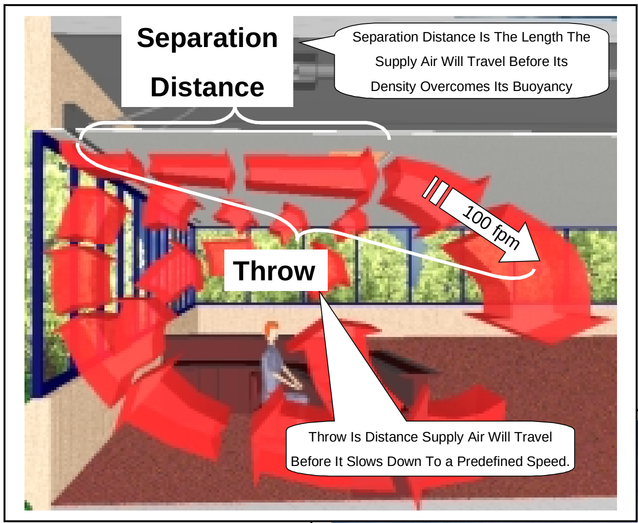
# Optimal Air

# Design Manual

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# Introduction

## Why Use Low Temperature Air?

While low temperature air distribution systems are not new, many engineers, architects and building owners/developers are becoming interested in the technology as greater emphasis is placed on reducing building construction costs and equipment room and ceiling plenum space.

Conventional supply air temperatures run between 54°F -57°F from the air handling unit. With duct heat gain, the supply air ranges from approximately 56°F-59°F out of the air diffuser. In low temperature air systems, supply air temperatures range from 39°F-52°F. This document outlines the benefits of what is termed Optimal Air systems and provides design guidelines.

## Optimal Air versus Low Temperature Air Systems

Reduced supply air temperature systems are normally separated into two broad categories:

### Optimal Air

- ❑ 45-52°F from the air handling unit to optimize energy consumption, reduce first capital cost and improve humidity control. Optimal Air has for years been extensively used in grocery stores and is gaining increasing popularity in comfort cooling applications such as offices and schools. Optimal Air systems do not use more energy than conventional systems on an annual basis.

### Low Temperature

- ❑ 39-42°F from the air handling unit to minimize duct and air handling unit sizing. Most often used with ice storage systems to take advantage of the cold air produced by these systems. Low temperature systems do use more energy annually than conventional systems. However, by taking advantage of time of use utility rates (i.e. at night and on weekends), they can actually cost less to operate. Low temperature air/ice storage systems are a topic by themselves and are not covered in this Design Manual.

## Benefits of Optimal Air Systems

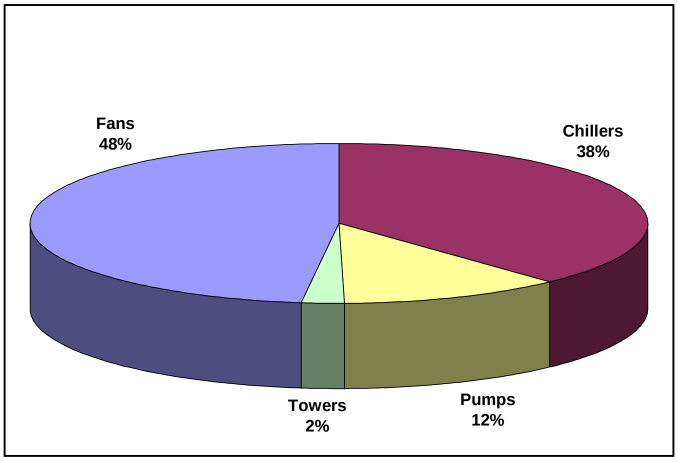
There are several benefits of Optimal Air that make it an attractive system for use in a wide variety of applications.

### Optimal Air Saves Space and Reduces Energy and Construction Costs

Optimal air increases the amount of sensible heat that each cfm delivered to a zone can absorb. While 50°F air may not seem much colder than 55°F air, the delta T rises from 20°F to 25°F. That is a 25% increase!

This affects the sizing of the ducts, air handling units and fan motors – all of which will be smaller – and results in a system that requires less space and uses less power. In many applications, fans can use more power annually than refrigeration (chillers, condensing units etc.). For example, Figure 1 shows the annual energy usage of HVAC components for a 10 story, 200,000 $ft^2$ office building located in Chicago. Building fan energy use is high because the fans operate every hour the building is occupied providing minimum air movement, ventilation air, heating, etc. In this case, an Optimal Air system would have a very real impact on overall energy costs.

*Figure 1 - Annual HVAC Energy Usage*




## Optimal Air Reduces Humidity, Improves Comfort

Optimal Air systems take more moisture out of the return and ventilation air mixture as it passes over the cooling coil. The lower moisture content in the supply air reduces the "psychrometric balance point" humidity level in the conditioned space. This allows the space temperature to be set higher while achieving the same comfort level for occupants and further reduces the supply air quantity and fan power requirement.

When coupled with the constant supply air temperature from variable air volume control, the low moisture content of Optimal Air helps maintain low space relative humidity at all operating conditions (from part load to full load). An Optimal Air system is ideal for humidity sensitive applications such as grocery stores, "super centers," retail stores, theaters, plastics molding plants, printing plants, etc. as well as for an increasing number comfort cooling applications in office building, schools, etc.

## Optimal Air Systems are Quieter and Improve Indoor Air Quality (IAQ)

The lower air volume required for Optimal Air systems makes them quieter than conventional systems. Fan sound generation is a function of fan type, static pressure and air volume. By reducing air volume (and often the total fan static pressure) Optimal Air systems generate lower fan sound which can result in more desirable space conditions. This reduced sound generation can also be used to reduce the cost of any required noise attenuation in critical applications.

The lower required air volume can also be used to reduce filter face velocities, allowing more efficient filters to be used without high energy cost penalties. The lower air temperature and resultant humidity levels also reduce the chance of mold growth in the air handling units, ducts or the occupied space.

# Optimal Air Systems

Using McQuay's Energy Analyzer™ modeling program, we have developed a model of an office building in Chicago to understand the changes that take place in an HVAC system as the supply air temperature is lowered. The office building is 10 story, 200,000 $ft^2$ with an outdoor air ventilation requirement of 26,667 cfm. The HVAC system is floor by floor VAV air handling units with a two chiller primary secondary system. Please note that this model has been developed for illustration purposes only. Optimal air works equally well with applied rooftop units or indoor vertical self-contained units.

***Table 1 - HVAC System Performance As Supply Air Temperature Is Lowered***

| SAT °F | TSP inches w.c. | Supply Air Volume cfm | Design Cooling Tons | CWST °F | Perform-ance kW/ton | Room Setpoint °F | Chiller Work kWh/yr | Fan Work kWh/yr | Total Work kWh/yr |
|---|---|---|---|---|---|---|---|---|---|
| 55 | 3.00 | 152,686 | 473 | 44.0 | 0.550 | 75 | 219,605 | 202,736 | 422,341 |
| 54 | 3.02 | 145,415 | 475 | 43.6 | 0.554 | 75 | 226,594 | 194,061 | 420,655 |
| 53 | 3.04 | 138,805 | 477 | 43.2 | 0.558 | 75 | 236,352 | 186,175 | 422,527 |
| 52 | 3.06 | 132,770 | 479 | 42.8 | 0.562 | 75 | 241,228 | 178,975 | 420,203 |
| 51 | 3.08 | 127,238 | 481 | 42.4 | 0.566 | 75 | 248,661 | 172,375 | 421,036 |
| 50 | 3.10 | 122,149 | 483 | 42.0 | 0.570 | 75 | 255,777 | 166,303 | 422,080 |
| 49 | 3.12 | 117,451 | 485 | 41.6 | 0.574 | 75 | 261,285 | 160,698 | 421,983 |
| 48 | 3.14 | 113,101 | 486 | 41.2 | 0.578 | 75 | 269,044 | 155,508 | 424,552 |
| 47 | 3.16 | 109,061 | 488 | 40.8 | 0.582 | 75 | 276,512 | 150,689 | 427,201 |
| 46 | 3.18 | 105,301 | 490 | 40.4 | 0.586 | 75 | 286,605 | 146,202 | 432,807 |
| 45 | 3.20 | 101,791 | 492 | 40.0 | 0.590 | 75 | 292,832 | 142,014 | 434,846 |

Table 1 shows the HVAC system performance as the supply air temperature, to the duct, is lowered. It is important to differentiate between supply air temperature off the cooling coil and supply air temperature into the duct. This issue is covered in detail in Blowthrough Vs. Drawthrough, page 13. This Design Manual refers to supply air temperature as the temperature supplied into the duct.

To accommodate the lower supply air temperature, the chilled water supply temperature (CWST) was gradually lowered, the air handling unit coils deepened to allow for closer approaches, and chiller performance was adjusted to deal will the increased lift. Because of their basic operating differences, DX rooftop and self-contained systems may have a different Optimal Air temperature than a chilled water system. When considering multiple system options, it is important to use Energy Analyzer for each in order to identify the best option.

## Fan Energy and Duct Sizing

As expected, Table 1 shows that as the supply air temperature is reduced, fan energy requirements are reduced because less air is handled to produce the same cooling effect in the space. Conversely, the refrigeration energy increases to accommodate the lower CWST requirement. A 10°F drop in supply air temperature resulted in a 33% drop in supply air volume and a 30% drop in annual fan work. Please note that the annual fan work could be further reduced if coil and filter velocities were allowed to fall (versus reducing air handler size), reducing fan total static pressure.

For new construction, the ductwork sizing could be optimized for the reduced air volume because it would involve the lowest capital cost and the total static pressure would be approximately the same for any given supply air temperature. The static pressure in Table 1 was gradually increased to account for deeper coils that can accommodate closer approaches. For retrofit applications where the ducting is re-used, reducing the air volume will have the double effect of reducing the actual required cfms and significantly reducing the required static pressure.

