897		Resulting RH	43	%
ft²/Ton	273.1		Design supply temp.	58.0	°F
BTU/(hr·ft²)	43.9		Zone T-stat Check	3 of 3	OK
Water flow @ 10.0 °F rise	N/A		Max zone temperature deviation	0.0	°F

Figure 5 - Cooling Coil Sizing Results, 0.90 SHR, 58F SAT, 43% RH

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Keep in mind, however that there is a practical limitation as to how cold of SAT you can get from a typical packaged unit, generally about 54-55 F depending on airflow and entering coil conditions. In addition, note in Fig 5 the resulting RH is 43%, which is nearing the low end of the range so we don't have much room to reduce the SAT as this will only further depress the room RH, making the room too dry.

Let's try reducing the SAT input from 58F to 56F:

Central Cooling Coil Sizing Data

Total coil load	35.0	Tons	Load occurs at	Aug 1600	
Total coil load	420.4	MBH	OA DB / WB	95.0 / 76.0	°F
Sensible coil load	373.0	MBH	Entering DB / WB	73.8 / 60.4	°F
Coil CFM at Aug 1600	16157	CFM	Leaving DB / WB	52.0 / 50.8	°F
Max block CFM at Aug 1600	16770	CFM	Coil ADP	49.6	°F
Sum of peak zone CFM	17056	CFM	Bypass Factor	0.100	
Sensible heat ratio	0.887		Resulting RH	40	%
ft²/Ton	274.0		Design supply temp.	56.0	°F
BTU/(hr-ft²)	43.8		Zone T-stat Check	9 of 9	OK
Water flow @ 10.0 °F rise	N/A		Max zone temperature deviation	0.0	°F

Figure 6: Cooling Coil Sizing Results, 0.89 SHR, 56F SAT, 40% RH

Notice the design supply air CFM decreased significantly from 17,952 to 16,157 CFM and now yields 462 CFM/ton,) (16,157 / 35), easily within the allowable selection range of most packaged equipment. Notice the Room RH has decreased from 44% to 40% as well so this is likely the minimum allowable SAT to prevent making the room too dry and causing comfort problems for occupants.

The designer may want to consider the use of a humidifier in these sorts of conditions with little latent heat gains and low room RH.

So to conclude, HAP calculates the total sensible load then solves for the supply air CFM using an assumed SAT (or CFM) and the delta-T (thermostat cooling setpoint - SAT). Sometimes the sizing results are outside the allowable selection range of most packaged equipment. With some trial-and-error adjustment of the SAT or by entering the supply air CFM directly often times you may get your coil selection parameters within an acceptable range to select equipment.

A future EXchange newsletter article will discuss how to actually select equipment to meet design load results. Please contact your local Carrier sales engineer for assistance with equipment selection and application.

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Frequently Asked Questions

Question 1: I am designing a constant volume rooftop system. I specified the design supply air temperature as 55°F. On the Air System Sizing Summary report, in the Central Cooling Coil Sizing Data section I see the leaving dry-bulb temperature is 57.4°F. Why isn't the leaving temperature 55°F?

Answer: The "design supply temperature" you specified defines the temperature of air at the supply grille. The air temperature leaving the cooling coil can differ from this design value for two principal reasons which are explained below.

1. The "design supply temperature" is only required for times when the zone is at its peak sensible load. If the zone sensible load is not at its peak when the peak coil load occurs, then a warmer supply temperature will be sufficient to meet the zone load. The peak zone load may not coincide with the peak cooling coil load for two reasons:
   a. While the coil load is strongly affected by the zone load, it is also affected by other system load components such as the ventilation load, supply fan heat gain and plenum heat gain. If these loads peak at a different hour than the zone sensible load, then the overall coil peak may be at a different time. Therefore, when the coil load peaks, the zone load may be at less than 100% of its peak load. Consequently a warmer supply temperature is sufficient.
   b. Effects of the thermostat throttling range and zone dynamics can affect the zone load. The supply airflow rate is determined based on a load estimate assuming the equipment runs 24 hours a day and maintains the zone exactly at the cooling setpoint. These loads are then corrected for actual operating conditions (thermostat throttling range, nighttime shutdown or setup period) to simulate system operation and determine the coil loads. These adjustments alter the zone loads and can cause the load to be less than 100% of peak when the peak cooling coil load occurs. For example, if operating at 77°F zone temperature due to the thermostat throttling range instead of 75°F, the zone load will be slightly less (see discussion below for more details).
2. If you have a draw-thru coil/fan configuration or duct heat gain, the off coil temperature must be slightly colder than that required at the grille to overcome these heat gains. However that explains why the off coil temperature might be colder than design.

Typically both factors above are at work at the same time. The temperature required at the supply grille rises as the zone load drops below 100% of peak. At the same time, duct heat gain and fan heat (for a draw-thru unit) require a lower temperature. Often the net of these two factors results in the off coil temperature being higher than the design supply air temperature you specified.

