
# HVAC Optimization with Cold Air Distribution

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# HVAC Optimization with Cold Air Distribution

Most conventional air conditioning designs are based on supplying 55°F air to the space. This temperature generally provides the required humidity ratio to maintain space conditions at 75°F within reasonable humidity control range of 55 ± 5%. Over many years, it has become a standard design parameter upon which equipment selection specifications and rules-of-thumb are based. Designers repeatedly use this norm simply because they know it works; even though it may not necessarily offer the best economics in terms of energy performance, cost or air quality.

What if we design a HVAC system to supply air below 55°F! Several questions will come to mind.

1. Is this been used elsewhere?
2. What are the benefits?
3. What are the drawbacks?

The answer to the first question is yes. The cold air design has been in use in many applications including industrial, manufacturing, control rooms, cold rooms, pharmaceutical and medical facilities and even in the commercial buildings. We will find the answers to the remaining questions in the sections below.

## What is cold air distribution?

The "conventional approach" to the design of the HVAC system is based on estimating the heat load and computing the air volume for delta temperature ($\Delta T$) between the room setpoint minus the supply air temperature. For comfort applications, the room setpoint temperature is 75°F and the supply air temperature is designed for 55°F; therefore the $\Delta T$ is taken as 20°F.

The "cold air systems" distribute air at a temperature much lower than 55°F. The coldest practical air temperature is about 38°F, with most cold air designs using 42-48°F.

## Why use cold air distribution?

For a given air-conditioning load, as the supply air temperature is reduced, the supply air volume is reduced proportionally. Let's check this for 1 ton of air-conditioning load.

The sensible heat gain equation is $Q = 1.08 \times CFM \times \Delta T$

Where:

Q is sensible heat in Btu/hr. *(Note that for 1 ton of refrigeration, Q is equivalent to the heat extraction rate of 12,000 Btus per hour)*

CFM is the air volume required

$\Delta T$ is the temperature differential of the space setpoint minus the supply air temperature

*Consider two cases:*

**Case # 1:** The room setpoint temperature is 75°F and the supply air temperature is 55°F

**Case # 2:** The room setpoint temperature is 75°F and the supply air temperature is 45°F

In Case # 1, the $\Delta T$ is 20°F and therefore the air volume per ton of air-conditioning load will be:

CFM = 12000 / (1.08 x 20) or = 555/ton

In Case # 2, the $\Delta T$ is 30°F and therefore the air volume per ton of air-conditioning load will be:

CFM = 12000 / (1.085 x 30) or = 370/ton

This shows that by simply lowering the supply-air temperature from the 55°F to 45°F, it reduces the supply-air volume by 33%.

## What are the Benefits of cold air distribution?

The primary advantage of cold air distribution lies in the dramatic reductions in the supply air volume. What this means is that the HVAC equipment will use smaller air handling units (AHU's), air ducts, terminal devices, insulation and fittings.

Let's take a closer look at the opportunities extended by considering an example of an office complex requiring 100TR of air-conditioning. On a conventional system design operating at 55°F, this facility will require 55,500 CFM of air, while following Case # 2, where cold air distribution at 45°F, the supply air volume will be 37000 CFM.

### 1) Smaller Air Ducts

| Sno. | Parameters | Case # 1 | Case # 2 |
|---|---|---|---|
| **A** | Air volume | 55,500 CFM | 37,500 CFM |
| **B** | Duct air velocity | 1,500 FPM | 1,500 FPM |

| Sno. | Parameters | Case # 1 | Case # 2 |
|---|---|---|---|
| C | Duct cross sectional area (A/B) | 37 sq-ft | 24.6 sq-ft |
| D | Assume round duct shape, diameter of duct (d) = $(C * 4 / 3.14)^{1/2}$ | 6.86 ft | 5.6 ft |
| E | Assuming 100 feet duct length, surface area of duct = (3.14 * D * 100) | 2,154 sq-ft | 1,758 sq-ft |

***Benefits***

1. Case # 1 will use 2154 sq-ft of duct work vs. 1758 sq-ft for Case #2. This is 18% reduction in the ductwork. What this means is:
   - Lower sheet metal
   - Lower insulation
   - Lower fittings such as volume control dampers, terminal devices, grilles, registers etc.
   - Smaller variable air volume (VAV) boxes

   All this represents a very large capital savings.
2. The duct diameter in Case # 1 is 6.86 ft vs. 5.6 ft in Case # 2. This saves 1.26 ft of plenum space, thereby reducing the total height requirements of the building. More space above the ceiling allows more space for cable trays, control cabling, and fire protection sprinklers.
3. The smaller ducts mean smaller core areas or vertical air shafts providing additional floor space.
4. Shorter floor-to-floor height, attributable to smaller ductwork, may significantly reduce the cost of glass and steel in a multistory building; perhaps even add a floor of rentable space.
5. Smaller ductwork means smaller penetrations on the structural elements, easy installation, transportation and labor handling.