## Refrigeration

Refrigeration annual energy requirements increase for three reasons. First, the refrigeration must operate at somewhat lower suction temperatures to produce the lower leaving air temperatures. Second, greater cooling effort must be expended on ventilation air to cool and dehumidify from outside ambient conditions to the lower supply air conditions. Finally, more refrigeration operating hours are required per year because the number of hours where the economizer can reduce the cooling load are reduced. This is due to the lower supply air temperature. A 10°F drop in supply air temperature resulted in a 25% increase in chiller plant work.

In Table 1, the actual building tonnage climbed slightly as the supply air temperature was lowered. There are two opposing issues affecting design capacity. While the smaller air handling fan motors result in a drop in the overall design sensible cooling requirement, the lower design supply air temperature increases the latent load. A job by job analysis needs to be performed to determine the affect on design cooling capacity. As can been seen in Table 1, the change is less than 5%.

## Optimal Air Balance Point

Reduced fan energy must be "traded off" against increased refrigeration energy. This trade off varies with the type of building, the type temperature control system, the type air conditioning system and geographic locale. Therefore, the "optimal" supply air temperature is different for every job. When only energy costs are a factor and no thermal storage is involved, this optimal supply air temperature generally falls in the 47°F -52°F range. It can be determined by comparing total system energy consumption with varying supply air temperatures using an energy analysis program such as McQuay's Energy Analyzer™.

***Figure 2 - SAT vs. Total HVAC Work***

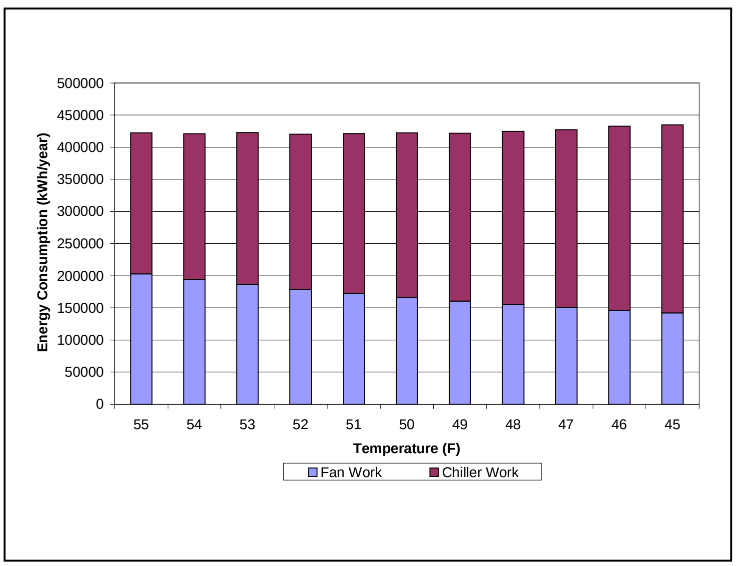


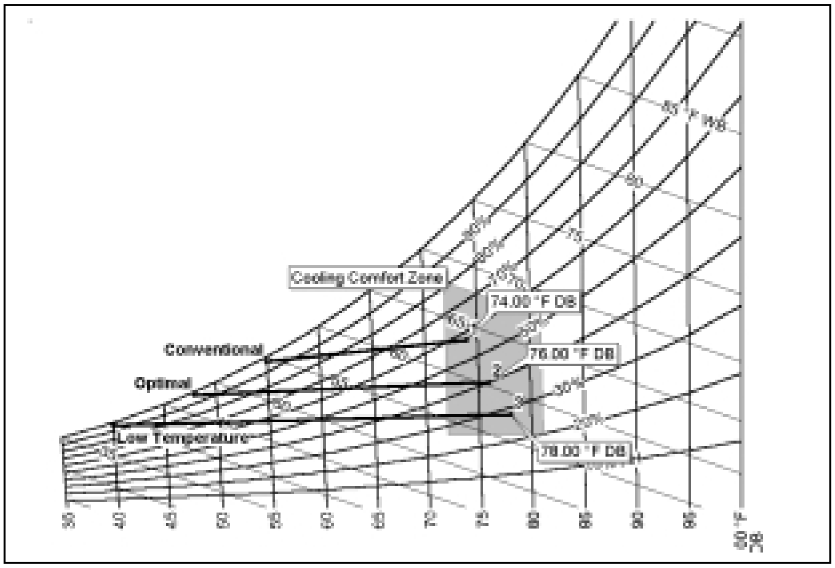

Figure 2 shows the total annual HVAC work (in kWh/yr) versus supply air temperature for the Chicago office building example. In this example, the balance point is 49°F. Below this temperature, the annual work starts to increase as the chiller work overtakes the fan savings.

Designing with supply air temperatures lower than the balance point provides capital savings at the expense of increased operating costs. Up to the balance point, the capital cost declines without any penalty on operating cost.

## Space Design Temperature and Related Comfort

Temperature, humidity, air velocity and mean radiant temperature directly influence occupant comfort. Conventional designs are usually based on maintaining 75°F and 50% RH (Relative Humidity) in the occupied space. Figure 3 shows the ASHRAE comfort zone where 80% of the people engaged in light office work are satisfied. As the relative humidity is lowered, the space air temperature can be raised and still provide occupant comfort. ASHRAE Standard 55, Thermal Environmental Conditions For Human Occupancy covers space conditions in detail.

*__Figure 3 - Equivalent Comfort Chart__*




The leaving air condition from the air handling unit. is the primarily control of the relative humidity in the occupied space. The internal moisture gains from people, kitchens, etc, as well as infiltration also play a part.

In most climates, the lower the supply air temperature, the lower the humidity ratio and the drier the space. Figure 3 shows sensible heat ratio lines for conventional, Optimal and low supply air temperatures. As the space relative humidity is lowered, the space temperature setpoint rises from 74°F to 78°F.

*__Table 2 – Optimal Air System Performance With Varying Space Temperature Setpoint__*

| SAT °F | TSP inches w.c. | Supply Air Volume cfm | Design Cooling Tons | CWST °F | Perform -ance kW/ton | Space Setpoint °F | Chiller Work kWh/yr | Fan Work kWh/yr | Total Work kWh/yr |
|---|---|---|---|---|---|---|---|---|---|
| 49 | 3.12 | 117,451 | 485 | 41.6 | 0.574 | 75 | 261,285 | 160,698 | 421,983 |
| 49 | 3.12 | 112,549 | 480 | 41.6 | 0.574 | 76 | 255,231 | 153,695 | 408,926 |
| 49 | 3.12 | 107,997 | 475 | 41.6 | 0.574 | 77 | 249,350 | 147,438 | 396,788 |

Using the Chicago office building example,

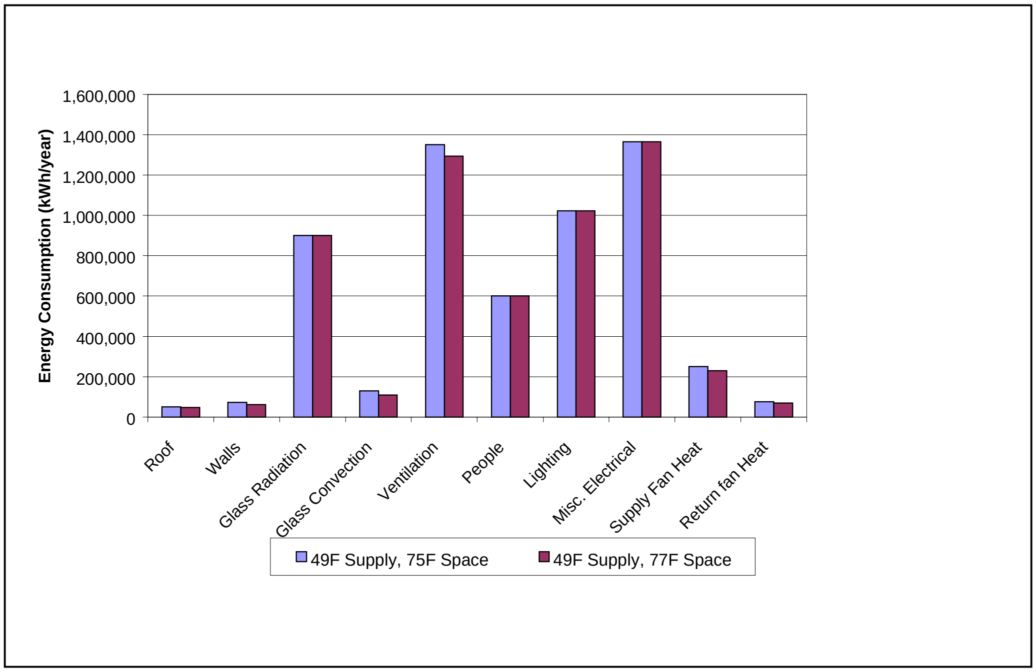
Table 2 shows the effect of raising the space temperature setpoint within a constant supply air temperature. Raising the space setpoint again increases the sensible cooling effect each “cfm” of supply air can provide. Each degree the space temperature setpoint is raised, reduces the supply air volume by almost 4%. The result is smaller ducts, air handling units and fan motors.

*__Table 3 -Room Temperature vs. Supply Air Temperature__*

| Supply Air Temperature | Constant Comfort Summer Room Design Temperature | Constant Comfort Winter Room Design Temperature |
|---|---|---|
| 54-56°F | 75°F | 70°F |
| 45-49°F | 76-77°F | 72°F |
| 39-42°F | 78°F | 74°F |

Raising the space temperature setpoint also improves the refrigeration work. Smaller fans lower the design sensible-cooling requirement, and the smaller design temperature gradient difference between space and ambient conditions also lowers the design sensible-cooling load. The result is a smaller chiller plant and lower annual chiller work. Figure 4 shows the difference in design cooling loads at 75°F and 77°F space conditions.

