Commercial

and Industrial

Climate Control

Systems

Dedicated Outdoor Air Systems (D.O.A.S.)

AMDOAS-1

Applied Air

Applied Air

D.O.A.S.

- What is it?
- Why use one?
- What are my options?
- Selection considerations
- Design steps

Applied Air

D.O.A.S.

- What is it?
  - A separate unit dedicated to the management of outdoor air quality

Outdoor air is a necessary component of any well-designed HVAC system. In many climates however the air outside is either so full of moisture or so contaminated that bringing it into a building can create problems for a building's occupants and contents. Many engineers today have concluded that the best way to handle these issues is to use an HVAC unit that specifically solves these problems.

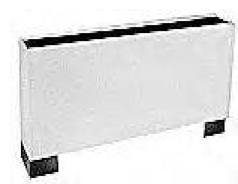
Applied Air

D.O.A.S.

- ## What is it?

  A "Dedicated Outdoor Air System" is just that….a system. The DOAS unit provides the ventilation air control but the system also requires a parallel sensible cooling system to handle the other building loads. There are many options for the parallel sensible cooling system.

Radiant Cooling Panels



Fan Coil Units

Rooftop Units with no OA Section

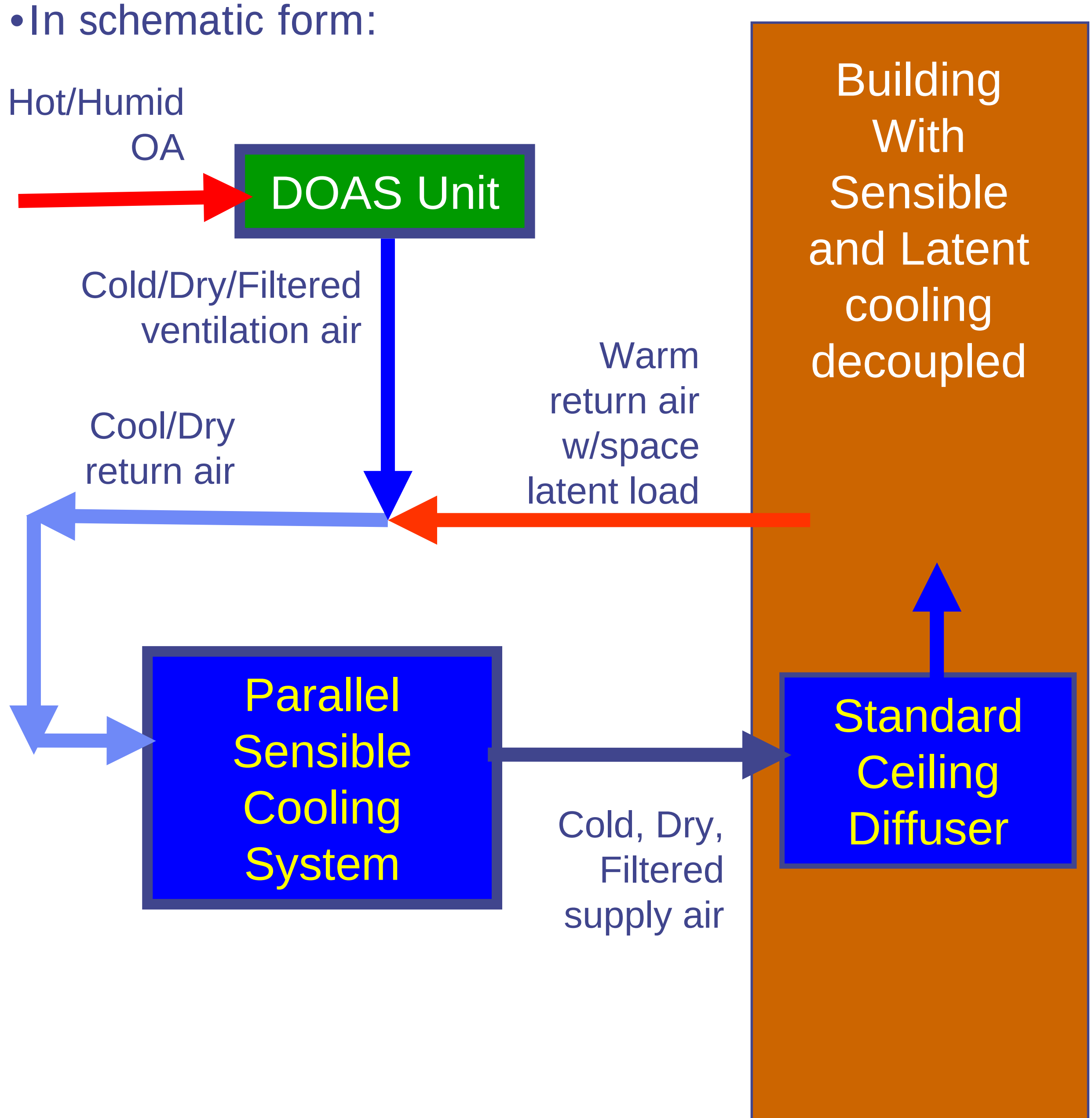
Packaged Terminal Units

Applied Air

D.O.A.S.

◆What is it?

• In schematic form:



Applied Air

D.O.A.S.

- Why use one?
  - Higher employee productivity

The significance of ventilation air is illustrated by estimates that US companies lose as much as ***$48 Billion*** annually to cover medical expenses and ***$160 Billion*** annually in lost productivity as a result of sick-building illnesses.

Sick-building illnesses can most often be traced to a lack of effective ventilation air in the occupied spaces. Out-gassing of construction materials, fumes from office equipment, and normal build-ups of CO2 from human occupants all contribute to poor indoor air quality and sick-building illnesses.

For these reasons ASHRAE has developed the ventilation requirements in Standard 62.1-2004.

Applied Air

D.O.A.S.

- Why use one?
  - Better humidity control

At the same time, designers must be aware of the amount of moisture they are adding to the building environment when they introduce the required amount of outside air.

Human occupants and certain machines in a building will generate moisture that must be controlled, but the largest element of moisture in a building comes from the outside. If an uncontrolled amount of moisture is continually added to a building then an environment for the growth of mold is the end result.

