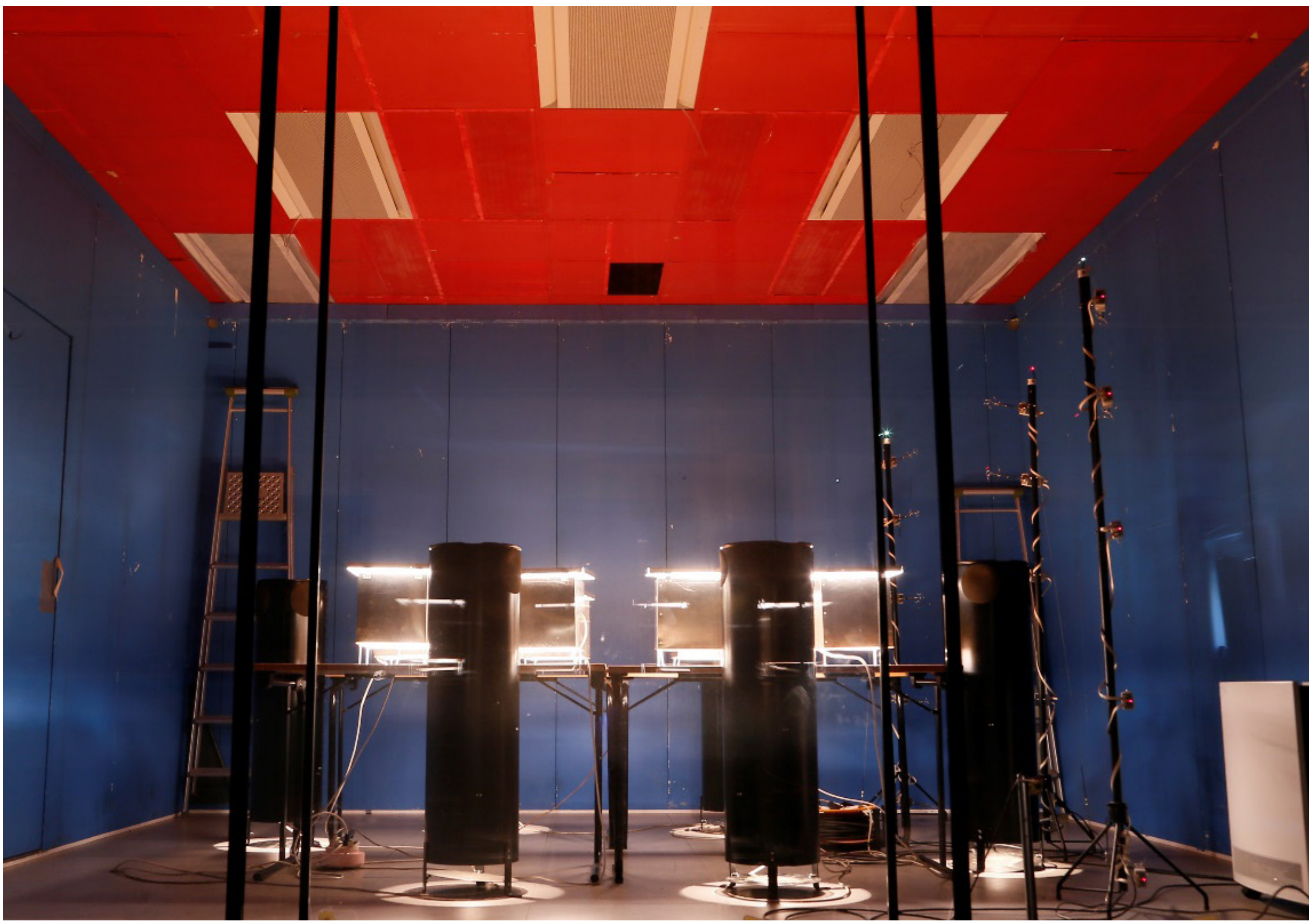
APPLICATION GUIDE

A2

Establishing Primary Airflow for Waterborne Climate Systems



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Contents

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Introduction

We refer to “climate beams” as Waterborne Climate Systems (WBCS) because they can both heat and cool the occupied space whereas a “chilled beam” systems sounds like it is cooling only. Waterborne Climate Systems are a relatively new HVAC solution in North America.

In comparison, Variable Air Volume (VAV) systems are very popular in North America. The consulting and construction industry is very comfortable using VAV in a wide variety of building types. In many applications the experience level is so high that the designer does not even think about why the design parameters are what there are, they know it will work.

Figure 1: Climate Beam

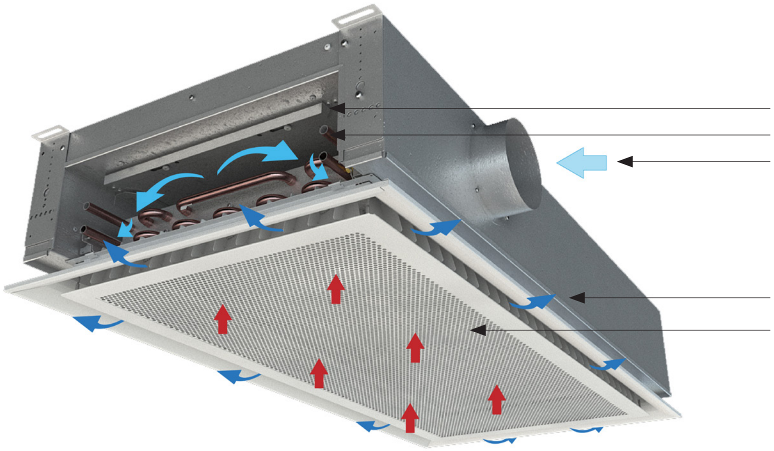
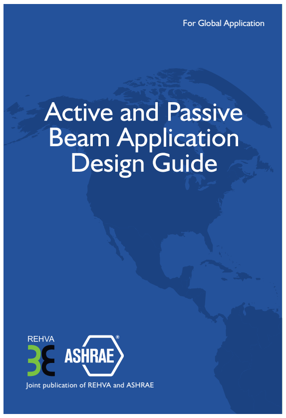


Figure 2: Climate Beam Design Guide



Fortunately WBCS systems have been in use in Europe for much longer. Swegon originated the active climate beam in the 1970s has been delivering over 400 WBCS systems a year.

This Guide will study how cooling loads impact WBCS system design, how to establish primary airflow to meet the design conditions, selection parameters of beams and the energy impact of the primary air system design. It will compare a WBCS system to a VAV system for reference. An additional reference is the Active and Passive Climate Beam Design Guide published by ASHRAE and REHVA.

Part 1: Zone Cooling Loads

As with all HVAC design it starts with the zone cooling loads. The key difference between a VAV and WBCS is the WBCS system is clearly broken down into the latent and sensible load in the zone and the cooling load due to the ventilation air requirement while the VAV system load calculations tend to blend the zone and ventilation loads together. In a WBCS system, the two loads are handled by two different pieces of equipment (the climate beams and the primary air handling unit) while in a VAV system the loads are blended and handled by the supply air unit. In this respect, cooling loads calculations for a WBCS system are more like fancoils, WSHPs or VRF systems.

Figure 3 shows how the loads need to be grouped for a WBCS design. The Zone sensible load is used to size the beams, while the other loads are used to size the primary air handling unit. It should be noted, that the loads are not different between HVAC systems (supply fan work aside) as they are a product of the building design and application.

Figure 3: Design Cooling Load Grouping

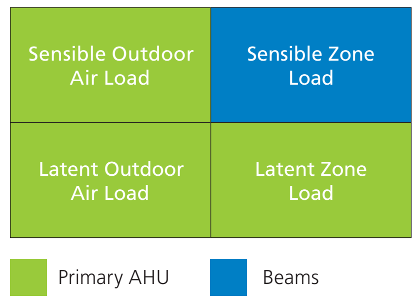


Table 1: Typical Classroom Zone Cooling Load

	Sensible			Latent			Total		
	Btu/h	Btu/h-ft²	%	Btu/h	Btu/h-ft²	%	Btu/h	Btu/h-ft²	%
Wall 1 load	1124	1	5%	0	0	0%	1124	1	4%
Glass 1 Conduction	109	0	0%	0	0	0%	109	0	0%
Glass 1 Solar	10143	10	43%	0	0	0%	10143	10	33%
Wall 2 load	0	0	0%	0	0	0%	0	0	0%
Glass 2 Conduction	0	0	0%	0	0	0%	0	0	0%
Glass 2 Solar	0	0	0%	0	0	0%	0	0	0%
Roof Load	0	0	0%	0	0	0%	0	0	0%
Light Load	1365	1	6%	0	0	0%	1365	1	4%
Plug Load	3412	3	14%	0	0	0%	3412	3	11%
Occupant Load	7424	7	31%	7424	7	100%	14848	15	48%
Infiltration Load	0	0	0%	0	0	0%	0	0	0%
Subtotal	23577	24	100%	7424	7	100%	31001	31	100%
Safety Factor	1179	1		371	0		1550	2	
Total	24756	25		7795	8		32551	33	

Table 1 shows the sensible cooling loads for a typical classroom with a south exposure in Chicago. The ventilation load is not included here as the load is managed by the primary air handling unit.