*Figure 4 - Design Cooling Loads At 75°F and 77°F Space Conditions*



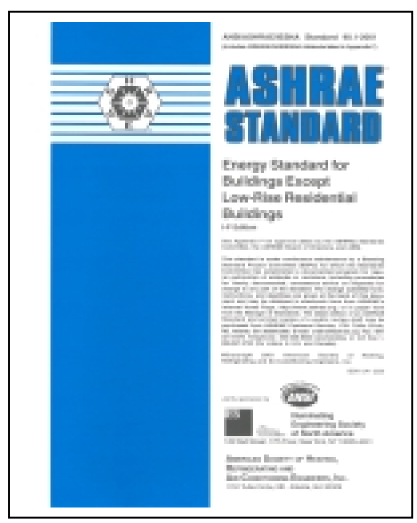

## Sound Considerations

Optimal Air solutions are also used to improve sound levels within the building because their reduced air volume, reduced fan static pressure and reduced fan power requirements can significantly reduce overall fan sound generation. Lower sound energy released into the duct system, can result in the need for less attenuation to achieve desired sound levels. In addition, the smaller air volumes required to cool the space can reduce the sound generation from diffusers and terminal devices. These advantages can be very important in sound critical applications such as theaters.

## ASHRAE Standard 90.1 Compliance

The 1999 and 2001 version of ASHRAE Standard 90.1, Energy Standard for Buildings Except Low Rise Residential Buildings, have mandatory requirements for refrigeration equipment and prescriptive requirements for fan work.

The Standard recognizes that Optimal Air systems improve fan work significantly and provides credits to account for improved fan performance. In addition, refrigeration system performance is rated at conventional conditions or special tables are provided to account for non-standard operating conditions (as is the case with centrifugal chillers). In either case, ASHRAE Standard 90.1 does not penalize Optimal Air systems.




## Indoor Air Quality

Most design engineers follow the procedures outlined in ASHRAE Standard 62.1 to provide acceptable indoor air quality (IAQ). Using Optimal Air does not require changing this approach.

# Design Considerations

## General

Design of refrigeration and air handling equipment for an Optimal Air system is similar to the design of a conventional air temperature system. Attention must be paid, however, to air distribution, controls and duct design. Conventional diffusers, when properly applied, will work with Optimal Air. Controls also require only minor changes from conventional systems. In particular, programming of economizer controls and supply air temperature reset. Finally, the ducting system must be sized for the reduced air volume to take full advantage of the potential capital savings. Duct insulation and sweating should also be reviewed to provide a trouble free system.

## Applications

Not every building type is a good candidate for Optimal Air. When air volumes are dictated by air turnover rates, such as some health care applications, Optimal Air offers no advantage. In fact, there would be increased reheat costs. Office buildings are a strong candidate for Optimal Air. They have high sensible heat ratios and typically less than 20% ventilation loads. Schools can also be a possibility. Generally speaking, as the percentage ventilation load increases, Optimal Air becomes less attractive.

Location and climate also impact whether or not Optimal Air is a good candidate. Locations where weather provides significant economizer hours between 45 and 55°F will limit the savings. Ultimately, each project must be checked by performing the calculations described in the next section.

## Building Envelope

The building envelope is key to controlling infiltration which may lead to duct sweating. Infiltration is a larger concern in humid climates such as the coastal southeastern United States. Most modern building construction is adequate for Optimal Air applications. However, care should be taken in retrofit applications where the building envelope may be leaky or of unknown quality.

## Load Calculations

To evaluate Optimal Air systems requires annual energy load calculations. Software such as McQuay's Energy Analyzer™ can be used to quickly identify the balance point and perform life cycle analysis. Energy Analyzer ™ can be used at the earliest stages of the project prior to any design load calculations being prepared. Most buildings can be evaluated in less than an hour.

### Calculate the Balance Point

The first goal is to quickly identify how much the supply air temperature can be lowered before the operating costs begin to increase. This cannot be "estimated" from full load performance. It must be based on annual energy usage. Issues such as part load performance, equipment sequencing and hourly weather must be considered. A table of annual HVAC energy usage and HVAC energy cost vs. supply air temperature should be developed (See Table 1 - HVAC System Performance As Supply Air Temperature Is Lowered, for an example).

As the supply air temperature is lowered, several assumptions must be made with regard to system design. At conventional temperatures, the supply air temperature is about 55°F and the chilled water (if chillers are used) is approximately 44°F. As the supply air temperature drops, it will be necessary to reduce the approach (Supply air temperature – chilled water temperature). This would require deeper coils and result in a larger air pressure drop. Although every job is different, the following is a starting point:

- ❑ Perform analysis for a supply air temperature range from 55°F to 45°F.
- ❑ Increase the supply fan total static pressure 0.02” w.c. per degree F decrease. This will result in a 0.2” w.c. increase by 45°F or about 2 rows on a typical coil. This does not apply to applied rooftop and vertical self-contained systems that use 5 or 6 row coils regardless of temperature.
- ❑ Decrease the chilled water setpoint 0.4°F per degree F. This will result in 40°F supply water temperature by 45°F.
- ❑ Increase the full load chiller performance by 0.004 kW/ton per degree F. This will result in a 0.04 kW/ton increase by 45°F.
- ❑ Increase the full rooftop performance by 0.006 kW/ton per degree F. This will result in a 0.06 kW/ton increase by 45°F.
- ❑ Increase the full load vertical self-contained performance by 0.004 kW/ton per degree F. This will result in a 0.04 kW/ton increase by 45°F.
- ❑ Hold the space temperature setpoint constant at 75°F.

Review the data and use the lowest supply air temperature that doesn’t significantly increase the annual operating cost of the building.

### Evaluate Space Temperature Setpoint

As demonstrated in the previous section, raising the space temperature setpoint offers further capital and operating cost savings. To evaluate the possible savings, further calculations are required.

- ❑ Use the inputs for the selected supply air temperature from the first calculations.
- ❑ Perform additional selections each time increasing the space temperature setpoint one degree F to a maximum of 78°F.

Whether to take the additional savings will depend on good engineering judgment regarding many factors (local weather, building construction quality etc) and the understanding and support of the building owner.

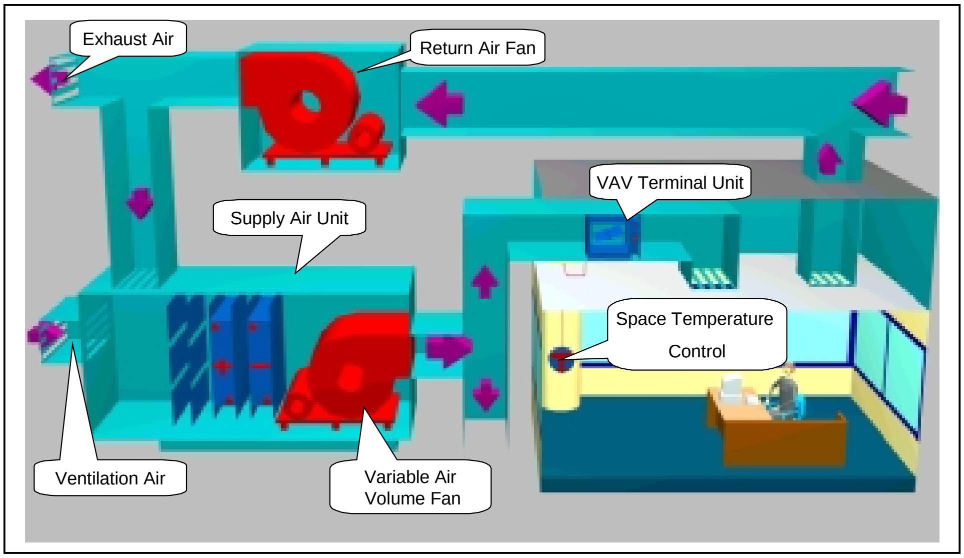
### Design Load Calculations

Once the supply air temperature and space temperature are established, then the balance of the design is similar to a conventional HVAC system design. Use the Optimal Air temperatures and complete the space design loads and air volume calculations. Fine tuning the equipment selections can offer further savings. Take advantage of specific model selections that offer improvements to static pressure and supply air temperature. Have your McQuay representative provide preliminary equipment selections as soon as the cooling load calculations are complete and take full advantage of specific equipment selections.

## Secondary System Selection

In most cases, some form of Variable Air Volume (VAV) is desirable for the secondary system selection (Figure 5). VAV systems are sized on the design load – not the connected load – which offers capital savings through smaller ducting, air handling units and fan motors. Since the air volume varies with the load, there are substantial fan power savings at part load. Another advantage of VAV is good humidity control at part load conditions. The supply air is always dehumidified to the saturation point of the supply air temperature setpoint. Constant volume systems reset the supply air temperature setpoint up at part load so the space is not over cooled. The result is poor humidity control.

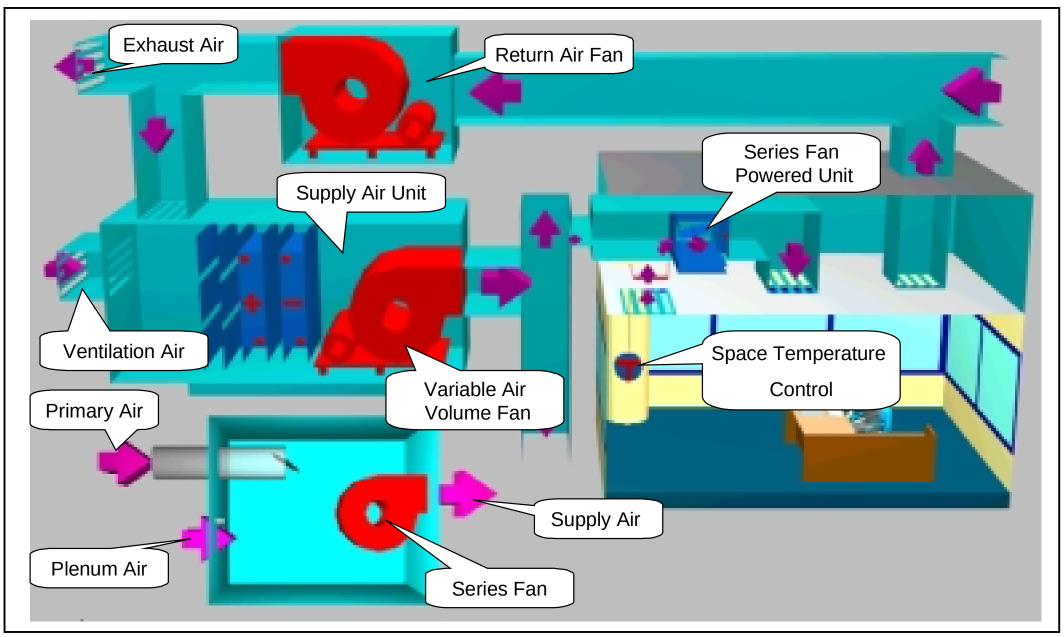
*Figure 5 - Typical VAV System*




## Fan Powered Boxes

A variation of VAV is either series or parallel fan powered boxes. Figure 6 shows a typical series fan powered box system. The fans in series boxes operate all the time and provide constant air volume to the space. The temperature supplied to the space is varied by mixing primary air with return air as required to meet the cooling load.