### 2) Smaller Air-Handling Units (AHUs)

| Sno. | Parameters | Case # 1 | Case # 2 |
|---|---|---|---|
| A | Air volume | 55,500 CFM | 37,500 CFM |
| B | Face velocity across coil | 500 FPM | 500 FPM |
| C | Coil Face Area (A/B) or<br>[Air volume (CFM) ÷ 500 (FPM)] | 111 sq-ft | 74 sq-ft |

***Benefits***

1. Case # 1 will use 111 sq-ft of cooling coil vs. 74 sq-ft for the Case # 2 at 500 feet per minute face velocity. This is 33% reduction in the size of AHU. What this means is:
   - Less sheet metal for the AHU
   - Less insulation and painting requirements
   - Smaller filters and dampers
   - Smaller Fan/s and motor/s
2. Note that the cooling coil has to be designed for 100 TR (1.2 MBH) heat extraction capacity. Since the face area of cooling coil is reduced, the increased heat transfer area will be compensated by increasing the depth of the coil, i.e. by adding rows to the cooling coil. This will have negligible impact on the size of the AHU length, though it will affect the air pressure drop across the coil. The fan power requirement will, however, reduce as the supply air volume reduction will far offset the marginal increase in fan static.
3. Smaller AHU means lesser foot print area in the mechanical room. The savings on the mechanical spaces could create significant extra useable/rentable floor space. The cost benefits shall be tremendous particularly in the premium real estate buildings.
4. Alternatively, the extra space made available by the smaller air-handler footprint can be used for attenuation in sound-sensitive applications.
5. The noise, vibration and the structural loading on the account of smaller AHU will be considerable.

**6.** The double panel insulation on the smaller AHU will be significantly lower. Epoxy painting requirements will also be lower.

**7.** Smaller air-handling equipment lessens capital expense. If architectural space can be reduced due to the smaller system components, additional construction cost savings can be realized.

### 3) Reduced Fan Energy Consumption

| Sno. | Parameters | Case # 1 | Case # 2 |
|---|---|---|---|
| **A** | Air volume | 55,500 CFM | 37,500 CFM |
| **B** | Assume fan static pressure (SP) | 2 in-wg | 2 in-wg* |
| **C** | Assume fan efficiency | 70% | 70% |
| **D** | Fan Brake Horse Power = A * B / (6,356 * C)<br>[Air volume (CFM) * SP (in-wg) ÷ (6,356 * Fan efficiency)] | 25 BHP | 16.6 BHP |
| **E** | Energy Consumption per annum assuming 7,200 hours operation = 7,200 * D* 0.746<br>[Number of hours * BHP * 0.746] | 134,280 kWh | 89,161 kWh |
| **F** | Energy Savings = 134,280 – 89,161 = ~45,000 | ~ 45,000 kWh or $ 3,600 @ 8c per kWh | |

*Note that the cold air distribution shall require deeper coil with higher pressure drop as compared to the conventional system. This however shall be largely offset by the lower pressure drop downstream from the fan in the ductwork due to reduced air volume.

***Benefits***

1. Case # 1 will use 25 BHP vs. 16.6 BHP for the Case # 2. This represents 33% reduction in the fan power savings.
2. Case # 2 represents a saving of approximately 45,000 kWh of electrical energy or $3600 per annum.

3. Less fan horsepower reduces the cost of the electrical installation. For reduced fan capacity, the electrical switchgear and cabling costs will be lower.
4. The peak demand charges will also be low due to lower installed capacity.

