HVAC – Guide to Demand Control Ventilation

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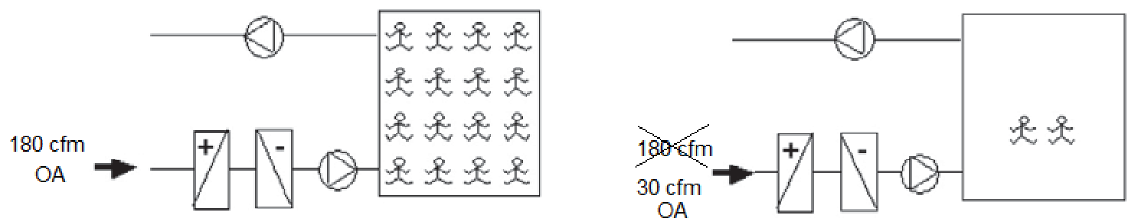
HVAC – GUIDE TO DEMAND CONTROL VENTILATION

Most building codes require that a minimum amount of fresh air be provided to ensure adequate air quality. Typically, most ventilation systems set up the fresh air intake at a fixed rate when they are installed, irrespective of the occupancy. This sometimes leads to poor indoor air quality (IAQ) and incurs penalty on energy consumption in cooling, heating and dehumidification. Good IAQ can be maintained when the fresh air supply rate responds to the load imposed by the number of people and by their activity in the room.



To make rational use of energy, the ventilation rate can be reduced when the spaces are only partially occupied. Demand-Controlled Ventilation (DCV) is a ventilation control strategy that provides automatic reduction of outdoor air intake below design rates when the actual occupancy of spaces served by the system is less than design occupancy. DCV involves ventilating and conditioning the air precisely to meet our needs; no more and no less. The potential for savings is substantial, especially in premises such as offices, classrooms and hotel rooms where there is considerable variation between high and low during times when there are few or no occupants. DCV offers great potential for both new as well retrofit projects, but if improperly applied, it can create a negative building pressure leading to undesirable infiltration, building envelope degradation, and indoor air quality (IAQ) problems.

This course provides the necessary background to understand how the DCV operates and how it is applied under current codes and standards. The course is divided into 8 Sections:

SECTION - 1:	CO_2 and Indoor Air Quality
SECTION - 2:	Design Ventilation Rates
SECTION - 3:	Outside Air Control
SECTION - 4:	Applying DCV to HVAC Systems

SECTION - 1: CO_2 and INDOOR AIR QUALITY

Ventilation is the process of bringing outside air into a building. Depending on weather conditions, ventilation air must usually be either heated, cooled, and/or be dehumidified. Because of this, ventilation air represents a significant portion of HVAC energy consumption. Maximum ventilation rates or the amount of fresh air in cubic feet per minute (cfm) that an air handler system brings into a building is provided in proportion to the maximum design occupancy of the building. In reality, the actual occupancy rarely approaches the maximum design occupancy and it is not unusual for an air handler to operate at the maximum ventilation rate continuously, even if the space is only partially occupied. This often results in over-ventilation, thereby resulting in higher-than-necessary energy costs.

Demand Controlled Ventilation (DCV)

Buildings do not require the induction of 100 percent fresh air all of the time. As the number of people in a building varies at different times, so should the demand for fresh air too. The requirement for fresh air can be lower at the times of the day when fewer people are in a building. Demand Control Ventilation (DCV) is a method of introducing variable amounts of fresh air "on demand" based on actual occupancy patterns. The system provides a means to adjust the rate of ventilation continuously and automatically. Essentially, the control is achieved by means of a sensor (or a series of sensors) which respond to the variation in occupancy. Output from the sensor is applied to a control system (usually damper) that adjusts the rate of outdoor air flow through the ventilation system, thus ensuring that good air quality is continuously maintained. Three primary strategies are often used:

1. **Occupancy schedules:** supply design outdoor air during occupied hours as scheduled, with minimum or no outside air during unoccupied hours. Simple timers may be used to switch the ventilation system on or off at set times.

2. **Occupancy sensors:** supply design outdoor air during occupied times as sensed, with minimum or no outside air when zone is unoccupied. System may use proximity detectors and counters that can control the rate of ventilation according to the detection of occupancy and number of occupants in a space at any time.

3. **CO_2 sensors:** supply sufficient outdoor air to keep CO_2 concentration within bounds.

The first two strategies rely on estimates and approximations and do not necessarily yield guaranteed results. Still these strategies are significant improvements over supplying a fixed quantity of outdoor air. Most modern DCV systems use carbon dioxide (CO_2) sensors to continuously monitor the indoor CO_2 levels and provide real time feedback to regulate the amount of fresh air admitted for ventilation. The use of CO_2 sensors allows much better control over ventilation and is recognized valid by the most model building codes including ASHRAE Standards. The U.S. Green Building Council gives points in its Leadership in Energy and Environmental Design (LEED™) rating system for use of CO_2-based ventilation control in buildings.