*Figure 6 - Series Fan Powered VAV System*




Advantages to Series fan powered boxes are:

- The supply air temperature to the space does not have to be the Optimal Air temperature. It is possible, even at design load conditions, to introduce plenum air into the series box so that 55°F enters the space even though 49°F primary air was distributed through the building.

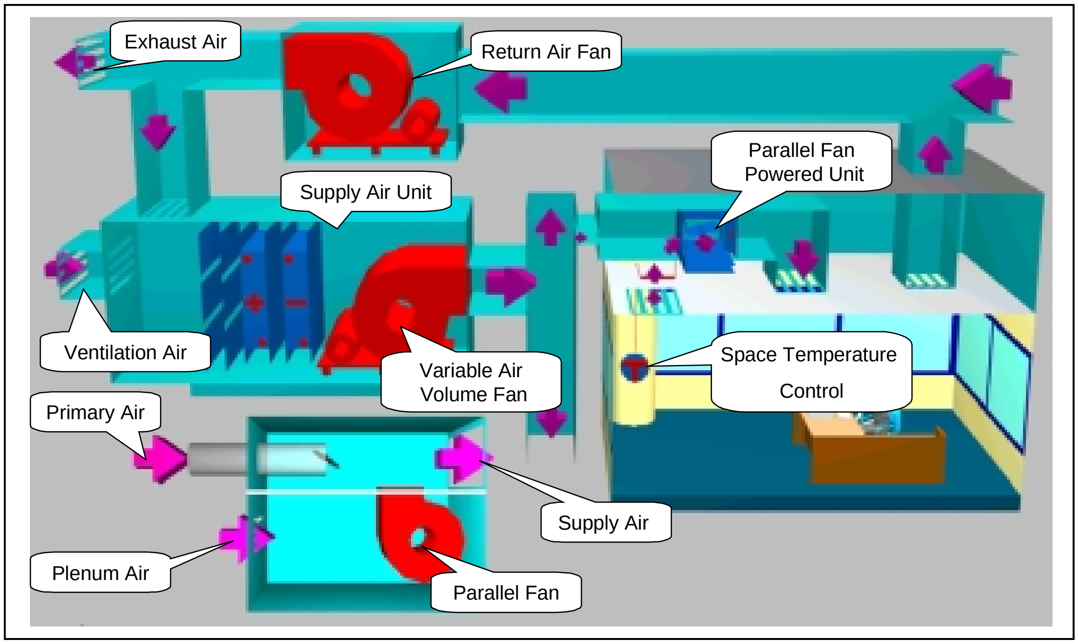
- Constant air volume to the space ensures proper diffuser operation over all operating conditions.
- Plenum air is used as reheat which is a form of heat recovery.
- Overhead heating is easily done.
- During unoccupied hours, only a local series box needs to be started to heat or cool a zone as opposed to starting up the main air handling unit with standard VAV.

Disadvantages include:

- The capital cost of the boxes
- The operational cost of the fans in the boxes
- Sound
- Maintenance

Figure 7 shows a typical parallel fan powered box system. The fans in parallel boxes operate only during heating mode. In cooling mode, the box behaves like a typical VAV box. In the heating mode, the fan operates providing constant volume. The temperature is varied by mixing plenum air with primary air to meet the space conditions or by a heating coil mounted in the box.

***Figure 7 - Parallel Fan Powered VAV System***




Advantages to parallel fan powered boxes are:

- Box fans only operate when necessary.
- Constant air volume to the space during heating ensures proper diffuser operation.
- Plenum air is used as reheat which is a form of heat recovery.
- Overhead heating is easily done.
- During unoccupied hours, only a local series box needs to be started to heat or cool a zone as opposed to starting up the main air handling unit with standard VAV.

Disadvantages include;

- The capital cost of the boxes
- The operational cost of the fans in the boxes.
- Sound. In particular, the starting and stopping of the fans in the ceiling plenum adds an additional challenge.
- Maintenance.

## Mixing VAV and Fan Powered Boxes

HVAC systems do not need to be all VAV or all fan powered boxes. A common approach is to use series fan powered boxes on the perimeter and perhaps the top floor where heating will occur. Conventional VAV boxes can then be used throughout the core.

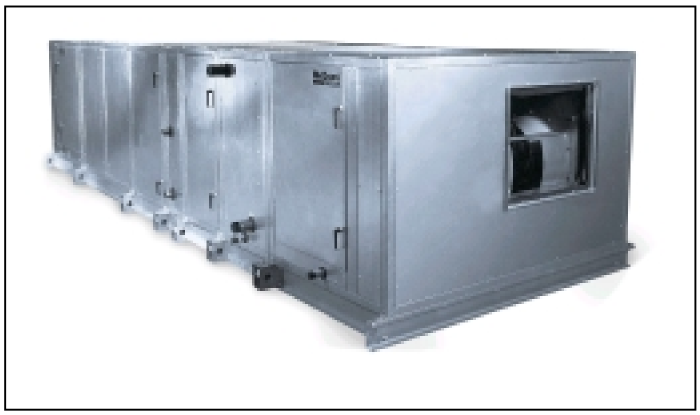
## Perimeter Heating

Optimal Air systems have the same requirements for perimeter heating that a conventional temperature system would have. Appendix 1: Perimeter and Skin Heating Systems covers common solutions for perimeter heating.

## Component Construction

Equipment used in Optimal Air systems must be designed and built for the application.

***Figure 8 - McQuay Vision AHU Designed For Optimal Air***



Air handling units should have additional insulation and thermal breaks where necessary. This applies for indoor air handling units as well as the air handling portion of a rooftop or vertical self-contained unit. Casings with 2” insulation and liners should be considered for air handlers located in untreated ambient air. The additional condensate created by the lower building RH requires properly sloped drain pans and correctly installed traps. Stainless steel drain pan construction should always be considered.

Coil face velocity is also important. The coils are inherently deeper and the latent load is higher in Optimal Air temperature air systems. Care should be taken to avoid condensate carryover. Design face velocities of 400 to 550 fpm are recommended. Coil height is also important. Condensate builds up on the coil face as it runs down. The taller the coil, the greater likelihood of condensate being blown off the coil. Coils should not be taller than 54 inches unless an intermediate drain pan is provided between coils. The minimum face velocity value can be as important as the maximum value especially in DX system. In VAV systems, too low an initial value can lead to poor coil performance at light loads, causing coil frosting and performance issues.

VAV and fan powered boxes should have primary air valves thermally isolated from the casing, non-conducting damper shafts and special insulation.

## Blowthrough Vs. Drawthrough

Blowthrough units have the fan upstream of the cooling coils. Fans add enough heat to raise the supply air temperature about 3°F. By adding the fan heat first, the supply air temperature entering the duct is the same as the leaving air temperature from the coil. In addition, the log mean temperature difference (LMTD) (entering air temperature is higher) of the coil is raised, improving coil and refrigeration performance. Finally, the relative humidity of the air entering the coil is lower, resulting in a higher sensible heat ratio.

Drawthrough units add the fan heat after the cooling coil, raising the supply air temperature in the duct by about 3°F. For a given coil leaving air temperature, this results in significantly more supply air (10% more air at conventional conditions) being required for the same cooling load. To achieve the same supply air temperature in the duct, a drawthrough system has to have a coil leaving air temperature approximately 3°F cooler than the blowthrough system. Lowering the air temperature from the coil in a drawthrough system to provide the same supply air temperature as provided by a blowthrough system will lower the space RH, increase the refrigeration work and decrease the hours available for economizer work. Since Optimal Air systems already provide lower space RH, lowering it further with a drawthrough system is not an advantage.

The advantage of blowthrough systems is typically less supply air (almost 10%) is required for the same cooling load. Less fan work is used annually to cool the building. The disadvantage is the supply air can be saturated (100% RH), while in drawthrough systems the fan heat has raised the temperature and lowered the relative humidity. Saturated air may lead to microbial growth in the ductwork.

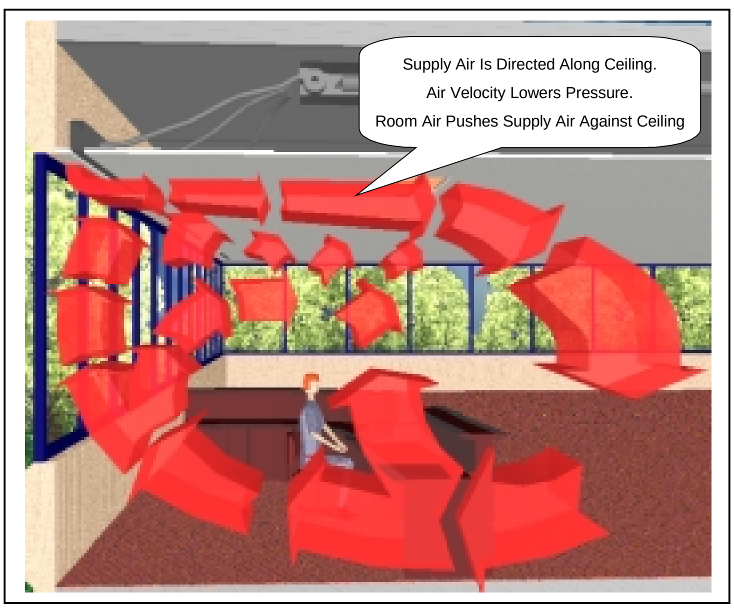
Applications with high sensible heat ratios, such as offices, are good candidates for blowthrough systems. Energy Analyzer™ can be used to estimate the reduction in supply air volume and the annual energy savings by comparing drawthrough to blowthrough. In many cases, blow through is a win-win (lower capital cost-lower operating cost) situation.