Even if mold is not a concern it should be noted that occupant comfort is not simply a matter of the room dry bulb temperature. A comfort index is a combination of the dry bulb temperature, local air velocity, and moisture content of the air. All three factors must be addressed in a well-designed building.

Applied Air

D.O.A.S.

- Why use one?
  - LEED certification

As an additional motivation to use the DOAS concept engineers need to consider the impact of LEED-NC2.2. In many states, California being the most notable, all government buildings must meet the minimum LEED certification level. Other institutional owners are also requiring LEED certification. LEED is also gaining ground with corporations such as Bank of America and Starbucks.

In order to earn any LEED-NC2.2 points a building must be designed to meet ASHRAE 62.1-2004. Since 62.1 is a ventilation standard and the DOAS concept is focused on providing the best ventilation air quality, the application of a DOAS system generates up to 80% of the points required for a LEED certification.

D.O.A.S.

Why use one?

- Limitations of conventional equipment

Conventional packaged rooftop equipment has been the most common type of application for buildings of all sorts. When outside air requirements are minimal such as in sparsely occupied buildings or in climates where moisture control is less of an issue then a conventional rooftop unit can effectively handle the necessary outside air (although there may be other reasons to treat outside air separately).

But conventional rooftop units have some specific limitations when it comes to handling outside air:

1. Conventional rooftop units are designed to provide 350 to 450 cfm/ton, but effective dehumidification typically requires only 200 to 300 cfm/ton. ASHRAE 62.1-2004 (section 5.10) limits RH to 65% or less at design dewpoint.

D.O.A.S.

- ## Why use one?
  - Dehumidification and drain pans

2. Coils in conventional rooftop units are typically only 3 to 4 rows deep with a drain pan that extends just past the end of the coil. Effective dehumidification often requires coils up to 8 rows deep. It is also important to provide additional drain pan depth and IAQ-style designs in order to properly handle the amount of moisture that will be removed.

   ASHRAE 62.1-2004 (section 5.11) requires that drain pan lengths be at least ½ of coil height or designed to limit moisture carryover to .0044 oz.per coil sq.ft. per hour at peak dewpoint condition. On most units this means that the drain pan must be at least 18” long and could approach 30” in larger tonnage units. This will almost never be found in a conventional packaged rooftop unit.

Applied Air

D.O.A.S.

Why use one?