Benefits of CO_2- Based DCV

During heating and cooling periods, energy is required to add or remove heat to fresh air introduced into a building. Over-ventilation is one of the largest indirect contributors to a building's energy use. The fact that a majority of Canadian and U.S. buildings deliver fresh air to the building's occupants at a fixed or constant volume represents the heart of the business case for Demand Controlled Ventilation (DCV). Compared to a fixed ventilation approach, DCV offers considerable advantages:

1. Excessive over-ventilation is avoided while still maintaining good Indoor Air Quality (IAQ) and providing the required cfm-per-person outside air requirement specified by codes and standards.
2. Saves energy by avoiding the heating, cooling, and dehumidification of more ventilation air than is needed.
3. Provides an opportunity to monitor both occupancy and ventilation rates in a building all the time.
4. Provides valuable information about occupancy trends, which can be useful for business analysis, operational & maintenance planning of equipment and ensuring safety in the premises.
5. In some buildings, infiltration air or open windows may be a significant source of outside air. A CO_2 sensor will consider the contribution of infiltration in a

space and only requires the mechanical system to make up what is necessary to meet required ventilation levels.

6. When integrated with the appropriate building control strategy, ventilation can be controlled zone by zone based on actual occupancy. This allows for the use of supply air from under-occupied zones to be redistributed to areas where more ventilation or cooling is needed.
7. A CO_2 control strategy can be issued to maintain any per-person ventilation rate. As a result, this approach is highly adaptable to changing building uses and any changes that may occur in future recommended ventilation rates.

According to the observations, the savings range from 5 to 80 percent depending on the application and ambient conditions. System paybacks from CO_2-based DCV will be greatest in higher density spaces, where occupancy constantly changes (e.g. schools, theaters, retail establishments, and meeting/conference areas).

CO_2 LEVELS & VENTILATION

People continuously exhale predictable quantities of CO_2 as they breathe. If the number of people in the space is doubled, the amount of CO_2 produced will double. Because CO_2 production is so consistent and predictable, it can be used as a reliable indicator of the air quality and ventilation rate. A high level of CO_2 in a room (>1000ppm) indicates insufficient ventilation and a low level (<600ppm) may suggest that the ventilation rate could be turned down while maintaining a satisfactory IAQ with lower energy costs.

What is a good CO_2 level?

There is no single good value of CO_2 level. Many studies have been performed on human perception to establish the relationship between optimum CO_2 levels and occupant comfort, and the studies show that a 20% dissatisfaction criterion corresponds to a CO_2 level of 1000 ppm. In other words, when the CO_2 level is above 1000 ppm, 20% of the people will find the air quality unacceptable.

ASHRAE Standard 62–2001, Section 6.1.3 states that comfort (odor) criteria is likely to be satisfied if the ventilation rate is so set that the 1,000 ppm of CO_2 is not exceeded. The absolute 1,000 ppm value was often interpreted as the *ceiling* CO_2 concentration for acceptable indoor air quality. But since, an indoor CO_2 measurement is a dynamic measure of the number of people in a space, it is not appropriate to go for the absolute value of 1000 ppm. Rather it is much more logical to determine the cfm/person ventilation rates by measuring the indoor-outdoor CO_2 difference. The 2004 edition of ASHRAE 62 revised wording of Section 6.1.3 specifically to include 700 ppm difference between indoor and outdoor CO_2 concentrations as an acceptable level of human bio-effluents. This value is based on a specific ventilation rate (15 cfm/person), activity level (1.2 MET*) and outdoor CO_2 concentration of 300 ppm.

*The CO_2 generation from people is function of activity level. The term used to define the activity level is the "MET", which stands for "metabolic equivalent task". The higher the duration and intensity of the physical activity, the larger will be the oxygen consumption and the larger will be the exhaled quantity of CO_2.

Important!

1. Carbon dioxide is not a contaminant in occupied spaces. This a major misconception of many that use CO_2 levels to reset the outside airflow rate. The table below shows various carbon dioxide thresholds and their descriptions.

CO_2 Level (ppm)	**Description**
90,000	NIOSH, lethal after 5 minutes
40,000	OSHA, immediate danger to life
30,000	OSHA, Safe for 10 minutes
5,000	OSHA, Safe for 8hr/day – 40hr week
1,000	Recommended indoor level (odor)
350 - 450	Average outside ambient level

   It is important to note that the 1,000 ppm level is a recommendation and not a ceiling.

2. An elevated indoor CO_2 concentration is related to the occupants in the building, the building's ventilation rate, and the CO_2 level in the outside air.

3. Indoor CO_2 can accumulate if ventilation is not adequate to dilute and remove the CO_2 that is continuously generated by building occupants.
4. CO_2 measurement does not provide the count of people; it can be used only as an indicator of occupancy pattern and is only a measure of effective ventilation.

CO_2 differential and ventilation rates

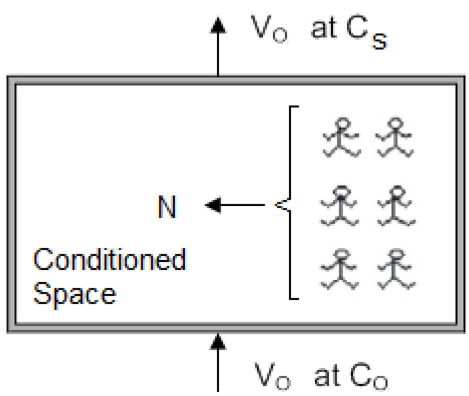
If the ventilation rate in an occupied space decreases, the carbon dioxide concentration will begin to increase and vice versa. Once people enter a room, the CO_2 concentration will begin to increase. This level will continue to increase until the amount of CO_2 produced by the space occupants and the dilution air delivered to the space are in balance. Such a state is called the "equilibrium" point.