It is also important to remember that the principle of constant volume system operation is that a constant volume of air is provided and the supply temperature is varied to meet the load. Because we are calculating loads according to 1-hour time steps, we are talking about the average supply temperature over one hour. With a DX unit the compressor is cycled or staged. The average hourly supply temperature varies according to the number of minutes the compressor is cycled on or off during the hour, or due to the compressor staging. For the minutes the compressor is on, air close to or below the design temperature is provided, and for the minutes the compressor is off neutral air is provided (for example at 75°F). In a chilled water unit, the water flow or water temperature to the coil is regulated to control the off coil temperature.

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Frequently Asked Questions

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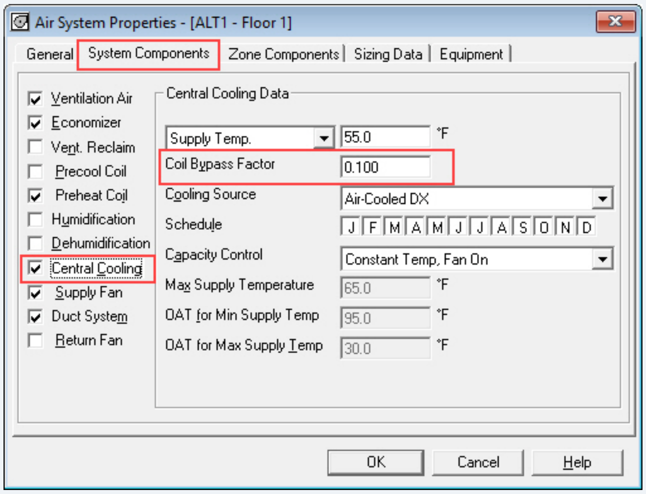
Further Information: Earlier it was mentioned that one reason the zone load was less than 100% of peak was the thermostat throttling range and zone dynamics. HAP uses a system-based design procedure utilizing the transfer function load calculation method. This approach requires that loads be calculated in two stages. In the first stage, loads are calculated assuming the equipment provides cooling 24 hours a day and maintains the zone at the cooling setpoint all the time. In the second stage, system operation is simulated to determine coil loads and in doing so the original loads are corrected for actual operating conditions. These conditions include the fact that the thermostat controls zone temperature within a throttling range rather than to a precise temperature, and that the equipment may not operate 24 hours a day, instead using a shutdown period or a setup period during unoccupied times. The corrections to the loads change the shape of the load profile. So where Stage 1 results yield a peak zone load at one time, the adjusted loads from Stage 2 can yield either a higher or lower zone load at the same time. To understand these corrections, review the Hourly Zone Design Day Loads report. It shows the "Zone Load" which is Stage 1 results and the "Zone Conditioning" which is Stage 2 results.

There are two schools of thought regarding this procedure. One group of users endorses this approach because it considers the real aspects of system control and building dynamics. Another group feels that design loads should be more idealized. For this second group we recommend using a thermostat schedule with all 24 hours in the occupied period and a throttling range of 0.1°F (or 0.1°C). These inputs will minimize the load corrections and dynamic effects so Stage 2 results will be nearly identical to Stage 1 results. Note that this still may result in differences between off-coil and design supply temperatures if the coil load and zone load peak at different times, or duct heat gain or a draw-thru unit configuration are used.

Question 2: On the System Components screen there is an input for "Coil Bypass Factor." What is that and is the default value of 0.100 OK to use?



Answer: The cooling coil Bypass Factor (BF) is a concept that Dr. Willis Carrier developed to indicate the effectiveness (efficiency) of a cooling coil to cool the air to the saturation condition. In other words for a "perfect" coil the BF = 0 and *all* the air passing through the cooling coil would come into direct contact with the coil surfaces and be fully dehumidified to the saturation condition. In the real world however, coils are not perfect and there are only so many rows and fins we can use before the coil is too deep and the airflow pressure drop is too high or the coil is physically too large to fit into the allotted space. Coil designers optimize the coil surface area at the lowest airside pressure drop in order to meet the required sensible and latent cooling loads given a particular space constraint inside the HVAC unit cabinet. The coil BF varies depending on four variables:

1. Number of rows in coil (depth)
2. Number of fins per inch or foot (fin spacing)
3. Velocity of air through coil (V = CFM/A); where A=coil face area and V=face velocity
4. Refrigerant (fluid) temperature inside coil tubes

So for a specific coil face area (A) the BF goes up as rows/fins decrease or as face velocity increases **or** as fluid temperature in coil increases. So a 4-row/10 fpi coil has a lower BF than a 3-row/10 fpi coil. Also if a particular coil has 1,000 CFM flowing through it and you reduce the airflow to 800 CFM the BF of that coil will decrease because the air velocity has decreased and the air has a longer time in contact with the cold coil surface. So the lower the BF the better for latent capacity and the higher the BF the more sensible (less latent) capacity the coil will remove.