Table 2: Typical Office Zone Cooling Load

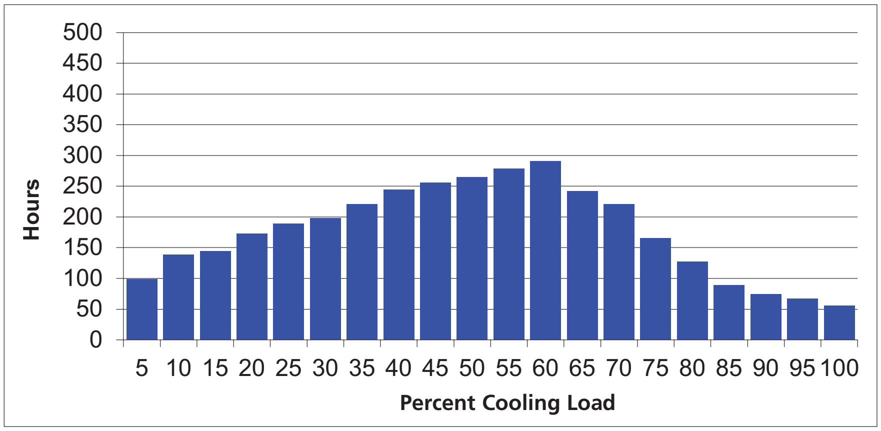
	Sensible			Latent			Total		
	Btu/h	Btu/h-ft²	%	Btu/h	Btu/h-ft²	%	Btu/h	Btu/h-ft²	%
Wall 1 load	1124	1	5%	0	0	0%	1124	1	5%
Glass 1 Conduction	109	0	0%	0	0	0%	109	0	0%
Glass 1 Solar	10143	10	46%	0	0	0%	10143	10	41%
Wall 2 load	0	0	0%	0	0	0%	0	0	0%
Glass 2 Conduction	0	0	0%	0	0	0%	0	0	0%
Glass 2 Solar	0	0	0%	0	0	0%	0	0	0%
RoofLoad	0	0	0%	0	0	0%	0	0	0%
Light Load	1365	1	6%	0	0	0%	1365	1	6%
Plug Load	6824	7	31%	0	0	0%	6824	7	28%
Occupant Load	2450	2	11%	2450	2	100%	4900	5	20%
Infiltration Load	0	0	0%	0	0	0%	0	0	0%
Subtotal	22015	22	100%	2450	2	100%	24465	24	100%
Safety Factor	1101	1		123	0		1223	1	
Total	23116	23		2573	3		25688	26	

Table 2 shows the sensible cooling loads for a typical office space with a southern exposure in Chicago. The loads are the same for both VAV and WBCS systems. The total classroom and office loads are 32.6 and 25.7 Btu/h·ft² respectfully. For interior spaces (no wall loads) the loads become 20.6 and 13.8 Btu/h·ft² respectfully. Note the classic "400 ft² per ton" works out to 30 Btu/h·ft² which is a blend of interior and exterior zones plus outdoor air load.

Impact of Oversized Cooling Loads

No one wants to undersize the cooling loads and end up with a building that cannot meet the cooling requirements however oversizing any HVAC system is also problematic. As a reminder, using ASHRAE 1% weather data means that (statically) the weather conditions will only be exceeded 1% of the time (87.6 hours) annually. When real world usage and weather are factored in, most buildings operate at 50% capacity most of the time (see Figure 4).

Figure 4: Typical Cooling Load Profile for Office Building in New York City



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All HVAC systems must provide some form of turn down (ability to operate at part load). Indeed this is often the capability that separates the marginal HVAC systems from the excellent systems. By overestimating the cooling loads and selecting an oversized HVAC system, we are forcing the system to operate at the bottom of its range for the bulk of the time.

Consider a constant volume all air system (CAV) system. The constant volume system meets part load conditions by raising the discharge air temperature. In short the system capacity is proportional to the difference between the AHU discharge air temperature and the space temperature. This is a basic packaged rooftop unit. The challenge with this system is that as the supply air temperature is raised, dehumidification is lost. The solution to the dehumidification issue is to introduce reheat.

Oversizing a CAV system results in the system raising the supply air temperature to meet the real world cooling load. The result of oversizing a CAV system is it may deliver little or no latent cooling leading to occupant dissatisfaction. (see Figure 5)

A Variable Air Volume (VAV) system varies the airflow in proportion to the cooling load. As the cooling load drops, the amount of supply air is reduced. A VAV system capacity is proportional to the supply air flow rate. By maintaining a constant discharge air temperature, a VAV system will implicitly manage dehumidification. From a comfort (and energy) point of view, a VAV system is superior to a constant volume system. (See Figure 6)

A VAV system can realistically turn down to about 30% of design capacity from an energy balance point of view. Below this point, issues around laminar flow over coils, fan stability and VAV box stability start to occur. A bigger issue is around comfort. Diffuser performance is often overlooked and at low airflows, the diffuser may not function properly which will lead to occupant dissatisfaction. A common example is the diffuser discharge velocity drops to the point where the coanda effect fails and the diffuser "dumps" cold air on the occupant. Without special attention the limit on most diffuser turndown is about 30 to 50%.

VAV systems are common, so why are there not more complaints about poor comfort? First, comfort is the number one complaint by building occupants. Second, the reheat starts and the comfort issue is resolved but the operating cost increases due to simultaneous heating and cooling. This is the reason that recent versions of ASHRAE Std 90.1 include considerable detail on minimum turndown (30%) of VAV systems and reheat control algorithms.

The temperature range used in a typical VAV system is 75 °F space temperature – 55 °F supply air temperature = 20 °F. For the example loads given above;

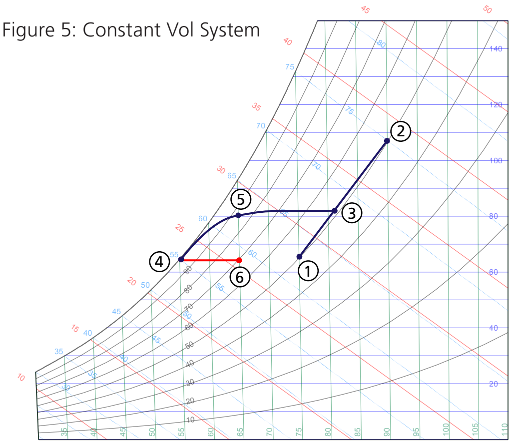
Classroom

Cfm/ft² = 24.8 Btu/h-ft²/(1.085 × 20 °F) = 1.14 cfm/ft²

Office

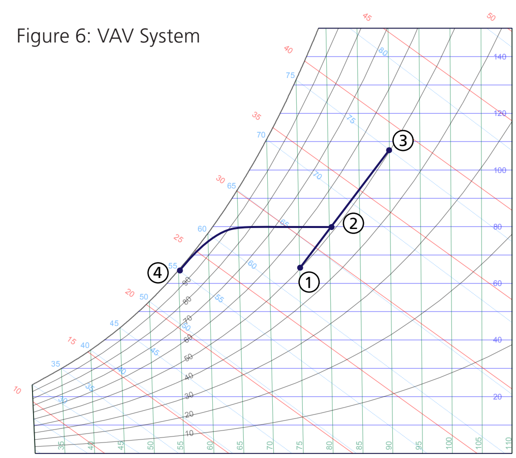
Cfm/ft² = 23.1 Btu/h-ft²/(1.085 × 20 °F) = 1.06 cfm/ft²

Figure 5: Constant Vol System



1. Space condition 75 ˚F DB, 50% RH
2. Outdoor Air 90 ˚F DB, 75 ˚F WB
3. Mixed Air 80 ˚F DB, 67 ˚F WB
4. Design Load Supply Air 55 ˚F DB, 54.5 ˚F WB
5. 50% Capacity Supply Air, 65 ˚F DB No Latent Cooling
6. 50% Capacity with Reheat Supply Air 65 ˚F DB, 65 ˚gr/lb

Figure 6: VAV System



1. Space condition 75 ˚F DB, 50% RH
2. Outdoor Air 90 ˚F DB, 75 ˚F WB
3. Mixed Air 80 ˚F DB, 67 ˚F WB
4. Design Load Supply Air 55 ˚F DB, 54.5 ˚F WB

If the design cooling loads are overstated by 20%, the new design airflow rates become 1.37 cfm/ft² and 1.27 cfm/ft² respectfully. At the high airflow rates it is extremely difficult to avoid drafting and thus comfort issues. From a practical point of view, the real, lower zone loads will cause the VAV system to reduce airflow until an energy balance is reached. However, the diffusers were sized for the design airflow and will struggle to maintain proper mixing. This will likely result in dumping. If reheat is available, it can be used to raise the minimum airflow and air temperature to maintain acceptable comfort but at a high energy cost.

WBCS systems have the advantage that the latent load is managed by a dedicated outdoor system (DOAS). The climate beams themselves only perform sensible cooling. A basic WBCS system is effectively a constant volume, variable temperature system like a fancoil. As the cooling load in the zone drops, the supply air temperature is raised by modulating the chilled water in the cooling coil. The advantage a climate beam has over a fancoil is there are no moving parts to circulate the air.

A WBCS system uses smaller temperature ranges. The primary air temperature difference can be the same as a VAV system (20 °F) but the beam temperature difference is closer to 10 °F. The actual airflow rate will depend on the induction ratio (the ratio of induced air to primary air). In a properly sized office system, the primary air provides 1/3 and the beam coil about 2/3 of the cooling. (See Figure 8)

When a WBCS system is oversized, the primary airflow rate increases and the beam induces too much local airflow. Comfort is hard to maintain due to drafting (too much airflow). In reality, the oversized system will respond to the actual load by closing off water flow to the beam. If the system is too oversized the primary airflow may overcool the space even with the beam water flow off particularly at part load. Drafts will still be an issue as the induction is driven by the primary airflow rate. It also increases the operating cost as the primary airflow is a more expensive cooling source than chilling with the beam.

No HVAC system is optimized if it is oversized. The penalties to comfort and operating cost can be severe. The best designs start with accurate loads.