- Better Filtration

3. Filtering capacity is one of the most significant differences between a DOAS unit and conventional equipment. A conventional rooftop unit is usually limited to 2” throw-away filters both from an available space and an available fan static pressure point of view. A well-conceived DOAS unit can accommodate much higher levels of filtration.

   ASHRAE Standard 62.1-2004 requires a minimum filter MERV rating of 6 upstream of any dehumidifying coil.

   A certification point for LEED NC2.2 EQ credit 5 can be earned by increasing filtration to at least a MERV 13 rating...again putting pressure on conventional rooftop package designs.

   Houston, Galveston, Riverside, Fresno, and Long Beach are also classified as EPA non-attainment zones and require even more aggressive filtration.

Applied Air

D.O.A.S.

Why use one?

- Better heat exchangers

4. Heat exchangers and burner designs for conventional rooftop units typically are based upon handling a relatively steady temperature. The return air/outside air mixed temperature varies only slightly over the course of a year. For that reason it is common to find heat exchangers that are made of mild steel that can be easily formed in tube benders and forming presses. When these heat exchangers are exposed to very wide temperature spreads and are controlled to reduce capacity as the load decreases then internal condensation will occur and mild steel heat exchangers will quickly corrode.

Applied Air

D.O.A.S.

Why use one?

As a final couple of points regarding use of a DOAS concept:

Conventional rooftop equipment is designed to operate efficiently in a relatively stable temperature environment. By mixing the outside air with return air the resulting temperature across the cooling section and the heating section stays within a fairly narrow and predictable range. When the amount of outside air required to meet ASHRAE 62.1 approaches 30 or 40% of the rooftop unit's total air then the mixed temperature starts to vary outside of the desired application range of the equipment. This can lead to reduced efficiency and life of the equipment.

It has also been said that using the DOAS concept simplifies the design process. This is because the engineer need only filter, temper, and dehumidify the outside air at a single point. Given the wide range of temperature, humidity, and cleanliness of outside air it does appear easier to solve the problem only once on a building instead of at every rooftop unit.

D.O.A.S.

What are my options?

When designing a dedicated outdoor air system the engineer has two basic decisions to make. He can design the outdoor air unit to supply air at the same temperature as the inside design temperature or at the temperature required to assure dehumidification of the air.

Once that decision is made he must then decide if he wants to try to put the outside air directly into the occupied space or mix it into the air supplied by the parallel sensible cooling system.

This decision process is characterized by this matrix:

Neutral air to the space	Cold air to the space
Neutral air to the parallel system	Cold air to the parallel system

Applied Air

D.O.A.S.

Selection considerations

When considering the use of a “Neutral Air” DOAS the engineer should consider the following:

Because the outside air temperature will be reduced below the outside air dewpoint to dry the air a neutral air system will almost always include a reheat load. This will typically result in coil leaving air temperatures in the low 50s. Since a neutral air system is designed to deliver air at room temperature then that 50-degree air must be reheated some 20 to 30 degrees in most cases.

Good design practice suggests that reheating air is not an energy sensitive approach. The engineer will need to locate and organize sources of waste heat that can be captured and routed through the DOAS in some fashion to provide “free reheating”. As an alternative the engineer could require that the DOAS include hot gas reheat coils/controls that essentially capture rejected heat from the condenser side of the DOAS unit.

Applied Air

D.O.A.S.

Selection considerations

Both reheat options can add significant expense to the DOAS unit and a payback analysis should be part of the process. The engineer should also consider the reliability of the method chosen for reheat since the entire system concept depends upon neutral air temperatures from the DOAS.

A neutral air system will produce air at a temperature that is very easy to mix with the occupied zone air. For that reason it is quite easy to introduce the ventilation outside air directly into the rooms using conventional diffusers. In applications that use some sort of perimeter parallel system such as a “through-the-wall-PTAC” this can be an advantage. The ventilation outside air need only be ducted to the room and not down to the PTAC unit.

In VAV applications there could also be advantages to the neutral air concept. Because the ventilation outside air can be introduced directly to the space the ventilation air can be totally separate from the parallel sensible cooling system allowing that parallel system to vary the amount of recirculated air to match the sensible load in the space.

D.O.A.S.

Selection considerations

The cost of running ductwork to every room can be eliminated by routing the neutral temperature ventilation outside air into the return side of the parallel system rooftop units. This might involve a simple “spider” arrangement to feed multiple rooftop units or an even simpler approach of putting the ventilation air into the return air plenum.