To understand how CO_2 sensors can be used to control ventilation, consider a conditioned space ventilated with outdoor air.

CO_2 enters the conditioned space in the ventilation air. The quantity of CO_2 in the ventilation air is the product of the outdoor air flow rate Vo and the concentration of CO_2 in the outside air Co.

CO_2 is generated in space from the occupants at a rate N.

CO_2 leaves the space in the exhaust air at concentration Cs.



A steady-state mass balance on the CO_2 entering and leaving the zone gives:

Vo * Co + N = Vo * Cs

This relation can be rearranged to give the ventilation rates as:

$$V_O = \frac{N}{(C_S - C_O)}$$

Where,

- V_o = outdoor airflow rate, cfm/person
- C_s = CO_2 concentration in the space, ppm
- C_o = CO_2 concentration in the outdoor air, ppm
- N = CO_2 generation rate, cfm/person

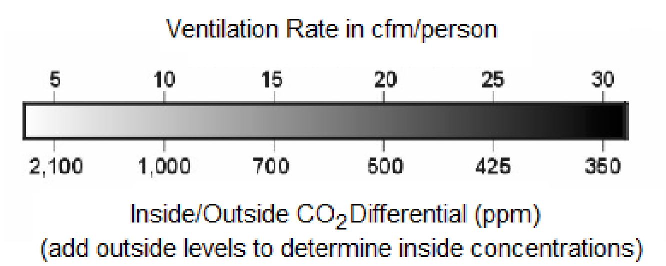
The CO_2 emitted from a typical person at light work "N" is about 0.0106 cfm/person corresponding to a typical activity level of 1.2 Met. If the maximum space concentration is to be held to 0.1 percent, 1000 ppm, and the outdoor concentration is 0.03 percent, 300 ppm, the minimum ventilation rate shall be:

$$V_O = \frac{0.0106}{(0.001 - 0.0003)}$$

Vo = 15.14 cfm per person

The example shows the CO_2 concentration in occupied space will not exceed 1000 ppm as long as 15.14 cfm per person of outdoor air with outdoor CO_2 concentration of 300 ppm is continuously being added to the space. This corresponds to maintain inside/outside CO_2 differential (Cs – Co) of 700 ppm. An inside/outside differential of 500 ppm is indicative of a ~20 cfm/person ventilation rate.

The figure below represents the relationship between the CO_2 differential (inside/outside) and the target ventilation rates, cfm per person.



The equation [$V_O = N / (C_S – C_O)$] can be simplified and converted to volumetric units of cfm and concentrations in ppm.

$V_O = 10{,}600 \div (C_S – C_O)$

Example:

What differential CO_2 setpoint must be maintained for a Met level of 1.2 in order to ensure 20 cfm/person of ventilation air?

(Cs - Co) = 10,600 ÷ 20 cfm OA per person

(Cs - Co) = 530 ppm CO_2 fixed differential.

Example:

What will be the indoor CO_2 concentration level, if ventilation rate is maintained at 15 cfm per person and the outdoor CO_2 concentration is 350 ppm? Assume steady equilibrium conditions with Met level of 1.2 or CO_2 generation rate of 0.0106 cfm/person.

Solution:

Mass-balance equation can also be restated so that the indoor space for a particular ventilation rate can be calculated using equation:

$$C_S = C_O + \frac{N}{V_O}$$

$$C_S = 350 + \frac{10600}{15}$$

Cs = 1056 ppm

Important!

1. The mass-balance equation assumes steady-state conditions within the space. Steady state implies that the occupants are generating CO_2 at a constant rate and their metabolic rate, diet, and level of activity are identical. According to Appendix D of ASHRAE Standard 62-1989, this generation is equal to 0.0106 cfm of CO_2, which corresponds to an activity level of 1.2 MET units and applicable to typical office activities. If the level of activity were more strenuous than that of typical office work, then the metabolic rate would go up with a corresponding increase in the CO_2 generation rate. For the same ventilation rate then, this increased level of activity would therefore result in an increased build-up of CO_2 concentrations.

2. The correlation between indoor/outdoor CO_2 differential and ventilation rate is independent of the volume of a room and the population density. However, both the volume of a room and population density will affect the time it takes for CO_2 to build up to an equilibrium level. The mass balance equation can be only applied when equilibrium conditions exist. This is particularly important when trying to infer space ventilation rates from a spot measurement when non steady-state conditions exist. To make an accurate determination of outdoor cfm per person rates one should take CO_2 measurements when the occupancy has stabilized. Measuring CO_2 concentrations that are still in transition to an equilibrium level can result in overestimation of the ventilation rate.

The Importance of Considering Inside/Outside Differential

Outside CO_2 concentrations must be considered when implementing CO_2 ventilation control strategy. The mass balance equation is based on the differential between the inside and outside CO_2 concentration and NOT on controlling indoor CO_2 to an absolute value.

As discussed before, a 700ppm differential is equal 15 cfm/person ventilation rate, if outside CO_2 level were 300ppm. If 1,000ppm were maintained in a building, where outside CO_2 concentrations were 600ppm (e.g. down-town city area), the actual ventilation required to maintain a 400ppm differential in CO_2 concentrations between inside and outside would be over 25 cfm/person rather than the 15 cfm/person intended in the ASHRAE Standard. This is not very reasonable. The use of an arbitrary set-point for CO_2 control without considering outside concentrations could result in excessive over-ventilation of the space.