Figure 7: Typical WBCS System

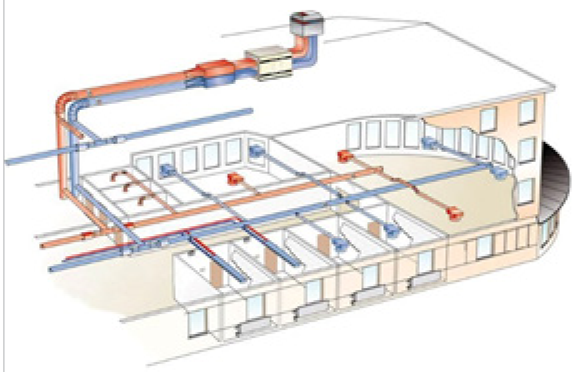
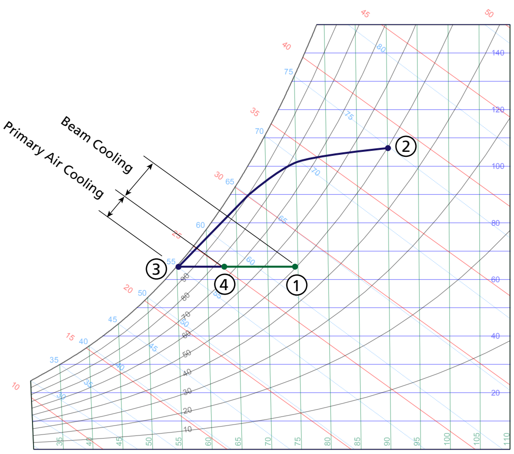


Figure 8: WBCS System



1. Space condition 75 °F DB, 50% RH
2. Outdoor Air 90 °F DB, 75 °F WB
3. Primary Air 54 °F DB, 53.5 °F WB
4. Induced Air 63 °F DB

Part 2: Establishing Supply Air Flow Rate

This section discusses how supply air flow rates for Variable Air Volume (VAV) systems and Primary airflow rates for WBCS systems are established. Both HVAC systems will be applied to the school and office example introduced in the previous section.

Variable Air Volume (VAV) System

The design airflow rate for a cooling zone will depend;

- On the minimum ventilation rate to provide acceptable indoor air quality.
- On the temperature range between the supply air and the space design condition and the zone sensible cooling load.

Temperature Range

Assuming the design space sensible temperature is 75 °F and the supply air temperature is 55 °F the temperature range is 20 °F. The actual supply air temperature will depend on several factors. First is the latent load. The supply air must be "dry" enough to meet the zone latent load. The larger the zone latent load, the lower the humidity ratio needs to be in the supply air. This means a lower saturated temperature leaving the cooling coil.

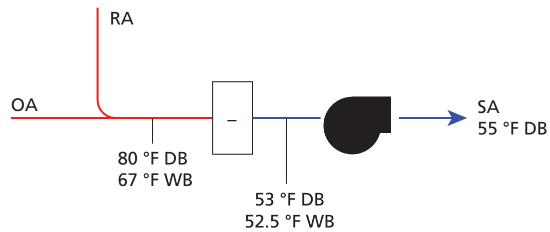
Referring to the examples in Table 1 and 2, the latent load in the class room in a much larger part of the class room load. Said another way, the office has a sensible heat ratio (SHR) (sensible load/total load) of 90% while the class room has a SHR of 76%.

Selecting the correct leaving coil temperature is not an easy calculation. Lowering the temperature to achieve dehumidification increases the temperature range and thus requires less supply air to meet the sensible load. This calculation requires iteration. In the past, it was often performed by graphing the loads on a psychrometric chart. Today, software load programs perform the calculation.

Another issue is whether the air handling unit is draw through or blow through. Regardless of arrangement, the supply fan motor heat will raise the supply air temperature 2 to 3 °F. If the fans are blow through the fan heat is added prior to the coil and is removed by the coil so the coil leaving temperature is the supply air temperature. If the fans are draw through, the fan heat will raise the supply air temperature reducing the temperature range available for zone sensible cooling.

This is an excellent example of how comfortable North Americans are with VAV systems. Most systems are designed with 55 °F supply air with a draw through fan arrangement. Assuming a 2 °F rise across the fan, the coil leaving air temperature is actually 53 °F. The lower humidity ratio is enough to meet most office zone latent loads. Unless there is something unusual occurring, most designers will not focus on these calculations.

Figure 9: Drawthrough VAV System



To calculate the fan heat temperature rise;
Obtain the fan bhp and convert to Btu/h
ΔT = motor bhp in Btu/h divided by (1.085 × cfm).

VAV Ventilation Rate

The ventilation rate is based on ASHRAE Std 62.1 multi-zone recirculating systems. This is a weighted average of multiple zone requirements taking into account that the supply air includes recirculated air. This will generally require a higher ventilation flow rate than for single zone systems.

$V_{ou} = D\Sigma_{\text{all zones}} (R_p \times P_z) + \Sigma_{\text{all zones}} (R_a \times A_z)$

$V_{ot} = V_{ou}/E_z$

Where

V_{bz} = outdoor airflow rate in the breathing zone (cfm)

V_{ot} = outdoor air intake (cfm)

R_p = People outdoor air rate (cfm/person)

P_z = number of people

R_a = Area outdoor air rate (cfm/ft²)

A_z = Area (ft²)

D = Occupant Diversity

V_{oz} = Zone airflow rate (cfm)

E_z = Zone air distribution effectiveness

The VAV supply air rate must be greater that the outdoor ventilation rate required to achieve acceptable indoor air quality. Using the classroom example, the ventilation rate is 0.42 cfm/ft² while the design airflow rate is 1.14 cfm/ft². The ventilation air rate is 37% of the total design airflow. In the case of the office example the ventilation rate is 0.11 cfm/ft² while the design airflow rate is 1.06 cfm/ft². The ventilation air rate is 10% of the total design airflow.

In most cases, the designer would not even check this knowing from experience with VAV systems that the ventilation flow rate will not be close to the required cooling air flow rate. However, in healthcare applications, the ventilation rates can exceed the flowrates required to just cool the space.

WBCS System

The first time someone designs a WBCS system, the ventilation rate can be complex, but only because the experience is not yet there. The following process will cover the steps required to establish an overall primary airflow and primary air condition to meet the building needs.

For a WBCS the air delivered to the climate beams is referred to as primary air. As mentioned before, it is helpful to consider a climate beam like a fancoil. The climate beam has a cooling coil to cool (sensibly only) local recirculated air. While the fan coil has a fan and motor to move the air, a climate beam uses the energy in the form of primary flow rate (cfm) and pressure (the static pressure drop across the nozzle bank). There are no moving parts to service and the sound levels are lower.

Figure 10: ASHRAE Std 62

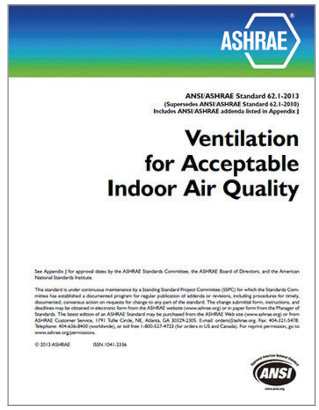
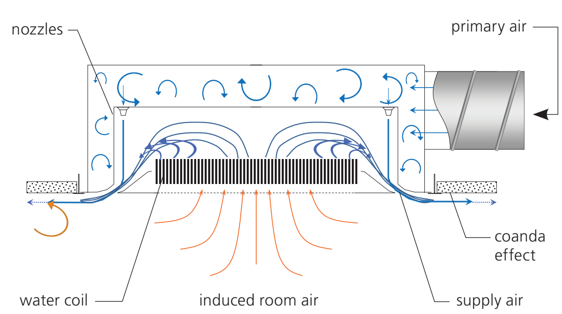


Figure 11: Climate Beam Operation



The primary airflow rate must be the larger of;

- The air flow rate to meet the ventilation rate required to deliver acceptable indoor air quality. This is the same as with VAV (or any other system for that matter) systems.
- The airflow rate to provide latent cooling in the zone. The requirement is the same as with VAV, the calculation is different.
- The airflow rate required to assist in meeting the zone sensible cooling rate. The requirement is the same as with VAV, the calculation is different.

Ventilation Rate

The ventilation rate is based on ASHRAE Std 62.1 single zone systems. This is the most straightforward method in ASHRAE Std 62.

$V_{bz} = R_p \times P_z + R_a \times A_z$

$V_{oz} = V_{bz}/E_z$

$V_{ot} = \Sigma_{\text{all zones}} V_{oz}$ (For 100% outdoor air systems like WBCS)

Where

V_{bz} = outdoor airflow rate in the breathing zone (cfm)

R_p = People outdoor air rate (cfm/person)

P_z = number of people

R_a = Area outdoor air rate (cfm/ft²)

A_z = Area (ft²)

V_{oz} = Zone airflow rate (cfm)

E_z = Zone air distribution effectiveness (E_z = 1.0 for ceiling climate beams)

Higher induction ratios mean the sensible cooling load can be met with a smaller amount of primary air but the fan total static pressure will increase due to increased pressue drop across the nozzles. A good rule of thumb for a typical office building is an induction ratio of 3 to 4. Schools will be closer to 1.

Latent Cooling Rate

Unlike VAV or other HVAC systems, all the latent cooling must be done in the primary air handling unit. Starting with a space design condition of 75 °F, 50% RH means the space humidity ratio is 64.6 gr/lb and the dewpoint is 55.2 °F. The primary air humidity ratio is calculated as follows;

$W_{\text{primary air}} = W_r - (P_{\text{latent}})/(0.68 * Q_p))$

where

$W_{\text{primary air}}$ = the humidity ratio of primary air in gr/lb

W_r = the humidity ratio of the design space condition in gr/lb

P_{latent} = the latent load in Btu/h

Q_p = the primary air flow in cfm

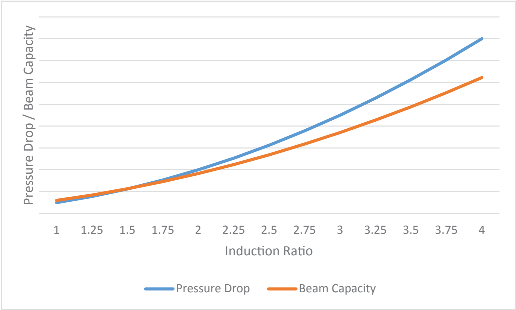
Just like a VAV system this requires iteration. As $W_{\text{primary air}}$ is decreased, the primary airflow rate will be reduced and the calculation needs to be repeated.

It is common in WBCS systems for the chilled water temperature serving the beams to be 2 to 3 °F warmer that the space dewpoint. Assuming a space dewpoint of 55 °F based on 75 °F DB and 50% RH, the supply chilled water temperature is 57 °F.