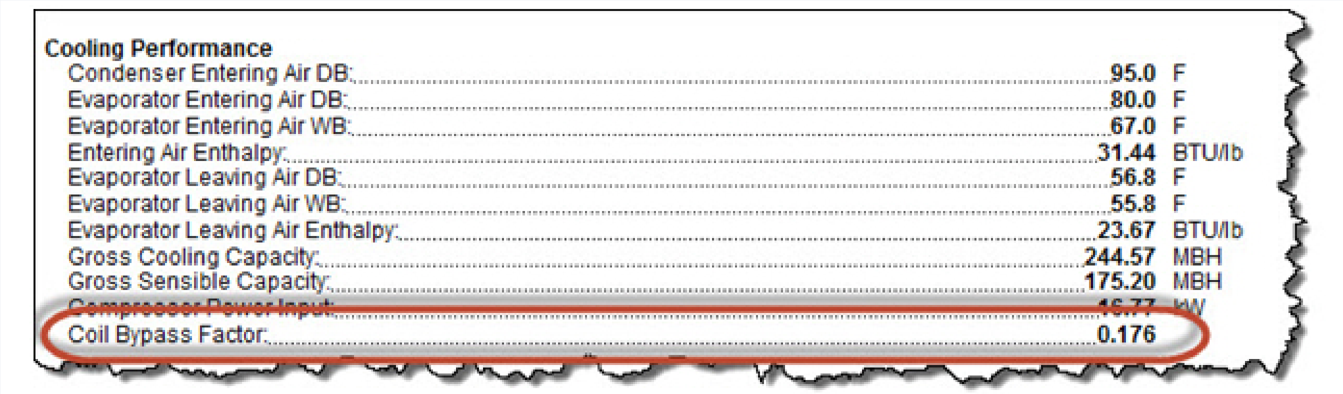
Most manufacturers of Packaged DX RTUs use either 3 or 4-row deep evaporator coils. Carrier is one of the only manufacturers that publishes the BF in equipment ratings. As a matter of fact the Carrier Electronic Catalog (E-CAT) equipment selection software displays the BF of the coil you select based on your particular selection criteria, as shown below: On the HAP "Air System Sizing Summary" report it lists the Coil Bypass Factor. Actually BF is a HAP user-input value that many people overlook and just accept the default value for. **You should not do so.** If you do not yet have an actual preliminary unit selection it is acceptable to temporarily assume the default 0.10 BF value, however after selecting a specific coil you should go back to the cooling coil setting in HAP and adjust this BF to match your actual unit selection. **Most people do not do this extra step but they should as BF can have a significant effect on the resulting loads and capacity of the equipment.**

	Rows	Fins per in (ft)	Velocity	Refrig (fluid) Temp
To Increase BF	↓	↓	↑	↑
To Decrease BF	↑	↑	↓	↓

Various types of equipment cooling coils have different bypass factors as shown below:

Equipment Type	Available Cooling Coil Rows	Bypass Factor (BF) Range
Residential Cased Coil	1-2	0.20-0.30
Small Packaged Unit (<10 tons)	2-4	0.05-0.30
Packaged RTU (> 10 tons)	3-4	0.03-0.20
Central Station AHU	3-10	0.002-0.12

Typical Bypass Factors (BF) for Various Equipment Types and Coil Rows



Cooling Performance

Condenser Entering Air DB:	95.0	F
Evaporator Entering Air DB:	80.0	F
Evaporator Entering Air WB:	67.0	F
Entering Air Enthalpy:	31.44	BTU/lb
Evaporator Leaving Air DB:	56.8	F
Evaporator Leaving Air WB:	55.8	F
Evaporator Leaving Air Enthalpy:	23.67	BTU/lb
Gross Cooling Capacity:	244.57	MBH
Gross Sensible Capacity:	175.20	MBH
Compressor Power Input:	16.77	kW
Coil Bypass Factor:	0.176	

Carrier E-CAT RTU Selection Example - Coiling Coil Bypass Factor

2017 Training Class Schedule

Location	**Load Calculation for Commercial Buildings** *System Design Load HAP*	**Energy Simulation for Commercial Buildings** *HAP*	**Energy Modeling for LEED® Energy & Atmosphere Credit 1** *HAP*	**Advanced Modeling Techniques for HVAC Systems** *HAP*	**Engineering Economic Analysis** *EEA*	**Block Load** *Block Load*
Loxley, AL	Jan 10	Jan 12	—	Jan 13	—	—
St. Louis, MO	—	—	Jan 17	Jan 20	—	—
Louisville, KY	Feb 14	Feb 15	—	—	—	—
Atlanta, GA	Feb 21	Feb 22	—	Feb 23	—	—
Toronto, ON	Apr 18	Apr 19	—	Apr 20	—	—
Washington, DC	May 9	May 10	—	May 11	—	—

Additional classes are being added.

eDesign Suite Software Current Versions (North America)

Program Name		Current Version	Functionality
HAP	Hourly Analysis Program (HAP)	v5.01	Peak load calculation, system design, whole building energy modeling, LEED® analysis
BSO	Building System Optimizer	v1.50	Rapid building energy modeling for schematic design
BLK	Block Load	v4.16	Peak load calculation, system design
EEA	Engineering Economic Analysis	v3.06	Lifecycle cost analysis
RPD	Refrigerant Piping Design	v4.00	Refrigerant line sizing
SDL	System Design Load	v5.00	Peak load calculation, system design

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