There are two approaches that can be used in integrating outside CO_2 conditions into a control strategy. Prior to implementing a control strategy, outside concentrations can be monitored and data logged for a week or more to determine an appropriate outside CO_2 baseline for that geographic location. The average concentration measured during the proposed occupied hours of the building can be assumed to be the outside concentration. The control point for sensors within the building can be based on the differential between inside concentrations and the outdoor baseline. In some cases the local EPA Air pollution branch may also have data on local CO_2 levels.

A second approach can be utilized where an HVAC system is operated as part of a computerized building management system. Carbon dioxide concentrations can be directly monitored in the outside air intake of the building. Ventilation control can be based on the real-time differential between inside and outside conditions. This approach is highly recommended where outside concentrations appear to be consistently varying more than 200~300ppm over the course of a day. Variations of this magnitude may occur if the building air intake is close to a major highway, vehicle idling area or loading dock. It may also occur in highly polluted cities.

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SECTION - 2: DESIGN VENTILATION RATES

ASHRAE Standard 62 is and has been a consensus standard for determining ventilation rates for acceptable indoor air quality. This standard provides system designers a choice of two procedures for determining the design outdoor air intake flow for the building's ventilation system – the Ventilation Rate Procedure (VRP) and the Indoor Air Quality (IAQ) procedure. The VRP is a prescriptive method based on occupancy category and density. The IAQ procedure is performance-based in that the ventilation system is designed to maintain the concentrations of specific contaminants at or below certain limits.

Most building engineers use the VRP approach. ASHRAE Standard 62 - (versions 1989 through 2001) "Ventilation for Acceptable Air Quality" requires minimum ventilation based on the number of occupants only. Refer to the table below.

ASHRAE 62-89 Recommended Ventilation Rates

Application	Ventilation Rate/Person	Application	Ventilation Rate/Person
Office Space	20 cfm	Smoking Lounge	60 cfm
Restaurants	20 cfm	Beauty Salon	25 cfm
Bars/Cocktail	30 cfm	Supermarkets	15 cfm
Hotel Rooms	30 cfm/room	Auditorium	15 cfm
Conference Rooms	20 cfm	Classrooms	15 cfm
Hospital Rooms	25 cfm	Laboratory	20 cfm
Operating Rooms	30 cfm	General Retail	15 cfm

The total design ventilation rate is worked out on the expected occupancy level multiplied by the ventilation rate.

$DV = V_P \times P$

Where,

- DV = The design ventilation value (cfm)
- V_P = required outdoor airflow rate per person, cfm/person
- P = zone population or the largest number of people expected to occupy the space.

Obviously, if there is an error in estimating the maximum number of people, it will cause an error in the ventilation rates.

The ASHRAE standard for ventilation, ASHRAE 62.1 (2004 and 2007) incorporates both occupancy and an area based component to estimate the design ventilation rate (DVR). The equation is as follows:

$DVR = (V_P \times P) + (V_A \times A)$

Where,

- DVR = The total upper limit of outside air required (cfm)
- V_P = Outdoor airflow rate per person, cfm/person
- P = Zone population or the largest number of people expected to occupy the zone during typical usage.
- V_A = Outdoor airflow rate per unit area, (typically 0.06 to 0.12 cfm/ft^2 depending on space type, refer table below)
- A = Zone floor area, ft^2

The occupancy component is required to dilute and remove metabolic pollutants which arise as a result of occupancy.

The area component of the requirement is to purge the building contaminants from outgassing of building materials, fabrics and furnishings, etc., irrespective of the occupancy. The base ventilation airflow rate is also required to balance supply, exhaust and building pressurization requirements.

The ASHRAE Standard 62.1 – 2007, Table 6-1 (reproduced below) is used by building engineers to determine ventilation rate requirements for outdoor air.

Occupancy Category	cfm/person	cfm /ft^2	Default combined outdoor air rate Note -2 (cfm/person)	Default occupant density Note -1 (#/1,000 sq-ft)
Classrooms (age 9 plus)	10	0.12	13	35
Lecture hall	7.5	0.06	8	150

Cafeteria	7.5	0.18	9	100
Conference/ meeting	5	0.06	6	50
Multipurpose assembly	5	0.06	6	120
Office space	5	0.06	17	5
Courtrooms	5	0.06	6	70

Note-1: Default Occupant Density: The default occupant density shall be used when actual occupant density is not known.

Note-2: Default Combined Outdoor Air Rate (per person): This rate is based on the default occupant density.

Example

Calculate the design ventilation rate for a classroom with 900 square feet of occupiable area and a class size maximum of 30 students:

$DVR = (V_P \times P) + (V_A \times A)$

DVR = (10 cfm/person x 30 students) + (0.12 cfm/ft^2 x 900 ft^2)

DVR = 300 cfm + 108 cfm = 408 cfm

This results in 13.6 cfm / student.

Historical Development of AHSRAE STANDARDS

Let's see how the ventilation rates based on ASHRAE 62 (1989 thru 2001 version) will compare with ASHRAE 62 (2004 - 2007 version) and how these affect the DCV.

ASHRAE 62- (1989 thru 2001)

Consider a lecture classroom of 1000 sq.-ft with a design population of 80. ASHRAE 62-1989 thru 2001 required 15 cfm of outdoor air per person for this space.

Therefore, the classroom must receive 1200 cfm of outdoor air (15 cfm / person × 80 people).