Sensible Cooling Rate

The zone sensible cooling rate is met by a combination of the cooling capacity of the primary air and the cooling capacity of the climate beam. There is a relationship between the amount of primary air and the amount of induced air (the local zone recirculated air). As mentioned earlier the recirculated air is “powered” by the volume flow rate and pressure of the primary air.

Figure 12: Capacity and Pressure Drop vs. Induction Ratio



Induction Ratio is the ratio of primary air to induced air. For example an induction ratio of 4 means that the air coming out of the climate beam is 1 part primary air and 4 parts induced (re-circulated air). The induction ratio is a feature of the actual product. Figure 12 shows the pressure drop vs. induction rate. Higher induction ratios require smaller nozzles and hence larger pressure drops. This is a design choice. Higher induction ratios mean the sensible cooling load can be met with a smaller amount of primary air but the fan total static pressure will go up due to the increased pressure drop.

The sensible cooling capacity is a weighted average calculation as follows;

$$P_{sensible} = 1.085 \times Q_p \times ((T_r - T_p) + IR \times (T_r - T_a))$$

Where

$P_{sensible}$ = the sensible cooling capacity of both the primary air and induced air in Btu/h

Q_p = the primary air flow rate in cfm

T_r = the sensible temperature of the design space condition in °F.

T_p = the primary air temperature in °F.

IR = induction ratio.

T_a = the induced air temperature in °F.

Solving this equation is an iterative process. As the values are changed, the calculation needs to be repeated. Since the parameters in this equation overlap with the latent calculation, changes here will require repeating the latent calculation.

While solving this set of equations seems daunting, this is where experience comes in.

1. To keep the air pressure drop in the range 0.3 to 0.6 in. w.c. an induction ratio of 3 to 4 is a good starting point for high SHR applications like offices. Schools will be closer to an induction ratio of 1.
2. With a typical space dewpoint of 55 °F the supply chilled water temperature should be 57 °F.
3. With 57 °F chilled water, the induced air supply temperature will be around 60–65 °F. This will depend on the actual climate beam selection.
4. Reducing the primary airflow rate will generally reduce the HVAC system energy usage. It also means the induction ratio needs to increase and hence the air pressure drop. Using an induction ratio of 3 to 4 for an office application is going to result in a primary airflow rate around 0.4 cfm/ft² which is good place to start.
5. The space latent load is the same for a VAV system as it is for a WBCS system (latent load has nothing to do with HVAC system choice). The design airflow rate for a VAV system is typically closer to 1 – 1.2 cfm/ft² while for a WBCS system the primary airflow rate is 0.4 cfm/ft². If a VAV system design is based on 55 °F leaving the coil (62.4 gr/lb) and the space is 64.6 gr/lb then the humidity ratio difference is 2.2 gr/lb. Assuming only 40% of the primary air flow for a WBCS system, the humidity difference should be in the range of 5.8 gr/lb. This can be achieved with 54–53 °F leaving the coil.

WBCS Primary Air Calculation Example – Office Space

Using the office example, establish a primary airflow rate and primary air conditions.

The approach will be to calculate the required airflow rates for ventilation, latent and sensible cooling and using the given rules of thumb establish the flow rate and air properties.

Ventilation Rate

Follow ASHRAE 62 for a single zone system with 100% outdoor air.

Occupancy (NP) = 1

Area = 1000 ft²

Occupant Density = 100 ft²/person

R_p = 5 cfm/person

R_a = 0.06 cfm/ft²

Number of people = 1000 ft²/100 ft²/person = 10

Ventilation rate = 10 × 5 cfm + 0.06 × 1000 ft² = 110 cfm

Latent Rate

The space humidity ratio is known (64.6 gr/lb) but the primary air humidity ratio is to be determined. The lower the primary air humidity ratio, the less primary air required. The red line in Figure 13 shows the humidity ratio for 75 °F DB, 50% RH space design condition. The Blue curve shows the humidity ratio of the primary air for varying off cooling coil conditions.

For office buildings and other high sensible heat ratio spaces, assume a delta of 8 gr/lb. Starting with 65 gr/lb the primary air should be about 57 gr/lb. Using Figure 13, the primary air will need to be cooled to 52.5 °F off the coil.

$Q_p = P_{latent} /(0.68 \times (W_r - W_{primary\ air}))$

Latent load Platent = 2573 Btu/h

w_r = 65 gr/lb

$w_{primary\ air}$ = 57 gr/lb

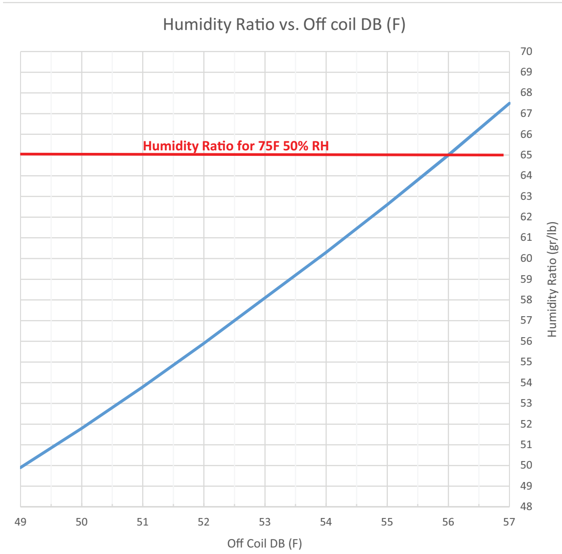
Primary air flow for latent load = 2573 / (0.68 × (65 – 57) = 473 cfm

Primary Cooling Rate

The space condition is known but the primary air conditions, induction ratio and induced air leaving temperature are not known.

Use the primary air conditions from the latent calculations as a starting point. The primary air temperature should not be more than 2 to 3 °F lower than space dew point. If the primary air is cooled more than this to lower the humidity ratio, then reheat may be required. This adds cost so it try to avoid reheat by raising the primary air temperature and allowing higher primary air flow rates. Using Figure 13, the off coil temperature for latent cooling needs to be 52.5 °F. Assuming 1°F reheat from fans or duct heat the supply air temperature to the

Figure 13: Humidity Ratio vs. Off Coil DB



beams will be within an acceptable range from the space dew point.

As a starting point assume an induction ratio of 4 and 64 °F induced air temperature.

$Q_p = P_{sensible} / (1.085 \times ((T_r - T_p) + IR \times (T_r - T_a))$

$P_{sensible}$ = 23116 Btu/h

T_r = 75 °F

T_p = 53.5 °F based on 52.5 °F plus 1 °F for duct heat gain

IR = 4

T_a = 64 °F

Primary airflow to meet sensible load = 23116 /(1.085 × ((75 – 52.5) + 4 × (75 – 64)) = 333 cfm

Table 3: Summary of Primary Airflow Requirements for Office Example

	cfm	cfm/ft²
Design Sensible Rate	333	0.33
Design Vent. Rate	110	0.11
Design Latent Rate	461	0.46

Table 3 shows the summary of the calculations. In this case, the latent airflow requirement is dominant so the primary airflow rate should be 461 cfm or 0.46 cfm/ft². The actual airflows to the specific climate beams will vary based on the exact model but the overall primary system can be designed.

it is worth noting that most real world climate beam office applications use 55 °F primary air. With duct losses, the off cooling coil condition is 54 °F or about 60 gr/lb. The primary air is not dry enough to meet the design latent load. Practically what happens is the space relative humidity climbs above 50% but is still well within std 55 requirements.

Optional Primary Air Calculation Example – Office Space

The previous example followed all the steps to illustrate the calculations and relationships. However, office design is relatively consistent so with practice the following approach can be made;

For cost reasons, try to avoid reheat in the primary air system. This limits the off coil condition in the DOAS unit and hence the humidity ratio to 52.5 to 54°F and 57 to 60 gr/lb respectfully.

Try to maintain the primary airflow rate for offices between 0.40 and 0.60 cfm/ft² for an energy efficient design.

Using the previous office example;

Zone sensible load is 23,116 Btu/h or 23.1 Btu/h-ft²

Zone latent load is 2,573 Btu/h or 2.6 Btu/h-ft²

Assume the primary airflow will pick up 1/3 of the total load

Primary air flow cooling capacity = 1/3 x 23116 Btu/h = 7700 Btu/h or 7.7 Btu/h-ft²

Given the space condition is 75 °F, 50% RH and 65 gr/lb, try 5 gr/lb delta so the primary air humidity ratio needs to be 60 gr/lb. Use figure 13 to pick leaving coil dry bulb temperature of 54 °F.

Assume 1 °F of duct heat gain so the primary air temperature at the beam is 55°F.

Assume 0.4 cfm/ft² of primary air and calculate the latent cooling capacity of the primary air;

The Latent cooling capacity is;

0.68 × 0.4 cfm/ft² × 5 gr/lb = 1.36 Btu/h- ft²

If the primary airflow latent capacity equals or exceeds the design latent load – stop. If not, you must increase the primary flow, lower the primary air dew point or a combination of both.

In this case, our first guess did NOT provide enough latent cooling so both the airflow and the primary air humidity ratio will need to be changed

Try 8 gr/lb delta so the primary air needs to be 58 gr/lb and increase primary airflow by 10 to 15% to 0.46 cfm/ft².

Latent Capacity

$0.68 \times 0.46\ \text{cfm/ft}^2 \times 8\ \text{gr/lb} = 2.5\ \text{Btu/h-ft}^2$

The zone latent load is 2.6 Btu/h-ft² so the lower primary air humidity ratio and increased primary airflow has basically met the zone load.

Check primary air sensible cooling capacity. Using an 8 gr/lb delta and Figure 13, the off coil drybulb temperature is 52.5 °F. Assume 1 °F of duct heat gain so the primary air temperature at the beam is 53.5°F.

$1.085 \times 0.46\ \text{cfm/ft}^2 \times (75-53.5) = 10.7\ \text{Btu/h-ft}^2$

The cooling capacity of the primary air is now more than 1/3 of the zone sensible cooling load so it is safe to assume that a climate beam can be selected that can meet the load. Since the primary air is carrying more than 1/3 of the zone sensible load, the induction ratio at the beams will be reduced (through the beam selection process) so the combination of beam coil capacity and primary air capacity will meet the zone sensible load.