### 4) Pump Energy Savings

Long standing conventional practice of designing the chiller water systems is based on 'ARI' conditions of 44/54°F chilled supply/return temperatures and 85/95°F condenser water supply/leaving temperatures. For cold air applications, the chilled water supply air temperatures are lowered or ΔT between the chilled water supply and return temperature is increased to 16 or 20°F from the customary 10°F. The higher ΔT between the chilled water supply and return, temperature demands much less chilled water flow rate (just as in the case of cold air supply) which means the pump and motor will be smaller and the operating expenses will be low. To evaluate this, let's look at the following table:

| Sno. | Parameters | Case # 1 | Case # 2 |
|---|---|---|---|
| A | Heat Load | 100 TR | 100 TR |
| B | ΔT between supply & return chilled water | 10°F | 16°F |
| C | Water Flow (A * 24/ B)<br>[Heat load (TR) * 24/ ΔT (°F)] | 240 GPM | 150 GPM |
| D | Assume pump head | 100 ft | 100 ft |
| E | Assume pump efficiency | 80% | 80% |
| F | Pump brake horsepower (C * D / (3,960 * E)<br>[Flow (GPM) x Head (ft)] / [3,960 * Pump efficiency] | 7.5 BHP | 4.7 BHP |
| G | Energy Consumption per annum assuming 7,200 hours operation (7,200 * F* 0.746) | 40,284 kWh | 25,244 kWh |

| Sno. | Parameters | Case # 1 | Case # 2 |
|---|---|---|---|
| | [Number of hours * BHP * 0.746] | | |
| H | Energy Savings = 134,280 – 89,161 = ~45000 | ~ 15,000 kWh or $ 1,200 @ 8c per kWh | |

***Benefits***

1. It's apparent that a higher temperature differential of 16°F results in 150 GPM chilled water flow while a 10°F results in 240 GPM pump capacity. This marks 60% reduction in flow capacity. Not only does this reduce the capital cost of chilled water distribution network because of reduced pipe sizes, fittings and insulation, it also reduces size of the circulator.
2. The smaller circulator will likely operate on lower wattage by ~2 kWh and will result in a saving of ~15,000 kWh or $1, 200 @ 8cents per kWh.

Note that the conventional 10°F rise for chilled water (1 GPM per 5,000 Btu/hr) has been a standard norm over the years and was primarily set to establish a high order safety factor against flow balance problems; however, it is wasteful because the system is generally over pumped when compared with other design of higher $\Delta T$ possibilities. This chilled water higher $\Delta T$ can be considered even with the conventional system with or without the cold air distribution system.

### 5) Indoor Air Quality (IAQ)

The cooling of air below its dewpoint results in dehumidification. Cold air systems employ cooling coils that are lower in face area but are larger rows deep. This type of construction is perfect for high latent loads or moisture removal. Condensation on cooling-coil fins is up to three times higher than conventional systems, which virtually turns the cooling-coil into an "air washer". The benefits are:

1. The cold air distribution provides improved humidity control, thereby enhancing the indoor air quality and the comfort that occupants perceive.
2. In the conventional system, the humidity control takes precedence over temperature control and therefore requires reheat. Reheat is a very inefficient way of operation where the air is first sub-cooled for dehumidification and than reheated to maintain the temperature. The humidity control takes precedence over the temperature control. The energy consumption is twice that of normal in cooling and heating.

3. In the cold air systems, the temperature and the humidity control is almost simultaneous therefore don't rely on the costly reheat.
4. Cold air systems offer greater reliability and efficiency for applications where the humidity control is critical (much below 50%) such as control rooms, data processing centers, medical facilities and industrial applications.
5. Under part load conditions, the cold air systems offer better control over temperature and humidity. The HVAC applications rarely run at peak load and run at part load majority of the time.
6. The large area applications such as shopping malls, theaters, or public spaces requiring high outside air for ventilation or for building pressurization needs shall be benefited with the cold air systems. The space relative humidity increases as the fraction of outside air increases. The cold air systems are better able to control relative humidity within comfort ranges.
7. The effective dehumidification and moisture condensate carry away undesirable impurities, particles, and toxins down to the drain. Lower space relative humidity deters the growth of mold and mildew. Carpets, furniture, and other building materials last longer and are less likely to develop moisture-related odors.
8. Overall air quality is perceived to be superior. Particularly in humid climates, cold air distribution systems can be designed to control humidity without expense of special equipment or the additional energy use of active dehumidification.