Assuming the CO_2 generation rate to be 0.0106 cfm per person and the design ventilation rate to be 15 cfm per person, the resulting indoor-to-outdoor CO_2 concentrations differential will be 700 ppm.

$C_S – C_O = N / V_O$

$C_S – C_O = 0.0106 /15 = 0.0007$ [or 700 ppm]

If the population drops to 40, the required quantity of outdoor air drops to 600 cfm (15 cfm/person × 40 people). In both cases, the classroom receives the same rate of outdoor airflow per person; that is 15 cfm/person. Therefore, the differential between indoor and outdoor CO_2 concentrations remain constant too.

By controlling to this constant differential, Cs – Co, CO_2-based demand-controlled ventilation can maintain the same per-person ventilation rate (Vo) to the space during periods of reduced occupancy.

ASHRAE 62 (2004-2007)

Let's revisit the classroom example.

The classroom requires 7.5 cfm of outdoor air per person **plus** 0.06 cfm of outdoor air per square foot of floor area. (Refer to ASHRAE Standard 62.1 - 2007, Table 6-1, explained in the previous section).

With a design population of 80 and a floor area of 1000 ft², the design ventilation rate (DVR) is 660 cfm of outdoor air.

$DVR = (V_P \times P) + (V_A \times A)$

DVR = 7.5 cfm/person × 80 people + 0.06 cfm/ft² × 1000 ft² = 660 cfm

Here, the classroom receives 8.25 cfm/person [585/80 = 8.25].

Assuming the CO_2 generation rate of 0.0106 cfm per person and the design ventilation rate of 8.35 cfm per person, the resulting indoor-to-outdoor CO_2 concentrations differential will be 1240 ppm.

$C_S – C_O = N / V_O$

$C_S – C_O = 0.0105 /8.25 = 0.00127$ [or 1270 ppm]

At 50% occupancy, with 40 people in the classroom, the required quantity of outdoor air drops to 360 cfm.

DVR = 7.5 CFM/person × 40 people + 0.06 cfm/ft² × 1000 ft² = 360 cfm

Here, the classroom receives 9 cfm/person [360/40 = 9].

Assuming the CO_2 generation rate of 0.0106 cfm per person, the desired difference in indoor-to outdoor CO_2 concentrations drops to 1160 ppm.

$C_S – C_O = N / V_O$

$C_S – C_O$ = 0.0105 /9 = 0.00116 [or 1160 ppm]

At 25% occupancy with only 20 people in the classroom, the required quantity of outdoor air drops to 210 CFM.

DVR = 7.5 CFM/person × 20 people + 0.06 cfm/ft² × 1000 ft² = 210 cfm

Here, the classroom receives 10.5 cfm/person [210/20 = 10.5].

Assuming the CO_2 generation rate of 0.0106 cfm per person, the desired difference in indoor-to outdoor CO_2 concentrations drops to 1000 ppm.

$C_S – C_O = N / V_O$

$C_S – C_O$ = 0.0105 /10.5 = 0.001 [or 1000 ppm]

At zero occupancy the required quantity of outdoor air drops to 60 cfm.

DVR = 7.5 cfm/person × 0 people + 0.06 cfm/ft² × 1000 ft² = 60 cfm

In this scenario, the indoor to outdoor air CO_2 concentration will match the outdoors.

Comparison of the different ASHRAE versions

The design ventilation rate (cfm per person) is lower in ASHRAE 62 (2004 – 2007) standards and it varies as:

- 8.35 cfm per person @ 100% occupancy
- 9 cfm per person @ 50% occupancy
- 10.5 cfm per person @ 25% occupancy

This is compared to a fixed 15 cfm per person with ASHRAE 62 (1999 thru 2001).

The minimum design ventilation value is 660 cfm in ASHRAE 62 (2004 – 2007) compared to 1200 cfm of ASHRAE 62 (1989 -2001). This means the air handling system shall be designed for lower outside air, thereby reducing the fan energy, as well as for lower energy due to cooling, heating and dehumidifying of less outside air.

This implies a cost saving. If these comparisons were made on densely occupied spaces, such as auditoriums, gyms, conference rooms, and cafeterias, the difference will be significant.

Because ASHRAE 62- (2004 and 2007) has introduced an area based component, it is clear from the aforementioned example that we can reduce the outside air ventilation quantity an amount equivalent to the area based component (60 cfm). This becomes the lower minimum value to be delivered irrespective of the occupancy. If we then apply this concept to DCV, we will require two positions for the outside air damper corresponding to the upper and lower minimum ventilation rate requirements. 60 cfm is termed the minimum “base” ventilation airflow, the lowest point to which CO_2 controls may modulate the outside air damper, with the 660 cfm being the upper point.

Why maintain base ventilation airflow?

A potential problem with demand controlled sensing is that the system may only operate once the sensor detects that the pollutant concentration has reached a pre-set threshold value. Until this concentration is reached, the fresh air supply could be very low or shut off completely. In turn, this may result in the concentration of other untracked pollutants (from building materials, furnishings, etc.), increasing to unacceptable levels. It is for this reason that a DCV system must be designed to maintain a minimum ventilation airflow rate to control non-occupant related contaminants. This should not be confused with the minimum outdoor air rate required by ASHRAE Standard 62 or other codes. The base ventilation rate is the lowest point to which CO_2 controls may modulate outdoor airflow during occupied hours.

Important!