Unfortunately neutral air does not provide any cooling assistance to the parallel system. Although the neutral air DOAS uses energy to lower the ventilation outside air temperature to a point that also provides sensible cooling, the neutral air concept “gives that energy back” by reheating the air to a temperature that does nothing to offset any building sensible cooling load. The parallel system cooling capacity will remain unchanged as will the parallel system air delivery requirements.

D.O.A.S.

Selection considerations

When considering the use of a “Cold Air” DOAS the engineer should consider the following:

A cold air system will not require a reheat load. Because the outside air temperature will be reduced below the outside air dewpoint to dry the air and then will be distributed in that condition.

When this cold air is routed into the return air stream of the parallel system it reduces the parallel system’s return air temperature (and enthalpy) and the resulting load on the parallel system cooling coil. The parallel system equipment can be downsized to take advantage of this sensible cooling assistance. The total amount of parallel air delivered to the space is not affected (since it still needs to satisfy the space sensible load) but the cooling capacity of the parallel system can be reduced. Thus the fan energy of the parallel system will not change but the compressor/condenser energy will be reduced.

D.O.A.S.

Selection considerations

Unfortunately it is sometimes quite difficult to mix the cold ventilation outside air with warm room air directly. The much more dense cold air will tend to drop rapidly and can result in comfort complaints due to drafts. Using high-induction ceiling diffusers can solve this problem but they are more expensive than low-induction diffusers. This can limit the application of cold air DOAS units in cases where the typical parallel system is a “through-the-wall-PTAC” or something similar.

VAV applications with cold air can sometimes be challenging. Since introducing cold ventilation outside air directly to the room might increase cost, it is tempting to introduce the cold air into the return of a VAV rooftop unit. However as the rooftop unit reduces airflow to match the changing sensible load it will also reduce the amount of ventilation air possibly below the ASHRAE 62.1 requirement.

Applied Air

D.O.A.S.

Selection considerations

In those applications that will use VAV parallel systems it is best to plan on using high-induction diffusers to introduce the ventilation outside air directly to the occupied zones. This approach will still reduce the sensible cooling requirement of the parallel system and will assure constant compliance with ASHRAE 62.1.

As with the neutral air system it is important to evaluate the economics of the cold air system. The higher cost of the high-induction diffusers may very well be more than offset by the reduction in the cooling capacity of the parallel system. Of course if the parallel system will not be VAV then the cold air system can also be distributed through a “spider” arrangement or directly to a return air plenum offering an obvious cost advantage over the neutral air system.

Applied Air

D.O.A.S.

Design Steps

- In general:
  - Three factors will determine the DOAS unit cooling coil load:
    - Ventilation airflow
    - Outside air enthalpy condition
    - Leaving air enthalpy condition
  - Normally the ventilation airflow is a constant
    - Enthalpy difference determines load:
      - 4.5 * ventilation cfm * enthalpy difference between outside air and coil leaving air
  - Determining leaving air design state is the key

Applied Air

D.O.A.S.

Design Steps

- In determining the required coil leaving air temperature for the DOAS:
  - Design for coil leaving ventilation air that is *drier* than the space humidity target
  - Design to handle the ventilation air sensible and latent load *plus* the space latent load
- Look for the worst humidity case for the outside air:
  - This will likely ***not*** be at the peak DB temperature for the location
  - The engineer must look at the peak dewpoint and the mean coincident peak WB for the highest enthalpy condition

D.O.A.S.

Design Steps

There are seven basic steps to determining the DOAS coil load and ventilation air requirement:

1. Determine outside air enthalpy conditions based on the project location.
2. Determine the maximum allowable space humidity by asking the end-user or using ASHRAE recommended levels.
3. Determine the space latent loads by determining the number of occupants and applying the ASHRAE latent load factors based upon their activity levels. Add any unusual latent machine contributions, if any.
4. Determine the required ventilation cfm by using the formulas in ASHRAE Standard 62.1-2004.
5. Determine the required dewpoint and enthalpy of the supply air.
6. Calculate the coil load as described previously.
7. Determine the required leaving air temperature by deciding if this will be a “neutral” or “cold air” system.

D.O.A.S.