As a final check, make sure that 0.46 cfm/ft² will meet or exceed the IAQ ventilation rate.

Use these design conditions and let the climate beam manufacturer supply specific beam performance schedule based on;

- 53.5°F primary air
- Approximately 0.46 cfm/ft² primary air
- 57 °F chilled water supply air temperature

The DOAS unit should be designed to achieve:

- 53.5°F dry bulb, 57 gr /lb primary air
- No reheat
- 100% outdoor air
- Total energy recovery device to minimize energy cost

Primary Air Calculation Example – Classroom

Whereas the previous example had a very high sensible heat ratio and is an excellent choice for WBCS systems, classrooms have a much higher latent load and require some additional care. Using the classroom example above;

Ventilation Rate

Occupancy (NP) = 30

Area = 1000 ft²

R_p = 10 cfm/person

R_a = 0.12 cfm/ft²

Ventilation rate = 30 × 10 cfm + 0.12 × 1000 ft² = 423 cfm

Latent Rate

The latent load in a classroom will dominant the primary airflow sizing. The change in humidity ratio will likely be 20 gr/lb or more.

$Q_p = P_{latent}/(0.68 \times (Wr - W_{primary\ air}))$
P_{latent} = 7795 Btu/h
w_r = 65 gr/lb
$w_{primary\ air}$ = 49.7 gr/lb (note: this is based on 49 °F leaving coil temperature)
Primary air flow for latent load = 7795 / (0.68 × (65 – 49.7) = 749 cfm

Primary Cooling Rate

In the office example, we used the primary conditions for latent cooling without reheat. However, for the classroom the latent primary air condition is too cool to use. While the supply ductwork can be insulated, the pressure box on the climate beam is not insulated. If the primary air is more that 2 to 3 °F cooler than the space dewpoint condensation may occur on the climate beam. For the classroom example, we will reheat the primary air to 54 °F to avoid the risk of condensation.

The large amount of primary air to meet the latent load means it will carry a larger portion of the zone sensible load so less air needs to be induced. For school applications an induction ratio of 1 is a good place to start.

$Q_p = P_{sensible} / (1.085 \times ((T_r - T_p) + IR \times (T_r - T_a))$
$P_{sensible}$ = 24756 Btu/h
T_r = 75 °F
T_p = 54 °F
IR = 1
T_a = 63 °F
Primary airflow to meet sensible load = 24756/(1.085 × ((75 – 54) + 1 × (75 – 66)) = 761 cfm

Table 4: Summary of Primary Airflows for Classroom

	cfm	cfm/ft²
Design Sensible Rate	761	0.76
Design Vent. Rate	423	0.42
Design Latent Rate	749	0.75

Table 4 shows the summary of the calculations. In this case, the latent airflow requirement is dominant (the sensible load is slighter higher but this can easily be reduced by choosing a beam with a slighter higher induction ratio) so the primary airflow rate should be 749 cfm or 0.75 cfm/ft ². The airflow per ft² is 39% higher for a school that the office even though the load is only 21% more. It is all to do with the latent load. The primary air flow will deliver more than half the required zone sensible cooling. The rest of the zone sensible cooling will be met by the induced air through the beam.

For this school example, the DOAS unit will need reheat. The primary air needs to be cooled to 49 °F and then reheated to 54 °F to lower the humidity ratio and avoid condensation in the ductwork and climate beams.

It is important to note that for classroom application, the required induction ratio is only 1. This is not common for climate beams (most beams are designed for higher induction ratios) so care should be taken to use a climate beam with a low induction ratio.

What can happen is the designer asks the climate beam supplier to select beams for a classroom that will receive 750 cfm primary air at 54 °F. The beam supplier realizes a standard climate beam can accept around 100 cfm of primary air so they recommend 8 climate beams to handle the high primary airflow. However, the standard climate beam has an induction ratio close to 4 so it has a cooling capacity close to 5,000 Btu/h. Eight beams × 5,000 Btu/h = 40,000 Btu/h sensible cooling capacity which far exceeds the required design zone sensible load. Drafting will likely be an issue and the climate beams will short cycle (the water cooling valve will rapidly open and close) leading to temperature control issues. It is critical that the climate beams be properly designed for the application.

Use these design conditions and let the climate beam manufacturer supply specific beam performance schedule based on:

- 54 °F primary air
- Approximately 750 cfm/classroom primary air (0.75 cfm/ft²)
- Induction ratio close to 1
- 57 °F chilled water supply air temperature
- 24.8 kBtu/h zone sensible cooling load
- NC < 20
- Acceptable air distribution to avoid drafts
- Acceptable air distribution in heating mode (if required)

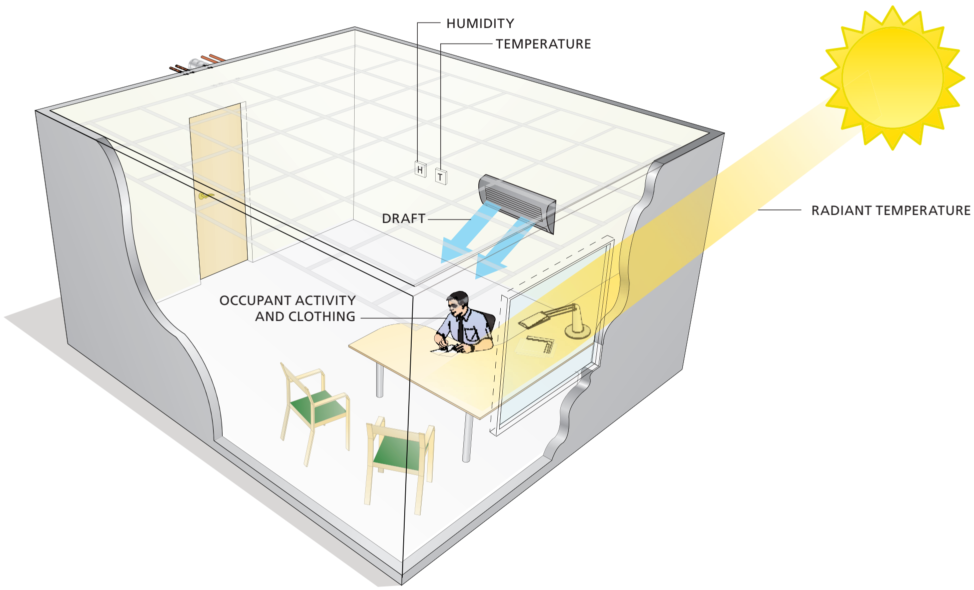
The DOAS unit should be design to achieve:

- 49 °F dry bulb, 49.7 gr /lb off the cooling coil with 5 °F reheat primary air
- Approximately 0.75 cfm/ft² primary air
- 5 °F reheat
- 100% outdoor air
- Total energy recovery device to minimize energy cost

Figure 14 – Classroom with ADAPT Parasol PF Climate Beams

Part 3: Comfort vs. Capacity - The Balanced Approach

Figure 15: Balancing Parameters – Factors Affecting Thermal Comfort



When selecting climate beams, it is very easy to become fixated on meeting the design cooling load. A good design covers much more than just an energy balance in the occupied space. The following must all be met for the climate beam design to be considered complete;

- Meet comfort criteria (50 fpm non isothermal covering entire occupied space)
- Meet sound criteria (NC 20 for individual climate beam)
- Meet sensible cooling load
- Acceptable air pressure drop (0.3 to 0.6 in. w.c.)
- Acceptable water pressure drop (less than 10 ft hd)

Climate Beam Selection Methodology

The process described in the previous examples allows the overall primary airflow rate and the design conditions for the primary air (DOAS) unit to be established. The actual climate beam selections are performed by the manufacturer based on the specific zone loads and other design criteria. The climate beam provider can adjust the following parameters;

Primary Air Conditions

Most WBCS systems use a dedicated outdoor air system (DOAS) with 100% outdoor air. They often include some form of energy recovery which greatly minimizes the penalty of introducing outdoor air above the ventilation requirement.

The primary air humidity ratio is established in the above procedure and through experience. If the design requires a coil leaving temperature to achieve the humidity ratio that is more than 2–4 °F below the space dewpoint, reheat will be required. This is to avoid condensation on the climate beam air box.

Climate Beam Size and Type

Most manufactures have a wide range of models that offer different sizes and air flow patterns to meet the space condition.

Figure 16: Climate Beam Types

Primary Airflow Rate per Climate Beam

While the overall airflow rate was established in the previous examples, the actual primary flow rate per zone can be adjusted. For example, a conference room will likely require a higher primary air flow rate to meet the latent load then the rest of the office space. The beam manufacturer may also need to make minor corrections to achieve the specific zone design conditions. A final primary airflow check should be made after the beams are selected by comparing the sum of beam primary airflow to the DOAS unit primary air flow.

Chilled Water Parameters

The chilled water temperature is typically set for the whole building 2 to 3 °F above the dew point of the space. Common supply water temperature is 57 to 60 °F. The chilled water range through the beams is usually 4 to 6 °F. The supply water temperature in not generally changed per beam but the flow rate and the range can be adjusted per beam to meet the zone conditions. The water pressure drops are generally less than 10 ft. w.c. The water control valve also needs to be considered for hydronic design.

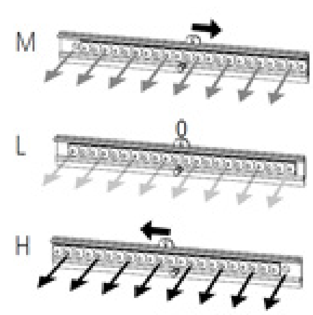
Do not use antifreeze in the chilled water servicing the climate beams as it greatly reduces the cooling and heating capacity. If antifreeze is required for part of the chilled water system, isolate it from the climate beams with a heat exchanger.

Induction Ratio

The amount of zone air that is induced through the beam is dependent the amount of primary air and the nozzle design. The quantity and size of nozzles can be changed. Some manufacturers can adjust how many nozzles are actually being used (Figure 17). This has the advantage of allowing the beam capacity to be changed in the field.