# Air Distribution

Optimal Air requires a little more careful analysis of air distribution for the system to work properly and provide thermal comfort to the occupants. As the supply air temperature is lowered, the challenge increases.

There are five main types of diffusers readily available. These are slot, radial, jet, swirl and perforated. Any of these can be made to work with Optimal Air systems as long as they meet the requirements. Slot, radial and jet diffusers are the most common.

***Figure 9 - Typical Slot Diffuser***

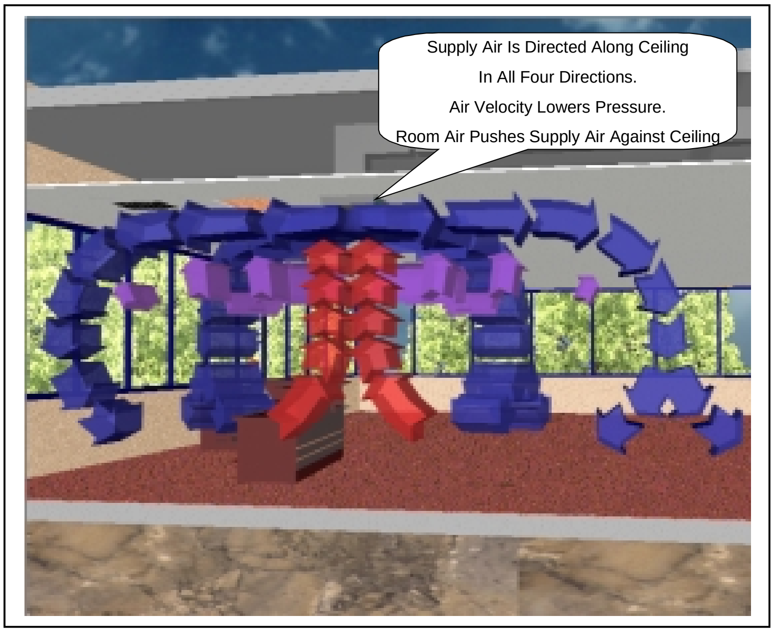



## Slot Diffuser

Slot diffusers are sometimes called linear diffusers. The diffuser can throw the air one-way or two-way, perpendicular to its length. Very little primary air is thrown parallel to its length.

## Radial Diffuser

***Figure 10 - Typical Radial Diffuser***




Radial diffusers are also known as layin type. Whereas slot diffusers introduce primary air from a line, radial diffusers introduce it from a point. The air flow pattern is outward in either a cross flow or circluar pattern.

## Jet Diffuser

Jet diffusers are either wall or ceiling mounted. They have multiple small outlets that introduce the primary air in fine jets in the space. These fine jets merge downstream of the diffuser into a single jet.

## Diffuser Basics

Diffuser selection is based on three main characteristics, room characteristic length, throw and separation distance. A key goal in diffuser selection is to promote entrainment. Entrainment causes mixing between primary air and space air which leads to energy absorption and balanced temperatures.

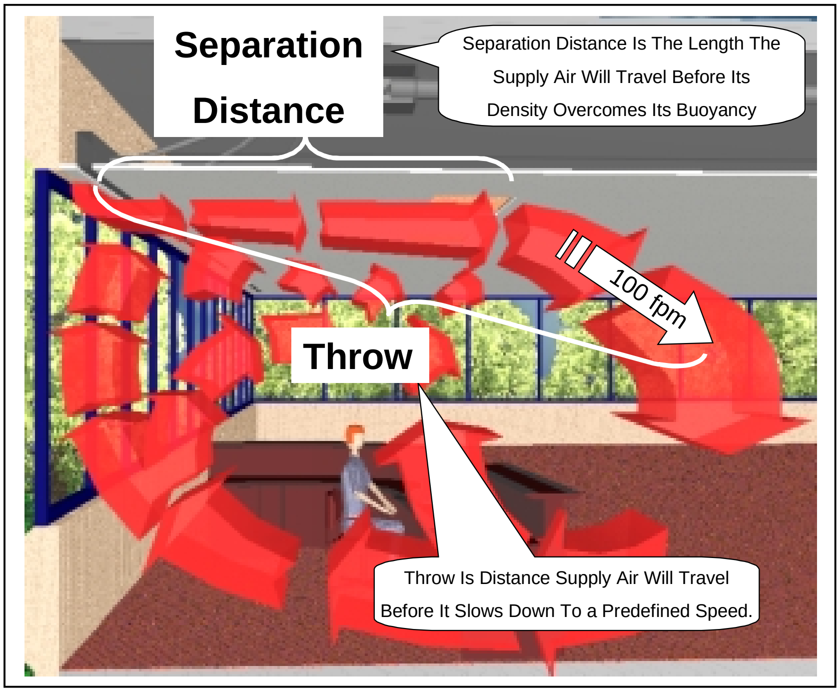
Primary air from a diffuser is introduced into the occupied space with velocity. The velocity gives the primary air inertia to keep moving and it lowers the pressure (The primary air's velocity pressure is lower than that of the space air). The higher pressure from the space air pushes the primary air up against the ceiling, allowing it to spread out across the spaces and entrain space air in a process known as Coanda effect.

For this process to occur, the diffuser must be near the ceiling surface. For example, in open plenum concepts where diffusers are installed in ducted "drops" from the main duct, there is no surface for the primary to be "pushed up against" (See Figure 11 - Separation Distance). The result is that the diffuser will not perform as catalogued. Another example is slot diffusers installed in a bulkhead at the perimeter of the building that is lower than the main ceiling height. Once the primary passes beyond the bulkhead surface, separation from the main, higher ceiling may occur. Either of these situations will cause difficulty with conventional systems and will be problematic with lower supply air temperature systems.

Manufacturers of diffusers catalog their products typically by:

- ❑ Throw (ft)
- ❑ Static Pressure Drop (in w.g.)
- ❑ NC (Noise Criterion)

***Figure 11 - Separation Distance***




The throw data is often based on three terminal velocities, 150, 100 and 50 fpm. Throw is the distance the primary air travels before it slows down to the terminal velocity.

Separation is the distance the primary air travels before it separates or "drops away" from the ceiling. The primary air will drop when its density is greater than the buoyancy provided by the reduced velocity pressure. If this happens prematurely, it is called "dumping". The cold primary air is more dense than the space air (unless the system is in heating mode). The colder the air, the sooner it will separate.

## Diffuser Selection

The following is a procedure for selecting diffusers:

**1. Determine the maximum and minimum airflow requirements.**

Maximum airflow rates are derived from cooling load calculations. Minimum flowrate cannot be lower than the minimum ventilation rate required by ASHRAE Standard 62.1. ASHRAE Standard 90.1 requires that it not be higher than 0.4 cfm/ft² to avoid unnecessary reheat (unless required by ASHRAE Standard 62.1).

**2. Select a diffuser type and location in the space.**

Diffuser selection will depend on the performance and style preferred by the engineer, architect and the owner. While many diffusers will work with lower air temperatures, slot, radial and jet diffusers are recommended.

**3. Determine the room characteristic length.**

The room characteristic length will depend on the diffuser type, diffuser location and the distance to the wall or symmetry plane.

**4. Select the throw/length (T/L) ratio.**

T/L ratios can be found in Table 4. The Air Diffusion Performance Index (ADPI) is the percentage of spaces within a room that have the desired effective draft temperature and velocity required to keep the occupants comfortable. The effective draft temperature is between –3 and +2°F and the air velocity is less than 70 fpm. The higher the APDI percentage, the more comfortable the space is.

ADPI values were originally developed for conventional air temperatures, but it has been shown that they will work for low temperature applications as well. The goal is to maximize the ADPI. Table 4 also provides the range of T/L that will maintain the APDI above 0.8. Table 4 also includes a range for T/L ratios. These can be used to test a diffuser at part air volume for dumping. For example, a slot diffuser can range from 0.5 to 3.3, which is a factor of 6. At the same time, a radial diffuser only has a range of 0.7 to 1.5 or a factor of about 2. This is the reason slot diffusers are more popular for Optimal Air and low temperature air systems than radial diffusers.

***Table 4 - Air Diffusion Performance Index***

| Diffuser Type | Terminal Velocity fpm | Room Load Btu/ft² | T/L For Max. ADPI | Maximum APDI | For APDI Greater Than | Range Of T/L |
|---|---|---|---|---|---|---|
| Slot | 50 | 40 | 1.0 | 91 | 80 | 0.5-3.3 |
| | | 20 | 1.0 | 91 | 80 | 0.5-3.3 |
| Radial | 50 | 80 | 0.8 | 76 | 70 | 0.7-1.3 |
| | | 60 | 0.8 | 83 | 80 | 0.7-1.2 |
| | | 40 | 0.8 | 88 | 80 | 0.7-1.5 |
| | | 20 | 0.8 | 93 | 90 | 0.7-1.3 |

5. **Calculate the throw distance.**

The throw distance is found by multiplying the T/L ratio from (4) with the room characteristic length from (3).

6. **Select diffuser size with correct throw and flow.**

Using the maximum flow rate and the required throw, a diffuser can be selected from the diffuser manufacturer's catalog.