CO_2-based DCV does not affect the design ventilation capacity required to serve the space; it just controls the operation of the system to be more in-tuned with how a building actually operates.

The occupancy component of ASHRAE 62 (2004-2007) ventilation equation is required to dilute and remove metabolic pollutants which arise as a result of occupancy. It is this aspect of ventilation that can benefit most from demand control.

The area component of ASHRAE 62 (2004-2007) ventilation equation is required to purge the building contaminants from outgassing of building materials, fabrics and furnishings, etc., irrespective of the occupancy. It is the minimum base ventilation rate that should be maintained irrespective of occupancy.

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SECTION - 3: OUTDOOR AIR CONTROL

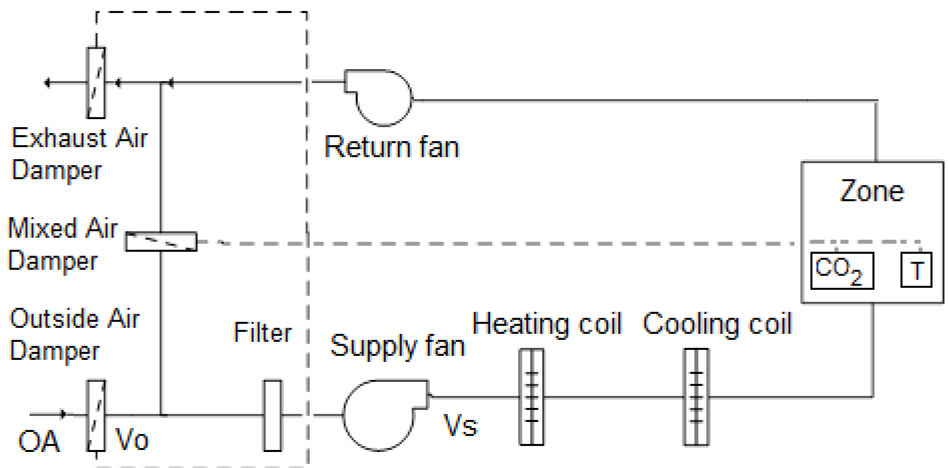
A key metric for characterizing the quantity of outdoor air introduced into a building is the "Fraction of outdoor to supply air or percent outdoor air OA%".

OA = Vo / Vs

Where,

- Vo = Outdoor air in cfm
- Vs = Total supply air through the air handler in cfm
- OA = Fraction of outdoor air to the total supply air

The fraction of outdoor to supply air in most commercial buildings is typically between 0.15 and 0.30. The OA is controlled by modulating the outside air, exhaust air and mixed air dampers in coordinated fashion in response to either the temperature or carbon dioxide levels. (Refer schematic below).



The table below shows the extremes of damper position w.r.t the fraction outside air.

OA	Exhaust Air Damper	Mixed Air Damper	Outside Air Damper
0.0	0% open	100% open	0% open
0.3	30% open	70% open	30% open
1.0	100% open	0% open	100% open

Strategies for Damper Control

1. Energy Balance Approach using temperature, "T" as an indicator.
2. Mass Balance Approach using Carbon dioxide, CO_2 as an indicator.

Energy Balance Approach

The percentage or fraction of outside air can be calculated using these three simple temperature measurements.

1. **OAT- Outside Air Temperature:** This is the temperature of the air entering the system or equipment from the outdoors.
2. **RAT - Return Air Temperature:** This is the return air entering the equipment. This temperature may be different from the temperature entering the return grilles due to duct loss or gain.
3. **MAT - Mixed Air Temperature:** This is the air temperature past the outside air inlet where the temperatures of the return air and the outside air have mixed together. This may be in the return plenum, or in the blower compartment.

Using these temperatures to calculate the percentages of outside air is called the "energy balance method" and the equation is:

OA % = [(MAT - RAT) / (OAT - RAT)] x 100

In absolute terms, the OA is calculated by multiplying the OA% to the total supply air volume handled by air handling unit.

Outside Air (cfm) = % Outside Air X Total Supply Air (cfm)

Example

To illustrate the concept, let's consider a 3 ton package unit air conditioner delivering 1200 cfm of supply air. The various temperatures with possible measurement errors are as below:

RAT: Return air temperature = 75ºF

MAT: Mixed air temperature = 80ºF

OAT: Outside air temperature = 100ºF

Using the nominal values, we calculate % OA as follows:

OA % = [(80 - 75) / (100 - 75)] X 100

OA % = (5 / 25) X 100

OA % = 0.2 X 100

OA % = 20%

In this example, 20% of the total airflow of the system is being pulled into the system from outside.

In absolute terms, the total OA = 20 x 1200cfm of supply air / 100 = 240 cfm

Application

The application of energy balance approach is the “Air side Economizer”.

When the temperature of the outside air is less than the temperature of the recirculated air, the damper adjusts to full open position to allow for free cooling and the return air is exhausted. This achieves energy economy since conditioning the outside air is more energy efficient than conditioning recirculated air.

The economizer control generates the largest savings in buildings that require cooling all year round and run air conditioning equipment to meet that load. Air-side economizers can reduce HVAC energy costs in cold and temperate climates while also potentially improving indoor air quality, but are most often not appropriate in hot and humid climates.