Higher induction ratios shift the cooling from the primary air to the climate beam which is generally more energy efficient. To increase the induction ratio, smaller nozzles are used to induce more air flow which tends to increase the noise level and the climate beam pressure drop. Induction ratios between 3 to 5 and an air pressure drop on 0.3 to 0.6 inches w.c. are common in high SHR spaces such as offices. It is a good idea to try and keep the climate beam pressure drops close to the same in a common duct branch to make commissioning easier.

Figure 17: Adjustable Induction Nozzles



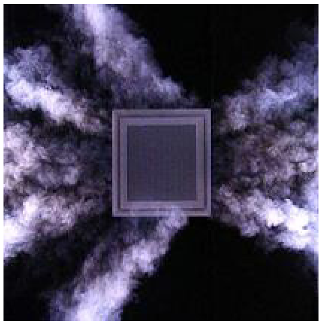
Airflow Pattern and Throw

Most manufacturers can adjust the airflow pattern from the beam using turning vanes and by adjusting the nozzles. This can be done to achieve an acceptable air velocity in the occupied zone to avoid drafts.

Sound Levels

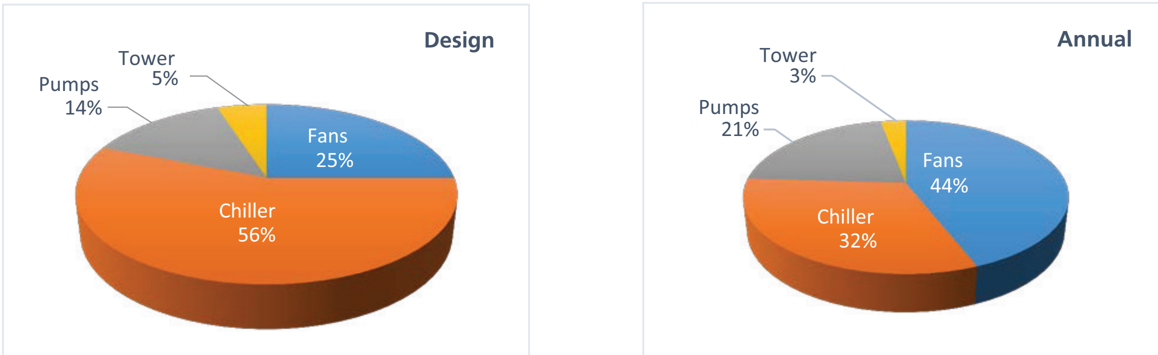
Beams are inherently quiet as there are no moving parts. However, some noise can be generated in the primary air inlet or the nozzles. Most manufacturers have sound data so they can estimate the NC level in the space and make adjustments in model and design parameters to achieve the desired sound level.

Figure 18: Adjustable Airflow Pattern



Part 4: Energy Considerations

Figure 19: Design vs. Annual Energy Usage



A key parameter of any HVAC system along with IAQ and comfort is energy efficiency. This is important both as an operating cost to the owner and an environmental issue (Carbon footprint).

Figure 19 shows that it is annual performance rather than design performance that dictates the real building energy usage. The charts are based on an office with a VAV/chilled water system in Chicago. While the chiller may be the largest component in an HVAC system and the largest energy user on a design day, in this application, the AHU fans represent the largest energy user on an annual basis. The chiller rarely runs at full capacity and only during summer months while the supply fans have to operate whenever the building is occupied.

Another observation that can be made is how much of the annual operating cost is spent on "transportation" costs – pumps and fans. Air is an expensive method to move energy around a building. You can move as much energy in a 10" duct, as in a 3/4" water line or a 5/8" refrigerant line. (Figure 20)

WBCS systems have several advantages to reduce the annual energy cost;

- In a high SHR application, 2/3 of the sensible space cooling is performed with chilled water in the beams. This is a low transportation cost method of collecting and extracting the heat from the zone.
- The primary air is reduced from 1.2 cfm/ft² for a VAV system to around 0.4 cfm/ft². VAV systems modulate the air so the annual fan work is greatly reduced but it is still higher than a properly designed WBCS system. A Demand Control Ventilation WBCS system also can modulate the primary air, further increasing the fan savings. (Figure 21)
- WBCS systems induce the zone air through the beams using energy in the primary air flow. This shows up as an air pressure drop though the beams of about 0.5" w.c. The work is done by the supply fans which can have very efficient fans and motors. This is comparable to a VAV system including the pressure drop

Figure 20: Transportation Cost

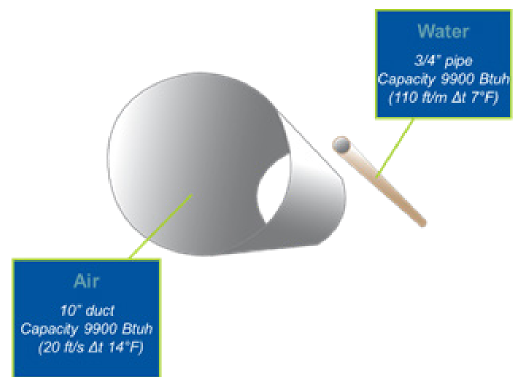


Figure 21: Adapt Parsol Demand Control Ventilation Beam

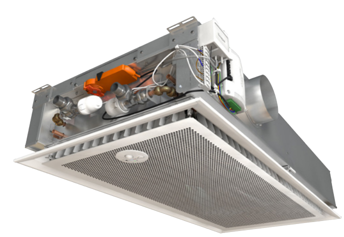
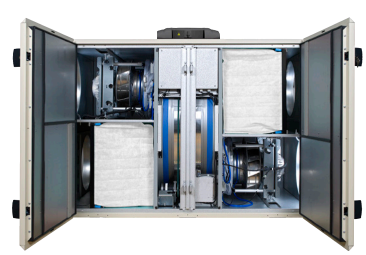


Figure 22: Energy Recovery Primary Air Unit



through the VAV box. A fan coil solution will also require about 0.5” w.c. at the fan coil but both the fans and motors are often very inefficient. A shaded pole motor (standard motor on a fan coil is around 30% efficient (ECM motors are available but are costly) and typically operate on 120v single phase which is costly and inefficient to deliver throughout the building.

- WBCS solutions use 100% air DOAS system which are very easy to add energy recovery to. Total energy recovery devices with 85% efficiency recover most of the energy in the exhaust air stream.
- Where the climate makes sense, an all air VAV system can use air side economizers. In the same climate zones a WBCS control sequence can prioritize primary air first (free cooling) then chilled water (mechanical cooling). As well a WBCS can utilize water side free cooling to avoid mechanical cooling in cooler weather.
- When chillers are dedicated to just serving the beams, the high (57 °F) chilled water can allow significantly more BIN hours for water side free cooling. Moreover, increasing the supply water temperature from 44 °F to 57 °F results in a 30% improvement in chiller efficiency.
- When the chiller plant serves both the DOAS and climate beams, the loads can be placed in series improving the temperature range and helping to minimize the pump work.
- Energy recovery in the chiller during operating hours can be used for heating loads such as domestic hot water.
- Heat pump chillers can be used to reduce heating loads.
- Simultaneous heating and cooling chillers can be used to move heat energy from the chilled water loop to the hot water loop during periods when there is both heating and cooling

Energy Considerations – Primary Air Design Conditions for Offices

Managing the use of primary air is critical to a successful climate beam design. Several installations where the primary airflow has been oversized to “ensure” there are no condensation have resulted in very little cooling being performed by the climate beams. It can be seen through BAS trending that the climate beam valves rarely open. There is so much primary air that the spaces are being over cooled during light load conditions. Not only are these systems failing to provide comfort (overcooling) they are expensive to operate.

Figure 25 shows a typical single energy recovery rotor Dedicated Outdoor Air System unit (DOAS) based on Chicago design conditions.

The primary air is heavily influenced by the latent loads. The latent load is an outcome of how the building will be used (i.e. office vs. classroom). The HVAC system must meet the load. The designer is faced with a range of primary air flow vs. primary air dewpoint conditions that can satisfy the latent load. Is it better to supply a small amount of low dewpoint primary air (low fan work but higher cost to dehumidify) or a larger amount of higher dewpoint primary air (higher fan work but lower cost to dehumidify)? Figure 26 shows the change in primary airflow as the humidity ratio of air is reduced. In this example, the primary air can deliver 258 Btu/h of latent cooling at any point on the curve.

Figure 23: Chiller with Integrated Free-cooling

Figure 24: Heatpump Chiller

Figure 25: Standard Energy Recovery DOAS Unit

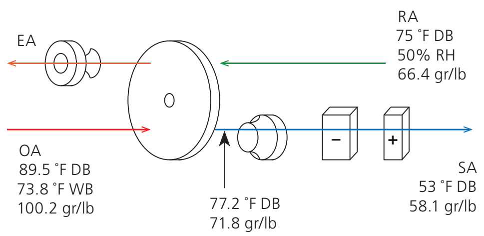
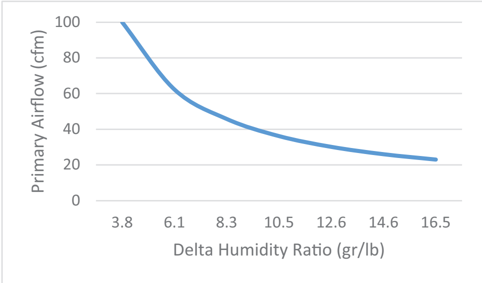


Figure 26: Primary Airflow vs. Delta Humidity Ratio to Deliver Constant Latent Cooling



As the primary air temperature and humidity ratio is lowered the zone sensible cooling capacity changes. There is less primary air being delivered (The latent cooling capacity remains constant) however the drybulb temperature is lower. Figure 27 shows how the zone sensible cooling capacity drops as the airflow is reduced even though the discharge temperature is lower.

An outcome of Figure 27 is that more of the zone sensible cooling load is being transferred to the water coils in the beams as the primary air humidity ratio is lowered. This is desirable from an energy usage point of view. To increase the beam capacity, the induction ratio needs to increase so more zone air passes through the coil. Higher induction ratios are achieved with higher pressure drops across the nozzle bank and tend to lead to higher noise levels.