Design Steps

Step four requires an understanding of ASHRAE Standard 62.1-2004:

The formula for calculating the amount of outside ventilation air uses both an occupancy factor and an area factor modified by a ventilation effectiveness factor that is based upon the air distribution system type.

$$V_{oz} = (R_p*P_z + R_a*A_z) / E_z$$

where:
V_{oz} = ventilation air cfm
R_p = cfm/person
P_z = maximum occupancy of the zone
R_a = cfm/sq.ft.
A_z = area of the zone
E_z = ventilation effectiveness factor

The ventilation effectiveness factor is typically 1.0 for cooling systems that use ceiling diffusers and 0.8 for heating systems that use ceiling diffusers. Systems that use sidewall supply systems or underfloor systems will have a different Ez that should be looked up in Table 6.2 of the Standard.

Applied Air

D.O.A.S.

Design Steps

The rate factors for calculating the amount of outside ventilation air can be found in Table 6.1 of Standard 62.1-2004 but some of the more common factors are shown below:

Occupancy category	Rp Cfm/person	Ra Cfm/sq.ft.
Offices	5.0	0.06
Classrooms (Elementary)	10.0	0.12
Lecture Classrooms	7.5	0.06
Retail sales	7.5	0.12
Auditoriums	5.0	0.06

D.O.A.S.

Design Steps

Step five is the most difficult step:

5. Determine the required dewpoint and enthalpy of the supply air.

In order to perform this step you will need to know the most stringent target humidity condition for the occupied zones. This will be provided by the occupants or by ASHRAE recommendations.

The formula that is used is constructed to calculate grains of moisture in the air. One of the factors is a number that varies with altitude but is worst at sea level. Using this factor for anything above sea level will result in only slight over-sizing of the DOAS coil.

D.O.A.S.

Design Steps

Step five is the most difficult step:

The formula is:

.69 * outside air cfm * target indoor grains/lb *less* the space latent load in btuh divided by .69 times the outside air cfm

$$\frac{(.69 \times cfm \times grains) - \text{space latent load*}}{(.69 \times cfm)}$$

The result will be presented in grains/lb. You can then use one of the readily available psychrometric calculators to determine the dewpoint temperature and enthalpy that corresponds to this temperature at saturation. This temperature becomes your *maximum* coil leaving air temperature.

Applied Air

D.O.A.S.

Design Steps

- Example:
  - 30’x40’ classroom with 30 students, 74FDB/60%RH design; 91FDB/79FWB peak enthalpy point from ASHRAE tables; 29,753 btuh sensible/5,250 btuh latent
    1. Determine entering air conditions based on location*
    2. Determine the maximum allowable space humidity*
    3. Determine the space latent loads*
    4. Determine the required ventilation cfm*
    5. Determine the required dewpoint and enthalpy of the supply air
    6. Determine the required leaving air temperature
    7. Calculate the coil load

Applied Air

D.O.A.S.

Design Steps

- Example:
  - 74FDB/60%RH design; 91FDB/79FWB peak enthalpy point (42.49 btu/lb); 29,753 btuh sensible/5,250 btuh latent *(latent load about 175 btuh/person)*

Steps 1, 2, and 3

  - Determine vent air quantity using factors from ASHRAE Standard 62.1-2004 Tables 6.1 and 6.2:

Step 4

    - (30 students * 10 cfm/person + 0.12 * 1200 sq.ft.) / 1.0 = 444 cfm ventilation air (rounded to 450)
  - Determine the max OA moisture content and corresponding dewpoint temperature:
    - Classroom air humidity ratio 75.2 gr/lb (74/60%)

$$\frac{(.69 \times 450 \times 75.2) - \text{space latent load*}}{(.69 \times 450)}$$

Step 5

    - Vent air max humidity ratio equals 58.3 gr/lb
    - 58.3 gr/lb equals 52 F dewpoint temp
    - 52 F air at saturation has an enthalpy of 21.43 btu/lb

D.O.A.S.

Design Steps

- Example:

Step 6

- OA load equals 42,646 btuh as calculated below:

4.5 x cfm x enthalpy difference (outside enthaply-dewpoint enthalpy)
4.5 x 450 x (42.49-21.43)

Step 7

- In order to supply "neutral air" we must reheat this 52 F air up to room conditions of 74 F, or leave it at 52 F.