### 6) Noise Levels

The air handling units, fan terminal units, etc., are reduced to approximately one half the size compared to the conventional system. Therefore, the fans and motors will be smaller, which lowers the mechanical sound power levels of the air handling units.

Since the cold air systems deliver much less air volume, these are substantially less noisy in terms of air sound, often minimizing or entirely eliminating sound attenuating requirements.

### 7) Potential for Thermal Storage

An added advantage of cold air distribution systems is their suitability for applications in thermal storage. By distributing lower temperature air at 40°F to 50°F throughout the building, a cold air distribution system can take great advantage of the chilled water at ~33 to 35°F produced by the ice storage system.

The thermal storage systems use the principles of demand side management (DSM). Normally the electrical usage during off-peak (night time) is low; and therefore, to encourage the use of electricity during such periods, the thermal energy storage systems produce and store chilled water or ice during night. The stored thermal energy is then used during day time, when the air-conditioning is the highest. What this means is that less refrigeration will work during day time and the facility's electrical demand won't exceed the permissible limits. Many utility companies provide rebate on this system and some offer discounted electrical tariff during night. Rebates offered for thermal-storage systems often apply to cold-air distribution, too.

### What are the potential problems associated with the Cold Air Distribution Systems?

The low supply air temperature results in reduction of supply air volume by 40 to 50%. Reduced air volumes may cause low overall air motion, flow balancing problems and/or stagnant areas not consistent with comfort and high indoor air quality. This requires special attention to air distribution, and sound engineering practices is critical ingredient in any successful cold-air comfort system.

Designers unfamiliar with these aspects typically cite three concerns:

1. Condensation
2. Delivering cold air to comfort zones
3. System energy consumption

#### 1) Condensation

Condensation on the duct surfaces is a perceived concern.

A successful cold-air distribution system must prevent condensation from forming on the walls, plenums, diffusers or other areas of the building under all possible operating conditions. Unwanted condensation besides spoiling the false ceiling, carpets and other materials may also lead to mold growth and other allergic toxins.

We have discussed earlier in this course that the cold air system is an excellent way to help solve unwanted moisture problems.  One of biggest benefits of cold air systems is the lower relative humidity, and for condensation to happen there should be moisture in the space. So why should we be concerned about condensation?

The concern is the 'Infiltration'. Moist and warm air leaking into the building is more likely to condense on cold surfaces. To overcome this problem the building must be adequately pressurized to prevent any uncontrolled infiltration. Maintaining a slightly positive pressure (0.3 to 0.5 in wg) relative to ambient is a key design feature in the air-conditioning of cold air.

In addition, the following design features must be addressed:

1. The cold surfaces must lie inside the humidity-controlled envelope, and any cold surface located outside this envelope must be completely insulated.
2. Both the supply and return ducts must be perfectly insulated with optimum thickness, sometimes higher than required for conventional designs. However, due to the reduced humidity maintained in the space, well sealed, standard 2-inch fiberglass insulation is more than adequate.
3. Select diffusers that are constructed of a self insulating material. Specify insulated duct hangers.

### 2) Dumping cold air

Dumping cold air has two associated problems: First, the dumping of cold air itself which will create cold spots, and second, since the supply air volumes are low, the balancing and throw of air to all the corners become a challenge. Designers typically use either the custom designed air terminal diffusers or fan-powered variable air volume units to mitigate these concerns.

#### Option #1 - High-aspiration diffusers

With cold air distribution systems, the choice of diffuser is critical due to potential concerns of condensation, ventilation and cold spots. Conventional diffusers are not capable of overcoming these problems.

Diffusers designed based on induction principle provide answer to successful cold air distribution. These diffusers typically have high-aspiration ratio which induces room air towards the supply air diffuser. This allows direct mixing of supply and room air, thereby increasing the mass flow rate, which will in turn improve both the room-air circulation and diffuser throw. The diffuser face remains 12 to 15°F higher than the supply air temperature, thus maintaining the diffuser surface above room dewpoint. The self-insulating construction material plus the room air impinging on the diffuser surface (due to the high induction rate) evades condensation.

Linear diffusers tend to offer the best performance since they have higher supply air velocities than lay-in type diffusers. For example, a linear slot diffuser recirculates 1 CFM of room air for each 1 CFM of supply air that it delivers.

Swirl diffusers guarantee the minimum flow rates even at reduced supply air rates. The design of these swirl diffusers guarantees a swift reduction in temperature and flow velocity by means of swirling discharge and the addition of induction air.