Figure 28 shows the change induction ratio required to increase the beam capacity to maintain the zone sensible cooling level as the primary air flow decreases. As a practical issue, induction rates above 5 tend to have unacceptable pressure drops and noise issues.

Another practical issue is that lower primary air drybulb temperatures can lead to condensation forming in ducting or the beams themselves. Experience shows that primary discharge drybulb temperature should be within 2–3 °F of the space dewpoint. For 75 °F, 50% RH space design condition, the dewpoint is 55.2 °F. Hence discharge primary air drybulb temperatures can be 52–53 °F without issue. Fan reheat (drawthrough arrangement) and duct heat gain will raise the supply air temperature to help avoid condensation issues.

Trending the primary air humidity ratio and airflow rate down improves overall energy performance but is limited by rising induction ratios and falling discharge primary air drybulb temperatures.

Figure 29 shows the working zone for the primary air condition when high induction ratios and low discharge air temperatures are taken into consideration. These two factors set practical limits.

Energy Considerations – Primary Air Design Conditions for Schools

Some applications such as schools need much more latent cooling than can be delivered by the basic DOAS unit shown above.

Figure 30 shows a typical DOAS unit with an enthalpy wheel and a plate type sensible heat recovery device to deliver reheat. Based on 100 cfm supply, the unit can deliver 1027 Btu/h (10.3 Btu/h-cfm) of latent cooling. Compare this to the DOAS unit shown in figure 23 which can deliver 470 Btu/h (4.7 Btu/h-cfm) latent cooling with a 53 °F discharge air drybulb temperature.

The chilled water coil cools the air to 49°F to lower the humidity ratio. Heat in the return air used to reheat the primary air 4°F to avoid condensation issues at the beam. The leaving return air (from the sensible heat exchanger) is now cooler than entering return air which improves the effectiveness of the main total recover energy wheel. The sensible heat exchanger does add another air pressure drop to both the supply and return fans (0.5" w.c.). The lower chilled water temperature required to cool the primary air lowers the chiller COP by 10%. Table 5 compares the key system parameters between the two approaches.

Figure 27: Primary Airflow vs. Primary Air Sensible Cooling Capacity

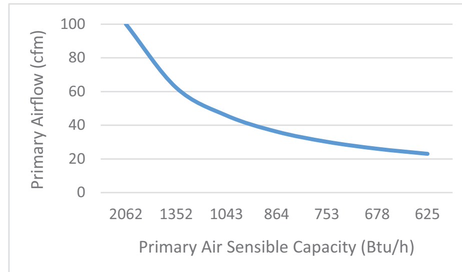


Figure 28: Induction Ratio vs. Primary Airflow to Deliver the same Zone Sensible and Latent Cooling

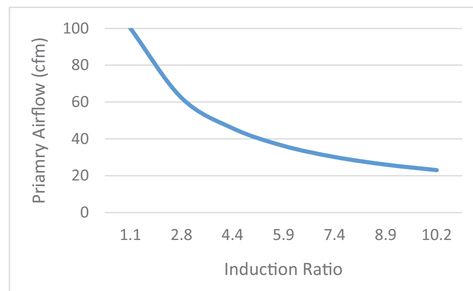


Figure 29: Acceptable Primary Airflow Range

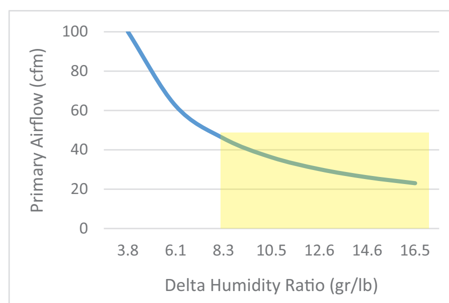


Figure 30: Low Dewpoint Energy Recovery DOAS Unit

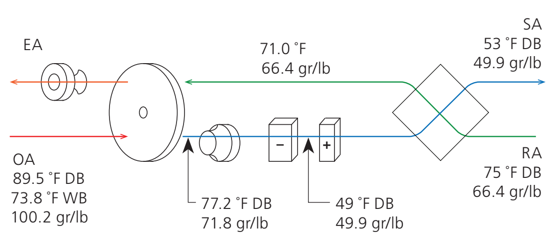


Table 5: DOAS Unit Design Conditions

	Standard DOAS		Low Dewpoint DOAS	
	Supply Air	Return Air	Supply Air	Return Air
Airflow (cfm)	100	100	100	100
Fan Eff. (%)	70	70	70	70
TSP (in wc)	4	3.5	4.68	4
Wheel Eff. (%)	85	85	85	85
Plate Eff. (%)	NA	NA	50	50
Chiller COP	4.8		4.3	
Supply Air DB (F)	53		53	
Supply Air HR (gr/lb)	58.1		49.9	

The low dewpoint DOAS unit is well suited for applications such as school classrooms with high latent loads.

Annual Energy Usage Examples

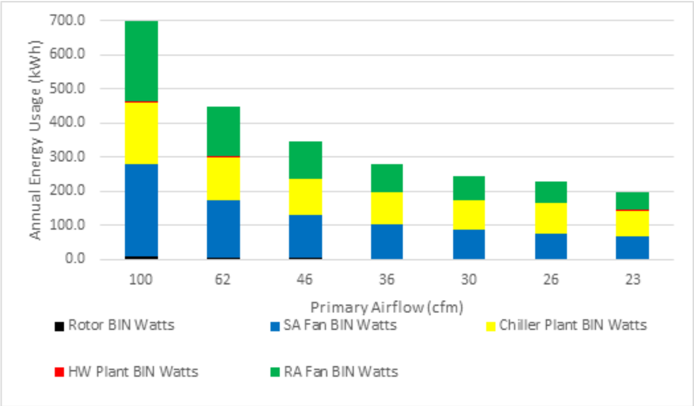
Table 6: DOAS Unit Design Conditions

Off Coil DB	SA DB	SA HR	Delta HR	SA Airflow
F	F	gr/lb	gr/lb	cfm
55	56	62.6	3.8	100
54	55	60.3	6.1	62
53	54	58.1	8.3	46
52	53	55.9	10.5	36
51	52	53.8	12.6	30
50	51	51.8	14.6	26
49	50	49.9	16.5	23

The two DOAS unit models described above were modelled in Chicago for standard occupancy (4015 hours). Each model was run at the following conditions;

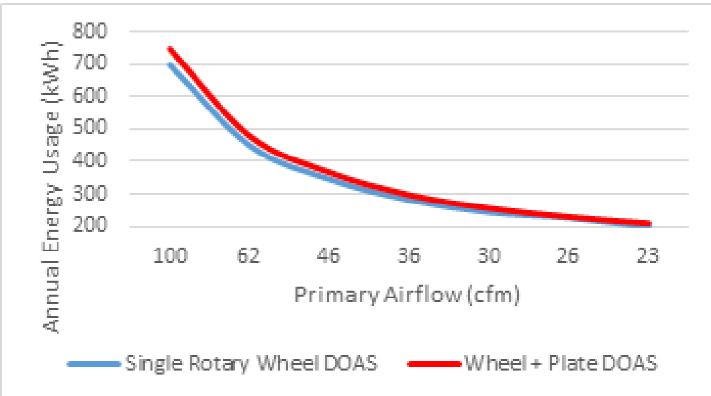
Figure 31 shows the annual energy usage for a standard DOAS unit by component. Each off coil condition is shown. As a reminder, each airflow and off coil condition provides the same amount of latent cooling in the occupied zone.

Figure 31: Annual Energy Usage for Single Wheel DOAS Unit



What can be inferred from Figure 31 is that the fan work is dominant over chiller plant work. This means it is better to use a drier primary air at a lower airflow to manage the latent load from an energy point of view. However, it must be remembered that there are limitations on induction ratio and discharge air temperature that set practical limitations.

Figure 32 – Comparison of Annual Energy usage for Std and Low Dewpoint DOAS units



Using a low dew point DOAS arrangement like the one shown above solves the condensation issue due to low primary air drybulb temperature. The reheat in the unit will raise the discharge air temperature to an acceptable level.

Figure 32 shows that the operating cost for a low dewpoint air DOAS unit is actually higher at any given design condition. This is to be expected as the additional air pressure drop from the second energy recovery device increases the total fan work on both the supply and return side. As the chilled water temperature is lowered to reach the lower dewpoint condition off the coil, the chiller plant is penalized. If the chiller plant is dedicated to just the DAOS system, then the penalty is localized to just the DOAS chiller plant. However, if the entire building chiller plant has to have the chilled water temperature lowered to serve the DOAS unit, then the energy penalty can be very significant. Either dedicated chillers or DX systems should be considered.

This indicates that for higher sensible heat ratio (i.e. offices) applications a standard DOAS unit with the discharge air temperature around 2–3 °F below space dewpoint will yield the best result. For lower sensible heat ratio (i.e. schools) applications, a low dewpoint primary air system is required that includes reheat to achieve an acceptable discharge air drybulb temperature.

Demand Control Ventilation

Figure 31 shows that regardless of primary air design conditions, the fan work (supply air plus return air) dominates the energy usage in an energy recovery DOAS unit. In this example in Chicago, it is almost 2/3 of the total annual energy usage.

Studies have shown that K to 12 classroom are occupied only 35% during school hours and office spaces are only occupied 22%–38% of business hours.

By far, the best way to improve a WBCS system energy performance is to use Demand Control Ventilation (DCV). DCV is the concept of reducing the primary airflow when it is not required.

Recall that the purpose of the primary air is to meet the needs of ventilation for IAQ, latent cooling and sensible cooling. All of these loads vary directly with the actual occupancy of the space. As the occupancy increases, it can be expected that all three requirements will increase as well.

ASHRAE Standard 90.1 requires DCV for spaces larger than 500 ft² and a design occupancy of greater than or equal to 25 people per 1000 ft² [for systems with outdoor air flow greater than 3000 cfm]. This allows the load created by the high primary airflow rate to be decreased when not required. ASHRAE Standard 62 allows DCV providing the minimum ventilation level does not drop below the building load component (Ra × Az).