- If we choose to use "neutral air" the reheat load equals 10,742 btuh as calculated below:

1.085 x cfm x db temperature difference (leaving air temp – dewpoint temp)

1.085 x 450 x (74-52)

D.O.A.S.

Design Steps

- Example:
  - Remember that "neutral air" does nothing to reduce the space sensible load of 29,753 btuh therefore:
  - Total load for the parallel system is 29,753 btuh
  - Total load for DOAS unit is 42,646 + 10,742 = 53,388 btuh
  - The DOAS unit can be controlled by discharge air temperature to maintain 74 degrees F

D.O.A.S.

- Design Steps
  - Example:
    - In the “cold air” case the outside air load is the same (but without reheat).
    - The difference is the direct reduction of space sensible load when the cold air is supplied directly to the space *or* a reduced mixed air temperature (MAT) at the parallel system if the cold air is supplied to the parallel system return air side.

Applied Air

D.O.A.S.

- Design Steps
  - Example:
    - With “cold air” the space sensible load is reduced if the air is introduced directly into the space
    - The parallel unit must handle:
      - Space sensible load – (1.085 x OaAcfm x (Space temp – dewpoint temp))
      - In this example: 29,753 – (1.085 x 450 x (74-52)) = 19,011 btuh load on the parallel system
    - By leaving the parallel system cfm at 1,500 in spite of the reduced sensible load the result is:
      - The parallel system supply air temperature should be:

        Space temp – (19,011/(1.085 x 1500)) or, 62.3 F

D.O.A.S.

- Design Steps
  - Example:
    - With “cold air” the space sensible load is not reduced if the air is introduced into the parallel system return air side...but the mixed air temperature is.
    - The mixed air temp will be on the line between the room condition and the dewpoint condition and can be calculated as:

$$\frac{(450 \times 52) + (1050 \times 74)}{450 + 1050}$$

      - or, 67.4 F

      - The parallel system discharge temp should be:

Space temp – (29753/(1.085 x 1500)) or, 55.7 F

    - The parallel system coil load is now 1.085 x 1500 x (67.4-55.7) = 19,042 btuh

Applied Air

D.O.A.S.

Summary

1. Determine outside air enthalpy conditions based on the project location.
   - Look up data in ASHRAE Handbook of Fundamentals weather data tables for city and state of project

2. Determine the maximum allowable space humidity by asking the end-user or using ASHRAE recommended levels.
   - User either 65% from ASHRAE 62.1-2004 or ask user for requirements

Applied Air

D.O.A.S.

Summary

3. Determine the space latent loads by determining the number of occupants and applying the ASHRAE latent load factors based upon their activity levels.
   - Look up latent load of occupants in ASHRAE Handbook...typically about 175 btuh/person

4. Determine the required ventilation cfm by using the formulas in ASHRAE Standard 62.1-2004.
   - (Rp*Pz+Ra*Az)/Ez
   - *Rp, Ra, and Ez are found in ASHRAE 62.1-2004*

Applied Air

D.O.A.S.

Summary

5. Determine the required dewpoint and enthalpy of the supply air.
   - *(.69xcfmxgrains)-space latent load**

     (.69xcfm)
   - *Use psych chart to find this point*

6. Calculate the coil load as described previously.
   - *4.5 x cfm x enthalpy difference across coil*

7. Determine the required leaving air temperature by deciding if this will be a "neutral" or "cold air" system.

Applied Air

D.O.A.S.

Summary

- Key decision factors
  - Ease of mixing the air
  - Simplicity of installation
  - Neutral air to the space might require a lower supply temp from the parallel system
  - Neutral air to the parallel system requires re-cooling the air that we just re-heated
  - Cold air to the space will likely result in drafts and dumping from diffusers
  - Cold air to the parallel system acts like a booster to the total system and can reduce the required size of the parallel system units.

Applied Air

D.O.A.S.

- Summary
  - Neutral air to the space might require a lower supply temp from the space HAC unit
  - Neutral air to the HAC unit requires re-cooling the air that we just re-heated
  - Cold air to the space will likely result in drafts and dumping from diffusers
  - Cold air to the unit acts like a booster to the total system and can reduce the required size of the space HAC units...this is probably our preferred choice.

Applied Air

Climate Control Systems

Applied Air

www.appliedair.com

4830 Transport Drive Dallas TX 75247

Telephone 214.638.6010

Fax 214.905.0806