7. **Calculate the separation distance at maximum and minimum flow rates.**

If the diffuser manufacturer cannot provide the separation distance based on the design conditions, it can be calculated using the following formula:

$x_s = a\ C_s\ K^{1/2}\ (\Delta T/T)^{-1/2}\ Q^{1/4}\ \Delta P^{3/8}$

Where:

$x_s$ = jet separation distance (ft)

a = Constant 11.91

$C_s$ = Separation coefficient, 1.2

K = Velocity decay coefficient

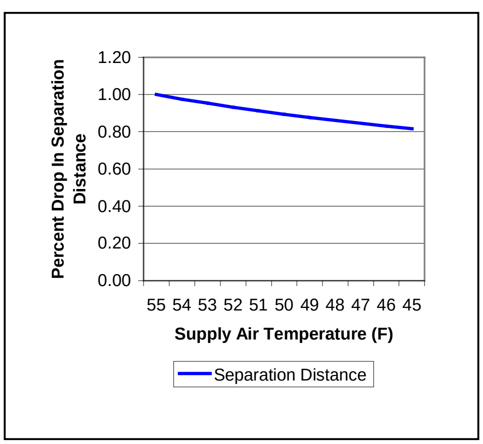
$\Delta T$ = Room- jet temperature difference (°F)

T = Average absolute room temperature (°R)

Q = Air flow rate (cfm)

$\Delta P$ = Diffuser static pressure drop (in. w.g.)

***Figure 12 - Separation Distance Vs. Supply Air Temperature***




The velocity decay coefficient should be available from the manufacturer. If not, use 1.1 for radial, 5.5 for slot and 7.0 for jet type diffusers.

Figure 12 shows the reduction in separation distance as a function of supply air temperature. Lowering the supply air temperature to 45°F reduces the separation distance by 20%. Separation distance is also linearly proportional to flowrate. That is, cutting the air volume in half will cut the separation distance in half. Special care should be taken not to oversize diffusers.

If the separation distance at minimum air volume is equal to or greater than the room characteristic length, then the diffuser selection should not dump.

8. **Check sound levels and diffuser pressure drops.**

   Optimal air and low temperature diffuser selection will have higher velocities than conventional systems so the NC levels should be checked. The diffuser pressure drop shouldn't be higher than the value used in the fan static calculations.

9. **Resize as necessary.**

   Repeat the process if a diffuser does not meet all the criteria.

### Perimeter Systems with Heating Requirements

Selection of the proper air distribution system for perimeter areas where heating is required is similar to that for conventional air temperature systems. Be sure to use air handling and air distribution equipment designed for low temperature air (including superior insulation and thermal inlet isolation) in any areas where infiltration will raise steady-state humidity levels. (See Appendix 1)

## Duct Design

Duct design for Optimal Air is similar to conventional temperature designs. Equal friction, static regain or T method optimization are acceptable. Because of the lower dew point, duct leakage is a larger concern, so more care may be required in construction.

The smaller air volumes will result in smaller ductwork. An opportunity exists to use spiral duct that will have approximately the same duct height as rectangular duct in a conventional system. Spiral duct offers lower system losses, better sound attenuation and the least surface area. It also makes static regain much easier to design.

Round spiral duct is recommended for the following reasons:

- ❑ With Optimal Air temperature air and round duct, a medium or high velocity design can be used. The result is a lower cost, more efficient air distribution system with the same headroom required as for conventional air temperatures with rectangular duct.
- ❑ The amount of surface area is reduced dramatically, 32%, with round duct vs. rectangular duct for the same velocity, and as much as 66% with round medium velocity duct handling Optimal Air temperature air. Reduced surface area means reduced heat gain as well as substantially reduced cost.
- ❑ Accurate static regain (total pressure) design with computer duct design programs is available.
- ❑ Lower fan horsepower is often attainable in computer static regain designed spiral duct systems.
- ❑ VAV terminal units (particularly those close to the fan) operate with lower pressure, meaning lower power requirements and less noise due to the inherent self-balance of a static regain spiral duct system.
- ❑ Round spiral ducts keep noise contained, substantially limiting low frequency "break out" noise often associated with rectangular duct air distribution systems. Containing the noise in the duct allows for increased attenuation from the duct liner.

### Duct Heat Gain

Even in conventional systems, heat from the plenum raises the supply air temperature between the air handling unit and the diffuser. The result is that more air is required (because the supply air temperature to space temperature delta T is reduced). Different duct runs will have different heat gains. Duct sizing software can calculate the different heat gains and adjust the required supply airflow accordingly. A common practice is to calculate the heat gain for the longest duct run and use this value for all runs (conservative), or to use 2 or 3°F.

As the design supply air temperature is lowered, it makes sense that the heat gain would increase. Opposing the increasing delta T, which is increasing the heat gain, is the reduced duct surface area from the reduced supply air volume. The two almost cancel each other out. With additional insulation often used in Optimal Air systems, the heat gain is typically reduced 40 to 80% from conventional systems. However, the reduced supply air volume means the same supply air delta T for conventional systems can be expected with Optimal Air and low temperature systems (there is less air to absorb less heat).

# Duct and Pipe Insulation

Duct and piping insulation must be carefully considered in Optimal Air systems to avoid condensation and limit duct heat gain. Table 5 shows the recommended insulation for chilled water piping as listed in ASHRAE Standard 90.1-2001.

***Table 5 - Minimum Pipe Thickness Per Std 90.1***
***Cooling Systems (Chilled Water, Brine and Refrigerant)***

| Fluid Design Operating Temp. Range (°F) | Insulation Conductivity | | Insulation Thickness for Nominal Pipe or Tube Size (in) | | | | |
|---|---|---|---|---|---|---|---|
| | Conductivity Btu•in/(h•ft2•°F) | Mean Rating Temp °F | <1 | 1 to <1-1/2 | 1-1/2 to <4 | 4<8 | ≥8 |
| 40-60 | 0.22-0.28 | 100 | 0.5 | 0.5 | 1.0 | 1.0 | 1.0 |
| <60 | 0.22-0.28 | 100 | 0.5 | 1.0 | 1.0 | 1.0 | 1.5 |

As the supply air temperature is lowered, the duct insulation will have to be increased to minimize the effects of duct heat gain. In general, if 1-inch insulation is used in a conventional system, then 1.5-inch will be required for Optimal Air systems. ASHRAE Standard 90.1 has duct insulation tables for standard conditions. These are based on the ducting location within the building, annual weather and life cycle analysis.

## Duct Sweating

Probably the greatest fear that designers have regarding low temperature air systems is condensation formation on ducts or other equipment. Research and experience indicates that with a properly designed air distribution ductwork, special condensation protected equipment, attention to possible problem buildings and areas, and with proper control, condensation will almost never be a problem.

The first thing to consider is that although the supply air is colder for Optimal Air systems, the space RH is also lower. The "relative" difference between duct temperature and space dew point is about the same for conventional and Optimal Air systems. As the supply air temperature is lowered, care should be taken during pulldown (especially if the space RH has been allowed to rise) and in areas with high infiltration.

To avoid condensation, the duct insulation should:

- Be thick enough to maintain the duct surface temperature above the space dew point temperature.
- Cover all surfaces that may be cooled below space dew point temperature.
- Have a vapor barrier to prevent condensation within the insulation.

Where ducts pass through conditioned spaces including return air plenums, the insulation values recommended above for duct heat gain are usually enough to avoid condensation. When ducts pass through unconditioned spaces or spaces where infiltration is an issue, calculations may be required.

The following equations can be used to calculate the required insulation thickness. Insulation values can be obtained from the insulation manufacturer or the values in Table 6 can be used. In most cases, the insulation will be compacted during installation. It is important to account for this in the calculations. The installed insulation values should be used.

The duct surface temperature can be calculated as follows:

$T_s = T_{sa} +[( T_a - T_{sa})* R_i /( R_i + R_s )]$

The required minimum insulation thermal resistance can be calculated as follows:

$R_i = R_s *( T_{dp} - T_{sa})/ ( T_a – T_{dp})$

The required insulation thickness can be calculated as follows:

$t_i = k_i R_i$

Where:

$T_s$ =Duct surface temperature (°F)

$T_{sa}$ =Supply air temperature (°F)

$T_a$ =Ambient drybulb temperature surrounding duct (°F)

$T_s$ =Ambient dew point temperature surrounding duct (°F)

$R_i$ =Insulation thermal resistance (°F*ft²*h/Btu)

$R_i$ =Surface air film thermal resistance, 0.62 °F*ft²*h/Btu)

$k_i$ =Insulation thermal conductivity (°F*ft²*h/Btu)

$t_i$ =Insulation thickness (in)

***Table 6 - Typical Insulation Values For Glass Fiber Insulation***

| Nominal Density Lb/ft³ | Nominal Thickness in. | Out of Package R-value °F*ft²*h/Btu | Installed Thickness iIn. | Installed R-value °F*ft²*h/Btu | Overall U-value Btu/°F*ft²*h |
|---|---|---|---|---|---|
| 0.75 | 1.5 | 5.0 | 1.125 | 4.2 | 0.2075 |
| | 2 | 6.7 | 1.5 | 5.6 | 0.1608 |
| | 3 | 10.0 | 2.25 | 8.3 | 0.1121 |
| 1.0 | 1 | 3.7 | 0.75 | 3.0 | 0.2762 |
| | 1.5 | 5.6 | 1.125 | 4.5 | 0.1953 |
| | 2 | 7.4 | 1.5 | 6.0 | 0.1511 |
| 1.5 | 1.5 | 6.0 | 1.125 | 4.8 | 0.1845 |
| | 2 | 8.0 | 1.5 | 6.4 | 0.1425 |

## Duct Coverage

It is very important that all cold surfaces be insulated. Condensation problems are usually linked to ducts or fittings that are not insulated versus not enough insulation. The following are some recommendations:

- ❑ Don't let the insulation get damaged during construction.
- ❑ Cover weld pins with vapor retarder.
- ❑ Seal penetrations in external insulation such as hangers, sensors etc.
- ❑ Place rigid insulation between unistrut and ducting.
- ❑ Insulate VAV box inlet collars.
- ❑ Leak test ductwork before insulation is added.