Common ways to control the primary airflow include;

- Temperature + Occupancy sensor
- CO_2 sensor
- VOC sensor

An occupancy sensor (Figure 33) can be used to detect the presence of personnel and switch the WBCS from minimum primary airflow to design primary airflow. This is a low cost way to manage primary airflow and works well for single or low occupant spaces as the primary airflow will go from minimum to design airflow once the sensor detects an occupant. There is no modulation.

CO_2 sensors (Figure 34) measure the CO_2 level is the occupied space. CO_2 itself is not a pollutant but an easily detected trace gas that increases with the rise of pollutants associated with occupants. As the occupancy level increases, the CO_2 increases and that measurement can be used to increase the primary airflow to maintain acceptable indoor air quality. ASHRAE Standard 62 requires that the CO_2 level not exceed 700 ppm from the background level.

An advantage of a CO_2 sensor over an occupancy sensor is that can provide a modulating signal so the primary airflow rate need only be increased enough to maintain an acceptable air quality level. As well, there is a time lag between occupants entering a space and the buildup of pollutants to a level where the primary airflow needs to be increased. An occupancy sensor will increase the primary airflow as soon as occupants are detected. A CO_2 sensor will take advantage of the time delay to minimize ventilation airflow. A good example is a movie theater where the time delay can greatly decrease the total primary delivered to the theater during the movie sitting.

A VOC or Volatile Organic Compound Sensor measures a common group of pollutants that can also be used a measure of indoor air quality. A VOC sensor offers the same advantages that CO_2 sensor has including a modulating signal.

WBCS System Changes to Achieve Demand Control Ventilation

Several changes to a WBCS system are required to achieve a working Demand Control Ventilation system. Standard WBCS have a constant flow primary air system. A Demand Control Ventilation system is a variable flow system. This requires the DOAS unit to be variable air flow including such components as fan VFDs or inverters, duct pressure sensors to modulate the fans and additional control algorithms to ensure the energy recovery, heating and cooling systems all work properly with variable airflow.

Mixing spaces that utilize Demand Control Ventilation with spaces that are constant volume is possible by using a constant airflow damper to maintain a constant airflow to zones that are not demand control ventilated.

The conventional method to reduce the primary airflow to the beam is introduce a VAV box upstream of the beam. A control algorithm

Figure 33: Occupancy Sensor

Figure 34: CO_2 Sensor

reduces primary airflow to minimum during unoccupied periods. The primary airflow is increased if the CO_2 or VOC level starts to rise or if the space temperature cannot be met with just the chilled after passing through the beam coil.

Modulating the primary airflow to a beam has a significant impact on how the climate beam operates. Recall that the beam uses the primary airflow and pressure to induce space air to pass through the cooling coil. This induction process at the nozzle bank is proportional to primary airflow². This means small changes in primary airflow have a big effect on beam performance. Figure 37 shows the impact of climate beam cooling capacity (climate beam discharge) as the primary airflow is reduced. The sharp drop in cooling capacity can lead to system hunting and comfort issues.

Another undesirable impact from modulating primary airflow to a standard climate beam is the impact on room mixing. The climate beam is both a cooling (heating device) and an air diffuser. The discharge air slots are designed to create coanda effect for good room air mixing. If the primary air rate is reduced to the point where the induced airflow drops off dramatically, the total airflow through the diffuser slot will not be correct and the coanda effect will likely breakdown. Further, the air that does discharge will be mostly primary air which is much colder (and denser) than the intended mixt primary-induced air so dumping can occur.

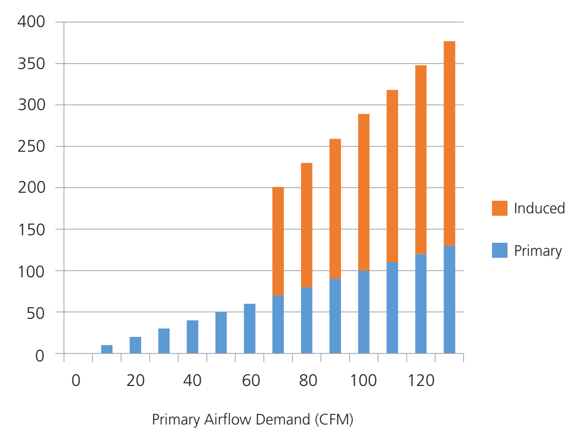
Using a VAV box will work but care must be taken to ensure comfort will be maintained. The turndown may be limited.

To improve performance, beams such as the ADAPT Parasol are purpose built to vary primary airflow. The primary airflow damper is part of the beam. More importantly the number of nozzles being used is changed based on the primary airflow rate. This helps maintain an acceptable induction ratio and good mixing. These beams can include factory mounted and tested occupancy, CO_2 or VOC controls to manage the primary airflow.

Figure 37: Conference Room using ADAPT Parasol

Figure 35: Airflow Control Damper

Figure 36 – Impact on Beam Cooling with Varying Primary Airflow



Energy Impact of Demand Control Ventilation

Since a Demand Control Ventilation system is a VAV system it will not follow the fan affinity laws (CAV) because the beams will adjust their damper changing the system curve. Instead the fans will follow an unloading profile closer to the blue line in Figure 39. The red line shows the fan power curve for a constant volume system following the fan affinity laws. Regardless, it can be seen that by the time the VAV primary airflow drops to 50% the fan power has been reduced to 1/3. Considering the fans are the dominant energy user in an energy recovery DOAS unit, a 50% reduction in airflow results in 63% reduction in power.

Couple this with studies that show how much time most spaces are unoccupied and it becomes clear that Demand Control Ventilation is the best way to improve the energy performance of a WBCS system.

Once Demand Control Ventilation is introduced to a system, the designer has a choice to use primary air first, then chilled water or chilled water first, then primary air as a means to increase the sensible cooling capacity. The control sequences are shown in Figures 40 and 41.

At first it would seem that the chilled water to the beams should be prioritized as the fan work in the DOAS unit should be minimized. However, when the weather allows air side economizing, then the chiller savings may offset the fan work penalty and air over water is desirable. When the outdoor air is either hot or cold, then it is generally better to prioritize water over air in a Demand Control Ventilation system. Which arrangement will yield the best overall energy result will depend on building location and usage.

Conclusions

Anytime something new is tried it can seem daunting. Designing a WBCS system may seem challenging but is actually easier than many other HVAC systems because;

- Load calculations are the same as any other HVAC system. Use a fan coil or other decentralized model in the load calculation program if climate beams are not offered.
- The ventilation rate is based on the ASHRAE single zone method.
- The general primary airflow and supply air conditions are usually based on latent load and can be quickly calculated.
- The climate beam manufacturer will pick the climate beams to meet the cooling and heating requirements, sound criteria and comfort criteria.
- The primary air system will be a DOAS unit with straight forward controls logic.

Figure 38: Fan Energy vs. Airflow for VAV Applications

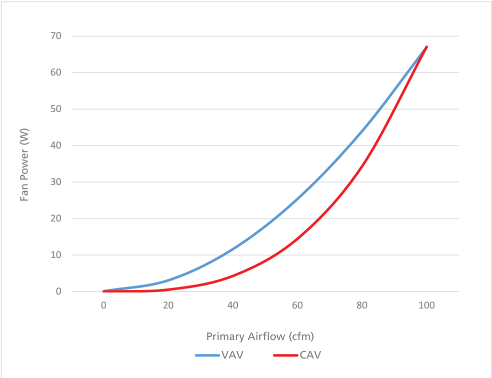


Figure 39: Air Over Water Cooling Priority

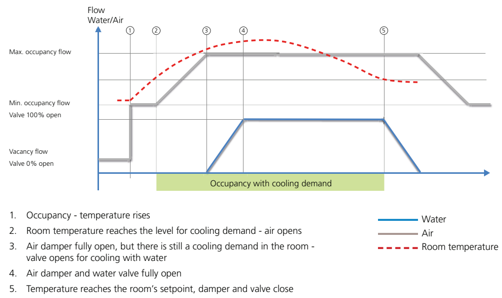
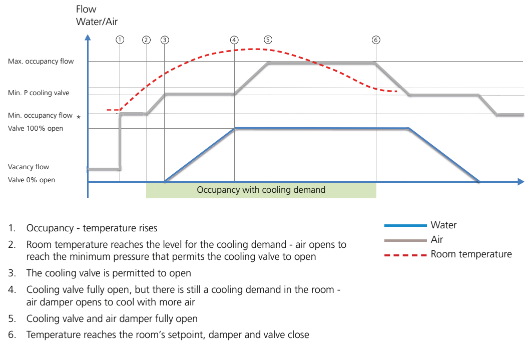


Figure 40: Water Over Air Cooling Priority



Design Parameters Summary

The following guidelines will assist in expediting the design;

- The zone sensible cooling load should be between 20 to 40 Btu/h·ft². If the loads are higher than this, review load calculations.
- Consider a space design condition of 75 °F DB and a relative humidity of 50 to 55%. Use the high end of the range for higher latent loads.
- The primary air dry bulb temperature should be within 2 to 3 °F of space dewpoint to avoid condensation. If a lower humidity ratio is required consider raising the space RH, using a higher primary airflow with demand control ventilation or reheat.
- The primary airflow will be mostly likely set by the zone latent load. A good range is 0.4 to 0.6 cfm/ft². If a higher primary airflow is required to meet the space condition, consider demand control ventilation to minimize the fan work and operating cost.
- A good office system has 1/3 of the load met by the primary air and 2/3 of the load met by the climate beam coil. Assume an induction ratio between 3 to 5. Start with 4.
- Assume the discharge air temperature for the air induced through the climate beam coil will be in the range of 60 to 65 °F. Start with 64 °F.
- The chilled water supply temperature should be 2 to 3 °F above space dew point. 57 °F is a common supply water temperature.
- The chilled water temperature range will be 4 to 6 °F. Consider putting the primary air system in series with the climate beams.
- Use Demand Control Ventilation wherever possible to reduce primary system energy usage.
- When Demand Control Ventilation is used, evaluate whether air or water should be prioritized at the beams.