**Diffuser Location:**

The following installation guidelines must be followed:

1. Air stream from opposing diffusers should not collide at greater than 150 FPM air velocity.
2. Air stream from diffusers should meet adjacent walls between 50 to 1,000 FPM air velocities.
3. Diffusers should be placed so that the lateral distance between a diffuser and the wall is the diffuser's throw (in feet) at 50 FPM terminal velocity times 0.404, or less. The lateral distance between adjacent diffusers is the diffuser's throw at 50 FPM terminal velocity times 0.808, or less.

**Option #2 – Fan-Powered Variable Air Volume (VAV) Terminal Units**

Often the choice of diffusers is driven by architectural concerns. Non-aspirating diffusers such as perforated plates or concentric grilles may not perform well in cold-air applications, but the architect may require it for interior aesthetics. If this type of device is used, couple it with fan-powered VAV terminals. The fan-powered VAV terminals avoid dumping cold air into the space by mixing 40°F primary supply air with return air and delivering 55°F air to the diffuser. This approach ensures high air recirculation rates in the space and requires the use of conventional air duct sizing downstream of the mixing box.

Series fan-powered VAV terminals may be preferred for large conference rooms or other applications where constant airflow is desirable. Parallel fan-powered VAV terminals on the other hand, are well suited for comfort zones where less air motion during off-peak conditions is preferred.

With either terminal configuration, the air-blending fans run continuously during occupied hours. This addition of forced power VAV does increase the number of electrical drops in the building, but still the overall installed capacity is lower due to reductions in fan horsepower.

### 3) System energy consumption

The cold air designs require the refrigeration system to work harder. The chiller energy consumption will increase by almost 7.5% if the chilled water is produced at 38°F instead of 42°F required in conventional designs. The refrigeration work increased as expected due to the lowered suction pressure required by the chillers; the increased load from the ventilation air (which must be cooled to a lower enthalpy than 55°F supply air systems); and the increased operating hours to offset the reduced economizer operation.

The fan energy savings operating on lower volumes will offset the large amount. Cold-air systems require much less energy to pump water and move air than conventional designs, and do not need reheat coils for humidity control. The additional costs of the refrigeration system or the high energy consumption for the refrigeration system are more than offset by smaller air handling units, air ducts, terminal devices and reduced electrical requirements. If architectural space can be reduced due to the smaller system components, additional construction cost savings can be realized.

*It is strongly recommended to carry out life cycle analysis of using cold air distribution systems v/s conventional systems. While cold air systems will save on the first cost, it is equally important to strike a balance between the first cost and recurring operational costs.* An analysis of a building by a supplier indicates the following typical results:

1. Annual savings in pump energy resulting from the use of colder chilled water typically range from 1¢ to 3¢ per square foot.
2. Moving colder air usually cuts 6¢ to 8¢ per square foot from supply-fan energy costs, but savings in excess of 10¢ per square foot are not uncommon.
3. Not every component of a cold-air system contributes energy savings, however. Despite small individual wattages, the continuous operation of fan-powered VAV terminals, as air blenders, during occupied hours adds up. Energy costs for parallel terminals increase by as much as 1¢ to 1.5¢ per square foot, and series terminals add more.
4. The costs of producing colder water and colder air require attention, too. Lowering the leaving chilled water temperature from 42°F to 38°F can increase chiller horsepower-per-ton by 6 to 10 percent, depending on compressor type. Maintaining a lower relative humidity in the building increases the amount of cooling required.

5. Offsetting these increases in chiller energy is the substantial and continuous reduction of heat generated by supply and return fans. (Reducing system airflow may be a reason to reconsider the need for return fans.)

6. The varying impact of cold-air designs on chiller energy reflects the complex relationship between building utilization, climate, airside design, and the intelligent reset of supply-air temperature. It is also a function of chilled-water-plant strategy; sometimes trading chiller energy for condenser-water-pump energy and cooling-tower energy. A low-flow, low-temperature, cold-air system that increases chiller energy by 2¢ to 3¢ per square foot in one application may reduce chiller energy by as much as 1¢ to 2¢ per square foot in another.