Mass Balance Approach - Using Carbon Dioxide (CO_2)

The "mass balance method" involves measuring the concentration of carbon dioxide in the outside air, return air and mixed air streams. The values are used in the equation below to determine the percentage of outside air.

$$OA\,\% = \frac{C_{RA} - C_{SA}}{C_{RA} - C_{FA}} \times 100$$

Where:

- C_{RA} is the carbon dioxide concentration (ppm) in the return air
- C_{SA} is the carbon dioxide concentration (ppm) in the supply air (or mixed air)
- C_{FA} is the carbon dioxide concentration (ppm) in the outside air

Let’s look at an example. An industrial hygienist takes three samples with the following values:

C_{RA} = Return air CO_2 concentration = 700 ppm

C_{SA} = Supply air CO_2 concentration = 640 ppm

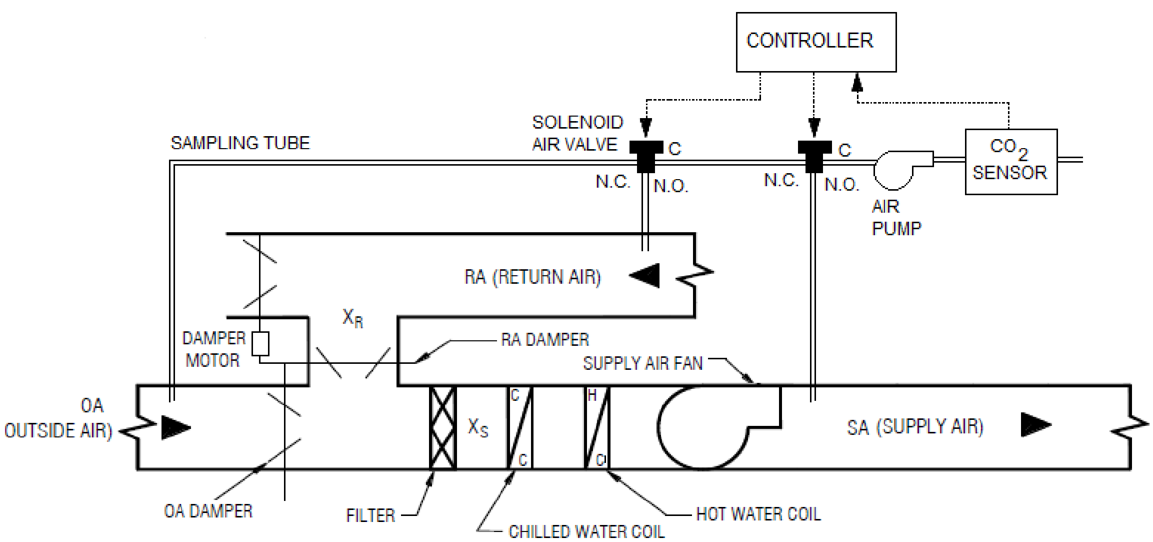
C_{FA} = Fresh air CO_2 concentration = 400 ppm

From the data above, the %OA can be calculated as:

$$OA\,\% = \frac{700 - 640}{700 - 400} \times 100 = 20\%$$

The accuracy of sensor is very important here. High tolerance in sensor accuracy exceeding ±50ppm can result in huge error. One approach to overcome this limitation is using multipoint sensing, which uses a single sensor to measure supply air, return air and outdoor air streams. With a single sensor, the inherent inaccuracy of the sensor is "cancelled" when the difference reading is taken. Note that the mass balance equation relates to the difference between CO_2 readings rather than the absolute value of the CO_2 level.

The figure below shows a multipoint sensing schematic. Here a single sensor is used to collect air samples from multiple locations. The two solenoid air valves are controlled to select any of the airstreams (fresh air, return air or supply air) drawn across a single CO_2 sensor. The controller activates the solenoid air valves according to a set time interval.



MULTI-POINT CO_2 SENSING

The advantage of the multipoint approach is that the error in differences in CO_2 levels between airstreams becomes small. For example, if the CO_2 sensor is reading high by

50 ppm because of the limitations of the sensing technology, the error between two readings taken with the same sensor is not affected.

Applications

The mass balance approach using CO_2 as an indicator is a recommended approach for Demand Control Ventilation especially when the building is crowded or fully occupied. With larger number of people in a building, the CO_2 concentration in return air will always be high and there will be sufficient differential between return and outdoor air CO_2 levels. The farther these two values are apart, the more accurate the %OA value will be. If there are few or no occupants, then there is little accumulation of CO_2 and the accuracy will not be as good.

What is difference between the Economizer and the Demand Control Ventilation?

Varying the quantity of outside air in accordance with the occupancy of a building is called demand control ventilation. Varying the quantity of outside air to take advantage of outside air conditions is called economizer control. Both types of control can reduce the load on the cooling coil.

Advanced control systems employ both demand and economizer control of outdoor air to minimize energy use while meeting ventilation requirements. As with many energy systems, the potential savings are large since the peak conditions used for sizing systems rarely occur.

Temperature Control as a Guide to CO_2 Control

There are strong similarities between temperature and CO_2 controls. In both, there is a desired control or set-point. In the case of CO_2 control, the desired set-point is the CO_2 equilibrium level that would be necessary to maintain the design ventilation rate (cfm/person) for the space being considered. The table below provides a reference chart of equilibrium set-points based on known outside CO_2 levels and the desired ventilation rate on a per-person basis.