## Vapor Barrier

The insulation will be degraded if condensation is allowed to occur in it. Microbal growth is also possible and can lead to possible IAQ issues. To avoid these problems a vapor barrier is required.

For internally insulated duct, the duct walls and seal are the vapor barrier. Seal all joints and seams to minimize moisture movement.

For externally insulated ducts, use two coats of a vapor retardent coating at least 1/16 in. thick with a layer of glass cloth in between.

## Duct Leakage

Duct leakage degrades any HVAC system. With Optimal Air systems, there is the added concern of condensation. ASHRAE Standard 90.1-2001 requires duct leakage testing for static pressures exceeding 3 inches w. c. The standard requires no less than 25% of the ducting be tested. The maximum permitted leakage is:

$L_{max} = C_L * P^{0.65}$

Where:

$L_{max}$ =Maximum permitted leakage in cfm/ft² duct surface area.

$C_L$ =Duct leakage class, cfm/ft² at 1 inch w.c.

6 for rectangular sheetmetal, rectangular fibrous ducts, and round flexible ducts.

3 for round/flat oval sheetmetal or fibrous glass ducts.

P =Test pressure, which shall be equal to the design duct pressure class rating in inches w. c.

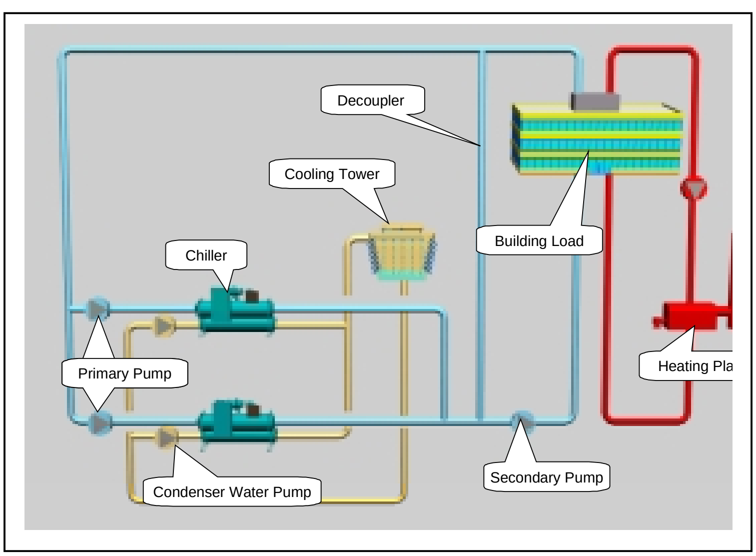
Where condensation from leakage is a concern, requiring testing for static pressures less than 3 inches w. c. may be prudent.

# Primary System Selection

As the supply air temperature is lowered, the air conditioning or refrigeration system is negatively impacted (See Refrigeration). Lower supply air temperatures put additional demands on the refrigeration systems, so special care in selection is required.

## Chiller Plants

***Figure 13 - Typical Primary-Secondary Chiller Plant Design***




Chiller plants will not require major changes for Optimal Air applications. The chillers must be selected with the operating conditions in mind. If retrofitting an existing building with an Optimal Air system, the chiller selections should be checked by the equipment manufacturer to confirm proper operation at the new conditions.

All chiller types and plant configurations may be used.

System and chiller selection should strongly consider the impact of 42 to 40°F supply water temperatures. Chilled water delta T can be increased from the typical 10°F since the air delta T is being increased. Larger chilled water delta Ts offer pump and pipe capital savings and pump work operational savings. Larger chilled water delta Ts will also assist the chiller performance but hinder coil performance. The balance will have to be found using software tools such as McQuay Energy Analyzer™. Series chiller plant design and either dual compressor or VFD chillers may make sense. More information on chiller plant design can be found in McQuay Centrifugal Chiller Fundamentals and Chiller Plant Design Application Guides.

## Applied Rooftop Units

***Figure 14 – McQuay RPS Applied Rooftop Unit***

Optimal air applications present unique challenges for rooftop systems. The lower supply air temperatures require flexibility in the DX coil and other refrigeration circuit components. In addition, VAV requires the product work well at part cooling load and part airflow.

Conventional packaged rooftop units were never intended for Optimal Air applications. Such products do not have the necessary flexibility in refrigeration components (coils, compressors, etc.) to meet the required design conditions. Further, the unit construction was not intended to handle the reduced temperature air.

Applied products (as shown in Figure 14) have the flexibility and construction to be used in Optimal Air applications. Mixing box arrangements are also important to avoid heightened stratification issues with ventilation air. Applied rooftop units can be equipped with specially designed mixing boxes and air blenders to mix the high percentage of ventilation air properly.

## Vertical Self-Contained Units

***Figure 15 -– McQuay SWP Vertical Self-Contained Unit.***

Vertical self-contained units such as that shown in Figure 15 are a good match for Optimal Air systems. Like rooftop units, the product must have enough selection flexibility to match the design conditions required for Optimal Air while still staying within good design practice. Unit construction must also meet the additional demands of Optimal Air.

Vertical Self-contained units are often applied floor-by-floor with waterside economizers. Units such as McQuay’s SWP product line, offer simultaneous waterside economizer “freecooling" with supplemental mechanical cooling. This capability extends the freecooling season for maximum energy savings. This is very important when considering Optimal Air or conventional systems.

Each self-contained unit can be selected to handle its own ventilation air, or a dedicated ventilation unit can be applied. A dedicated ventilation unit creates the opportunity for energy recovery, such as the enthalpy wheels available in both McQuay rooftop and Vision AHU products. The use of energy recovery for ventilation air can further improve system performance, while at the same time allowing for smaller units on each floor.

# ASHRAE Standard 90.1 Compliance

The 1999 and 2001 versions of ASHRAE Standard 90.1, Energy Standard for Buildings Except Low Rise Residential Buildings, has mandatory requirements for refrigeration equipment and prescriptive requirements for fan work.

Tables 6.2.1A through G list mandatory performance for various types of air conditioning equipment. In most cases the equipment is rated at conventional operating conditions. Equipment used for Optimal Air systems would have to meet the minimum requirements listed in the Standard if it were operated at conventional conditions (See section 6.2.1). At the actual Optimal Air conditions, the performance may drop below the minimum stated but that would still be acceptable.

An exception to the above paragraph is for centrifugal chillers. Tables 6.2.1H through J list minimum acceptable performance for centrifugal chillers operating at non-standard conditions. If centrifugal chillers are used for Optimal Air designs, then these tables must be used.

***Table 7 - ASHRAE Std 90.1 Fan Power Limitations (6.3.3.1)***

| Supply Air Volume | Allowable Nameplate Motor Power | |
|---|---|---|
| | Constant Volume | Variable Volume |
| <20,000 cfm | 1.2 hp/1000 cfm | 1.7 hp/1000 cfm |
| ≥20,000 cfm | 1.1 hp/1000 cfm | 1.5 hp/1000 cfm |

Section 6.3.3 of the Standard describes the prescriptive method for fan power limitations. Table 7 lists the conventional fan power limitations. Section 6.3.3.1b allows the limitations to be relaxed if additional filtration, energy recovery devices etc are included. Section 6.3.3.1c allows the limitations to be relaxed if the supply air delta T is larger than 20°F (for example Optimal Air applications). The correction method is;

Allowable Fan System Power = [Table 6.3.3.1]*(Temperature Ratio) + Pressure Credit + Relief Fan Credit

Where:

Temperature Ratio = $(T_{t\text{-}stat} - T_{space})/20$

Pressure Credit (hp) = Sum of $[CFM_n * (SP_n - 1.0)/3718]$ + Sum of $[CFM_{HR} * SP_{HR} /3718]$

Relief Fan Credit HP = $F_R$ HP * $[1 - (CFM_{RF} / CFM_n)]$

$CFM_n$ = Supply air volume of the unit with the filtering system (cfm)

$CFM_{HR}$ = Supply air volume of heat recovery coils or direct evaporative humidified/cooler (cfm)

$CFM_{RF}$ = Relief fan air volume at normal cooling design operation

$SP_n$ = Air pressure drop of the filtering system when filters are clean (in. w.g.)

$SP_{HR}$ = Air pressure drop of heat recovery coils or direct evaporative humidifier/cooler (in. w.g.)

$T_{t\text{-}stat}$ = Room thermostat setpoint

$T_{space}$ = Design supply air temperature for the zone in which the thermostat is located

$F_R$ = name plate rating of the relief fan in HP

For Optimal Air applications, it is important to consider the credit for lower supply air temperature and non-standard equipment conditions. Refer to the example in the sidebar.

> ***Example;***
>
> *Consider a simple Optimal Air system that is 20,000 cfm and 92 tons. It is VAV with chillers operating at 40°F supply water temperature. The Optimal Air balance point is 49°F, the room setpoint is 76°F. Find the fan power limitation and minimum chiller efficiency as per Standard 90.1.*
>
> ***Solution:***
>
> *Since the supply air delta T is greater than the standard 20°F, a correction is required.*
>
> *Temperature Ratio = (76°F - 49°F)/20 = 1.35*
>
> *Allowable Fan System Power = 1.5 hp/1000 cfm * 1.35 = 2.025 hp/1000 cfm*
>
> *Standard ARI 550/590 chiller ratings are based on 44°F supply water temperature. Since this is a centrifugal chiller, Table 6.2.1H must be used. Assuming the condenser flow remains at the normal 3 gpm/ton and temperature at 85°F, the revised minimum COP is 4.58. To convert to kW/ton divide 3.516 by the COP = 0.767 kW/ton.*

## Indoor air Quality

The same methods used to provide indoor air quality for conventional systems apply to Optimal Air systems. In most cases, ventilation air requirements are based on ASHRAE Standard 62.1.

Using 15 cfm per person and 150 ft² person for a typical office building yields 0.1 cfm per ft² ventilation air. This would be true for conventional systems as well as Optimal Air systems. This ventilation rate is required whenever the building is occupied.