Also the colder supply air temperature requires more refrigeration work and reduces the number of hours in a year when economizer operation can be used. For example, lowering the supply air setpoint from 55°F to 50°F removes the opportunity to cool the building with outdoor air when the ambient dry bulb is between 55°F and 50°F. With integrated economizers, some cooling effect can be gained, but supplemental mechanical cooling will be required.

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## What type of Control Sequence is required for the Cold Air Distribution Systems?

### 1) Temperature and Humidity Control

Cold air systems require temperature resetting because of two primary reasons:

***Part load performance during low load conditions:*** Cold air distribution systems are designed to deliver supply air to terminal devices at temperatures ranging from 38°F to 42°F during full load operating conditions. Since the air-conditioning systems operate at part load conditions the majority of the time, it is normally desirable to reset the supply air temperature so that the conditioned spaces are not overcooled.

***Temperature resetting for dehumidification:*** Attention must also be given to humidity control during the cooling season. If the supply air temperature is increased too high, and latent heat is being generated in the conditioned room, the humidity will increase to an undesirable level. The humidity control requirement, therefore, suggests the use of a supply air temperature override system based on return air humidity, or dew point.

The control strategy is based on the type of HVAC design employed. Two common types of designs are:

1. Constant Air Volume (CAV) System

2. Variable Air Volume (VAV) System

**Constant Air Volume (CAV) System:**

As the name implies, CAV systems deliver a constant air volume to the conditioned space irrespective of the load. When the load conditions (indoor temperature/humidity) vary from the setpoint, the CAV system responds by varying the temperature. This is achieved by modulating the chilled water flow rate through the cooling coil through a two-way control valve. The valve shall be fully open when the cooling is desired and close in the warming mode. The direct expansion system responds by loading-unloading or ON-OFF of the refrigeration compressor.

CAV systems are typically used in single zone application having similar cooling requirements throughout its occupied area so that comfort conditions may be controlled by a single thermostat.

**Variable Air Volume (VAV) System:**

The VAV air-conditioning system changes the quantity of air supplied to the space in response to changes in loads. The variable airflow volume is achieved by the VAV terminal boxes. The boxes have a modulating damper that throttles in response to the thermostat setting. When the indoor temperature conditions vary from the set point, the VAV box damper responds by restricting or increasing the supply air volume to the space.

The supply air fans shall have their air flow rates controlled by a variable frequency drive which gets a signal from the duct static pressure sensor(s). Airflow reduction brings about a corresponding reduction in fan horsepower and therefore the VAV systems are considered much more energy efficient.

VAV systems are typically used in multi-zone application having different cooling requirements throughout their occupied areas. The comfort conditions are maintained by using independent setback thermostats, thereby providing the opportunity to control comfort levels in each zone.

### 2) Outdoor Air Ventilation Control

Fresh outdoor air must be continually introduced into a building in order to maintain indoor contaminates below an acceptable level. The ASHRAE 62-1989 standards on Indoor Air Quality suggest minimum ventilation rate of 20 CFM air per occupant. This requirement is the same for both conventional and cold air systems.

The standard also states in Section 6, Paragraph 6.1.33 that outdoor air introduction rates must be increased, if mixing provided by the building's air distribution system is less than 100% effective. Ventilation effectiveness will be less than 100% unless complete mixing occurs between the primary supply air and the room ambient air (i.e. short-circuiting to the return air and/or the exhaust air system). Conventional diffusers typically fall short of complete mixing; an "$E_V$" can be as low as 60%. Properly applied inductive or swirl diffusers will always provide complete mixing and an "$E_V$" factor of one.

The HVAC design with a VAV system may present serious indoor air quality concerns. As airflow is reduced from design quantities under part-load conditions, it also means reduced ventilation airflow as well. Therefore a reliable control strategy is needed to ensure minimum ventilation air rate all the time. Two control scenarios are as follows:

**Constant Rate Outdoor Air Introduction:**

Outdoor air shall be introduced either by combination motorized outdoor air/return air mixing dampers, or a variable rate outdoor air fan controlled by a signal from an air flow measuring station that will maintain a constant rate of outdoor air introduction independent of the main supply air fan's point of operation.

**Variable Rate Outdoor Air Introduction:**

In some designs, the $CO_2$ sensors are used to monitor and control the volume of outdoor air introduction in sufficient quantities to maintain $CO_2$ concentrations below 1,000 PPM. This method is effective in ensuring energy conservation of the HVAC system by optimizing the introduction of outdoor air.

Outdoor air shall be introduced by either combination motorized outdoor air/return air mixing dampers, or a variable rate outdoor air fan controlled by $CO_2$ sensors located in each main return air duct. The outdoor air introduced shall be completely independent of the main supply air fan's point of operation.

### 3) Morning Reset

When the HVAC system is off for extended periods, say during night time, the building could be very hot and humid during the morning start hours. In this case, it may be desirable to limit the supply air temperature to 55°F for a period of time, to avoid any possible condensation from forming on the system's components.

*Supply Air Temperature Reset-Startup morning Hours*

When the air handling unit is cycled on during morning cool down, the supply air temperature will be limited to a minimum of 55°F for the first hour of operation. When the first hour of operation has terminated, the leaving air temperature will be controlled by its normal operating schedule. The supply air temperature will be reset from outdoor air temperature in accordance with the following schedule:

| Outdoor Air Temperature | Leaving Air Temperature |
|---|---|
| Above 85°F | Design Leaving Air Temperature |
| Below 85°F | Reset Ratio = 1:1 |

Whenever the space humidity rises above set point (50 to 55%) as determined by a return air dew point sensor, the supply air temperature will be decreased. That is, unless any VAV terminal unit is at its minimum flow rate and the room temperature is below its winter set point, then the reset ratio for the humidity controller will be 2:1.

*When the air handling unit is cycled on during the morning cool down or warm up period, the fresh air volume will be zero.*

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## Other Design Considerations

### Blow-through or Draw-through Air-handling Arrangement

The main objective of cold air system is to increase the temperature difference between the supply air and the space temperature to reduce the required supply air volume. The supply air temperature is the temperature of the air as it leaves the air handling unit and enters the ductwork – not as it leaves the coil.

This is an important consideration because a supply fan will add enough heat to raise the supply air temperature about 2°F to 3°F. Because blow-through air handling units have the supply fan upstream of the cooling coil, their leaving air temperature off the cooling coil is the same as the supply air temperature as it enters the ductwork. On the other hand, draw-through units add the fan heat downstream from the cooling coil. Both draw-through and blow-through arrangements will work in cold air systems, but with draw-through arrangement the coil leaving air temperature must be 2°F to 3°F lower than the supply air temperature to compensate for the fan motor heat.

*The sensible heat ratio provided by blow-through equipment is a good match for buildings with high sensible heat ratios (such as office buildings).*

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## Myth and Realities

***Myth #1:*** *It is a perceived notion that the cold air systems will result in sub cooling and require reheat to raise the temperature of air.*

This is not correct. Rather it is true for conventional systems. Let's check why.

With conventional *55°F* systems, the humidity usually increases at part load conditions and the system works continuously to process high amount of moisture. (The part load condition refers to the situation when the building's heat load is low compared to what system is designed for or in simple words the building do not have enough sensible load (cooling needs) but has to process the constant latent load (moisture) of the outside air). The humidity control takes precedence over temperature control and will necessitate the cooling coil to work at full flow. This results in sub cooling which will require consequent reheat to maintain indoor temperature unless variable volume system (VAV) system is provided. On the other hand, the cold air systems use high row deep cooling coil which offers more surface area for moisture to condense. The condensation on the cooling coil tubes is up to three times faster than the conventional systems and, as a result, the temperature and the humidity control sequence is almost simultaneous. No reheat is thus required.

***Myth#2:*** *The other perceived notion is that the humidity may drop much lower with the cold air system and it may require humidification to prevent static electrical charges.*

This is also not true. With a proper-engineered system, processing high quantities of outside air for pressurization needs, the low humidity conditions are rare. As soon as the humidity reaches below the set point, the controllers on the chilled water coil will block the flow partially or completely to maintain the indoor conditions. There is no doubt that the cold air systems remove moisture much more efficiently but with dry ambient conditions, the problem of humidification is similar for both conventional and the cold air systems.

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**APPLICATION CHECKLIST**

This checklist identifies the basic attributes of an HVAC system, designed for effective cold-air distribution.

### Equipment room

1. Plug all openings in the mechanical room.
2. Consider pressurizing and dehumidifying the equipment room with a small volume of supply air.
3. Duct outdoor air directly to the air handler; do not use the equipment room as a plenum.
4. Use the ducted return. If the return air is un-ducted, use a return fan to pressurize the equipment room.
5. Insulate the cold surfaces and vapor-seal all the openings such as ductwork, drain pipes and chilled water pipes (leaving and entering).