For the Chicago building example at 55°F, the supply air rate is 0.76 cfm/ft². The ventilation air makes up 13% of the total supply air. For Optimal Air (49 °F), the supply air rate is 0.59 cfm/ft². Now the ventilation air makes up 17% of the total supply air. The higher percentage of ventilation air can raise the minimum cfm where reheat is required. (This is not likely an issue with office buildings but care should be taken with schools). It also requires careful mixing in the mixing box of the air handling unit to avoid stratification and potential coil freezing. These issues exist with conventional systems but are magnified by reduced supply air volumes and increased ventilation ratios.

## Building Automation

Optimal Air systems do not require any major changes versus conventional building control systems. The following is a quick list of some minor changes.

- ❑ Discharge air control for VAV air handling units set at lower (Optimal Air) temperature.
- ❑ Zone temperature control with remote reset to account for RH suppression. Space setpoint should set for the true design condition – not just "75°F".
- ❑ Reset of supply air temperature downward from 55°F to Optimal Air temperature during pulldown to avoid duct "sweating."
- ❑ Enthalpy economizer control, which can allow "free" cooling at higher than the discharge air temperature set point. ASHRAE Standard 90.1 has minimum economizer requirements including controls logic (See section 6.3.1).
- ❑ Supply Air Temperature reset. The principle is to raise the supply air temperature when the load profile allows. Two things will occur. First, the fan work will increase since more supply air will be required. Second, the cooling load will decrease since the supply air will be warmer. Whether this will actually save energy is not clear. In locations where the weather profile will allow significant economizer savings by raising the supply air temperature, reset may be a good idea.

# Life-Cycle Analysis

Evaluating different engineering solutions is always part of a good proposal. Optimal Air systems are no different. In the case of Optimal Air, there may be no need to do any calculations because Optimal Air systems cost less to build (lower capital cost) and have the same operating cost as conventional systems (assuming the balance point was used for the design). Here is a list of Capital Items to consider:

- Duct sizing will decrease almost linearly with reduction in air volume. The installed cost will not change linearly because of the labor portion. A 20% reduction in air volume can result in 80% savings of the 20% reduction or 16% overall savings in sheetmetal cost. On the plus side, there are less pounds of steel and fewer man-hours to install it. On the minus side there is more insulation. Terminal boxes and diffusers will be a wash since there are fewer of them but the equipment cost will be higher than conventional equipment.
- HVAC equipment will cost about the same. This is conservative because the air handling equipment will cost less and refrigeration equipment will be slightly more. There is typically more capital invested in air handling than refrigeration.
- Building envelope should be the same for new construction. In the case of retrofit applications, it will depend on the quality of the existing building.
- The cost of space may also need to be evaluated. Not accounting for space savings is conservative. There will be space savings but they may be difficult to realize. If enough plenum height savings can be realized to add another floor within the same building envelope, then that rentable space should be accounted for.

Simple payback calculations do not take into account the cost of money, taxes and depreciation, inflation, maintenance or increases in the cost of energy. A more complete analysis should include Internal Rate of Return (IRR) and net present value (NPV). In the HVAC industry, many projects fail simple back (they are in the 5-year range) while passing IRR (they offer a 25% rate of return). Software analysis tools such as McQuay's Energy Analyzer™ can be used to perform both energy and life-cycle analysis that include simple payback, IRR and NPV.

# Summary and Conclusions

In many applications, particularly buildings with high sensible heat ratios, lowering the supply air temperature to Optimal Air temperatures can significantly lower the capital cost of the project and improve system sound performance without increasing the operating cost (In fact, it may reduce operating costs.). The design does not significantly affect the conventional VAV design. Optimal air systems can be applied using applied rooftop, vertical self-contained or chiller/AHU systems.

Proper annual energy and life cycle analyses are required to assess the advantages of Optimal Air.

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# Appendix 1: Perimeter and Skin Heating Systems

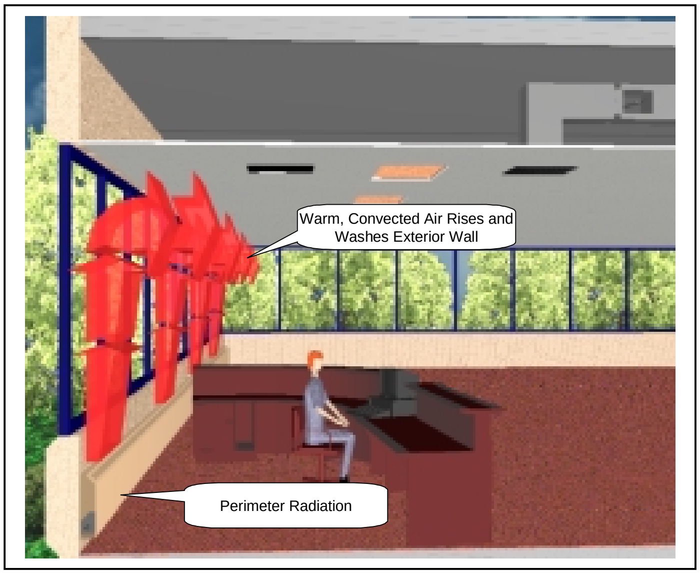
Design of perimeter and skin heating systems is the same for low temperature air systems as conventional systems. Attention must be paid to skin heating loss and the “cold wall” effect to assure Winter comfort.

## Perimeter Zones with High Skin Heating Loss

Whenever the heating requirements at design are 400 BTUH per linear or higher, a severe down draft condition can exist. To offset the down draft, one of three system types is generally used:

### Radiation

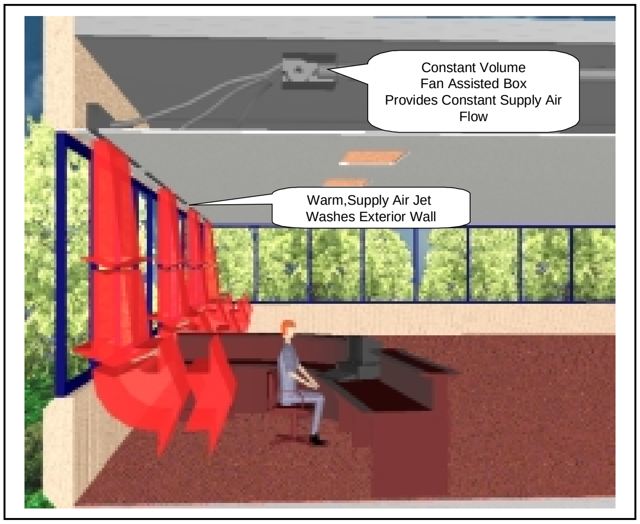
*Figure 16 - Perimeter Radiation*




Radiation along perimeter walls and under windows is often used, particularly in cold climates. Control is normally based on reset of supply water (boiler discharge) temperature upward as outside air temperature goes down. While this control method will provide adequate heating energy to offset downdrafts, energy is wasted in spaces with solar load as additional cooling air must be introduced to offset overheating from both the radiation and the sun. ASHRAE Standard 90.1 has specific requirements for zoning of perimeter heating to minimize unnecessary reheat. Better energy efficiency can be obtained from solar compensated radiation temperature reset or from reset by individual zone thermostats, although some downdraft control may be lost. (See Figure 16)

### Overhead Distribution – Series Fan Terminal – All Air

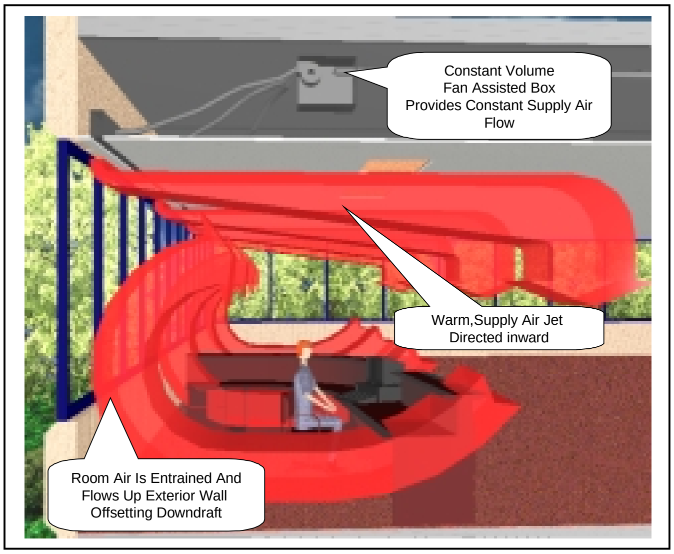
*Figure 17 – Series Fan Terminal Constant Volume Heating Flow – High Loss Wall*




The use of series fan terminals provides overhead air distribution for winter downdraft control but also provides some of the zone cooling during warm weather operation. Normally sized for the worst case skin (heating or cooling) loads, series fan terminals, when used on Optimal Air systems, mix cooled air and ceiling plenum air in varying amounts when “cooling” is required. They circulate plenum air during “no load” conditions and provide “heated” plenum air (without reheat) when heat is required. Control is normally by building face with solar compensated outdoor thermostat or by “worst case” perimeter zone.

## Perimeter Zones with Moderate Skin Heating Loss

***Figure 18 - Series Fan Terminal Constant Volume Heating Flow – Medium Loss Wall***




When heating requirements at design are between 250–400 BTUH per linear foot, moderate downdraft conditions can exist. Radiation or hot air systems may be used as above. However, slot diffusers blowing "in" toward the room will provide room air motion, which will offset downdraft. In this case, constant volume is again used to provide constant "energy of air motion" to provide downdraft control regardless of air supply temperature.

## Low Skin Heating Loss Systems

When heating requirements at design are below 250 BTUH per linear foot, downdraft is generally not a problem. In this case, diffuser selection is based on cooling with some means to introduce heat as required. Constant air volume is not required.

Post Office Box 2510, Staunton, Virginia USA • (540) 248-0711 • www.mcquay.com