### Duct Design

1. Design for highest practical air velocity to minimize duct heat gain and insulation cost.
2. Install spiral duct wherever possible; single wall or double with external insulation.
3. Specify duct leakage test. Recommend leakage rate not exceeding 1% at duct operating pressure in accordance with SMACNA test procedures.
4. Specify low temperature VAV terminal units with isolated inlets, sealed damper shafts and insulation to prevent condensation.
5. Specify composite "non-sweating" volume dampers.
6. Specify double wall insulated inspection doors.
7. Insulate all ductwork and specify insulated duct hangers.

### Air Handling Units

1. Size the cooling coil at velocities of 500 FPM or lower. Higher velocities shall cause moisture carryover and noise problems.
2. Place AHU inside an enclosed space within complete vapor barrier.
3. Use dual-slope drain pans to prevent standing water
4. Condensate pans shall be one piece with fully welded corners, be of the IAQ design and pitched in two directions for positive draining.

5. Plan for proper trapping: the air handler must be mounted high enough above the floor to accommodate total trap height and depth.
6. The entire unit casing shall be of double-wall construction complete with cold insulation and vapor barrier to completely eliminate condensation. The entire access door and frame assembly shall be insulated and the design must include a thermal break.
7. Gasket all access panels, door openings, and inspection windows in positive-pressure sections
8. Insulate all components including frame members and use a specially designed vapor barrier provided to completely eliminate condensation from forming with supply air temperatures down to 38°F and the ambient DB and WB temperatures air as required by location.
9. Specify sealing at all air-handler penetrations, including connections for coil piping and electrical service.

### Terminal devices

1. Select high-aspiration diffusers that work on induction principle.
2. Select linear slot diffusers with a high-aspiration ratio to provide proper air movement.
3. When using regular diffusers, install fan-powered VAV terminals in all zones.
4. Specify VAV terminals with gasketed panels and insulated surfaces.

### Intelligent control strategies

1. Maintain positive building pressure in cooling climates to prevent infiltration.
2. Use set-point reset for supply-air temperature to eliminate/minimize reheat.
3. Provide set-point reset for supply-air static pressure to minimize fan energy and improve zone control.
4. Operate fan-powered VAV terminals continuously during occupied periods and when primary airflow is less than 20 to 30 percent of design.

### General

1. Buildings with high sensible heat ratios are excellent candidates for cold air system.
2. The centrifugal and the rotary chillers available today in the market are very efficient with operating energy rate of as low as 0.5 kWh per ton. This has changed the

overall scenario of analyzing the energy results. The major proportion of energy consumption in many buildings is no longer the chiller but the fans and the pumps. Significant energy saving can be achieved with the cold air systems in large setups.

3. With cold air design, place all HVAC equipment within the vapor barrier. Use a vapor retarder on the warm side of perimeter walls to minimize vapor-pressure diffusion.
4. Use specially designed cooling coils, DX systems, and low temperature chiller (preferably rotary screw) due to wide compression ratios, and along with thermal energy storage.

*Applications where cold-air distribution should be used with caution include cases where:*

1. Generation of chilled fluid at 34 to 40°F (1 to 4°C) is not practical.
2. Space relative humidity must be maintained above 45%.
3. High volumes of ventilation air are required.
4. Economizer cooling is available with outdoor temperatures of 45 to 55° F for many hours of the year.
5. Do not install the cold-air system in unconditioned spaces such as attics or plenums.

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**Summary**

The colder temperature allows primary air volumes to be reduced, compared to a conventional 55°F supply air design. While the conventional 55°F maintains an acceptable comfort conditions in a building, it does not necessarily offer the best annual energy performance or capital cost.

The cold air distribution system provide more cooling with lesser air. The benefits are that the AHU and duct sheet metal sizes are dramatically reduced, which in turn reduces first cost and operative cost of the HVAC system. Additional cost reductions are due to additional space created due to smaller equipment, reduced electrical requirements, reduced structural load requirements, and the reduction of the mechanical rooms.

The downside is that the colder air system can lead to the condensation on the duct surfaces. Dumping of cold air and balancing becomes difficult. These aspects require careful attention and are most often mitigated by the appropriate selection and installation of HVAC equipment.