Equilibrium Set-points for control of Ventilation using CO_2

Outside Ventilation Rate (cfm/person)	In/Outdoor Differential (ppm)	Control point at various outside concentrations							
		300	**350**	**375**	**400**	**425**	**450**	**475**	**500**
10	1060	1360	1410	1435	1460	1485	1510	1535	1560
11	964	1264	1314	1339	1364	1389	1414	1439	1464
12	883	1183	1233	1258	1283	1308	1333	1358	1383
13	815	1115	1165	1190	1215	1240	1265	1290	1315
14	757	1057	1107	1132	1157	1182	1207	1232	1257
15	707	1007	1057	1082	1107	1132	1157	1182	1207
16	663	963	1013	1038	1063	1088	1113	1138	1163
17	624	924	974	999	1024	1049	1074	1099	1124
18	589	889	939	964	989	1014	1039	1064	1089
19	558	858	908	933	958	983	1008	1033	1058
20	530	830	880	905	930	955	980	1005	1030

Carbon dioxide distribution in a space is influenced by the same factors that influence temperature distribution. The factors include convection, diffusion and mechanical air movement. Much like temperature sensors for building control, placement of CO_2 sensors should be based on the zone to be controlled and anticipated loads (e.g. common occupancy density and patterns). For optimum control, there should be a CO_2 sensor placed in every location where temperature control is contemplated. If an HVAC system is serving a series of zones with similar occupancy patterns, sensors placed in the return air ducting may be appropriate.

Control Strategies for Damper Position

Three common strategies for damper control applicable to both temperature and CO_2 as control indicators are outlined below:

Set-point Control

Set-point control employs a simple on/off or damper open/closed strategy based on the CO_2 concentration in the space. Typically, a damper would be opened at a set-point and closed when levels drop 50 to 100ppm below the set-point. This simple strategy is best applied in an application where occupancy densities are high (20 ~ 50 people per 1,000 sq ft). Ideally, occupancy range from no occupants to full design occupancy over a very short period of time. Theaters, conference rooms and some school classrooms are good applications for this strategy. On/off strategy is however not a recommended approach for changeable occupancy (such as retail stores) and large space volume; therefore, the set point control would not be a recommended approach.

Proportional control

In proportional control of ventilation systems, a CO_2 sensor emits a signal (e.g. 4 ~ 20mA) that is proportional to the CO_2 concentration. Control would typically begin when inside concentrations exceed outside concentrations by 100ppm. Air delivery to the space would increase proportionally until 100% of the design ventilation rate would be provided. Compared to set-point control, this approach allows for the ventilation system to react to varying occupancy levels much faster than waiting for CO_2 levels to build up to the desired control point.

This type of control approach is best applied in an application where occupancy densities range from 7 to 30 people per 1,000 square feet. It is ideal for ventilation control in space where occupancies are highly variable and unpredictable, such as bars, restaurants, conference rooms, courtrooms, classrooms or retail spaces.

Proportional-Integral-Derivative (Rate of Rise)

One of the potential disadvantages of CO_2 control is the time it will take for CO_2 to build up to equilibrium conditions. For densities below 6 people per 1,000 square feet it could take hours for absolute CO_2 concentrations to reach an equilibrium level. These problems can be further aggravated if occupancy is staggered or varies over the course of a day, as it is typical in many high rise buildings.

Temperature control of large complex buildings encounters similar problems as a result of unpredictable changes in outdoor temperature, solar gain and internal heat generation. Simple control approaches such as set-point or proportional control can be a disaster in a modern building. A solution to the complex control problem faced in both CO_2 and temperature control is the use of a PID (Proportional-Integral-Derivative) control. It is a system which direct control sequences to look at how far away an input is from the set-point, how long it has been at set-point, and how fast is it approaching or moving away from set-point.

PID CO_2 control views trends and CO_2 level change rates. For example, minutes after people enter a building in the morning, the HVAC system reacts to adjust fresh air delivery. This adjustment is based on actual occupancy predicted by the CO_2 level rate of rise. Much like the proportional control system, the PID controller operates based on the linear output signal from a CO_2 sensor. Most DDC and building control systems use PID control algorithms. Stand-alone controllers are available that can translate a linear signal from a CO_2 sensor into a PID signal. However, only experienced control designers and installers should apply PID control approaches. When applied properly, PID control provides fast ventilation rate response to changing occupancy conditions.

It is best integrated in high-rise, multi-zone buildings with low densities and highly variable and/or unpredictable occupancy patterns. When applied properly, a PID control strategy can provide fast response in ventilation rates to changing occupancy conditions.

CO_2 Ventilation Control Guidelines

1. Determine equilibrium concentration for zone based on design density and ventilation rate (becomes upper set-point).
2. Actively measure or assume a representative value for outside CO_2.
3. Continuously ventilate at a minimum level to control non-occupant related contaminants (i.e. 10% of design occupancy ventilation rate).
4. Select a control strategy based on occupancy density and variability

Occupancy Density	Recommended Control Strategy
Low density (7 or less people per 1,000 square feet)	Use PID or Proportional Control
Highly variable (unpredictable changes > 25% occupancy)	Use PID or Proportional Control

Medium density (7~20 people per 1,000 square feet)	Use Proportional Control
High density (20~50 people per 1,000 square feet)	Use Set-point or Proportional Control

5. For proportional or rate of rise (PID) control strategy, begin control at 100ppm above outside conditions.

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SECTION - 4: APPLYING DCV TO HVAC SYSTEMS

The essential element of CO_2 based DCV are:

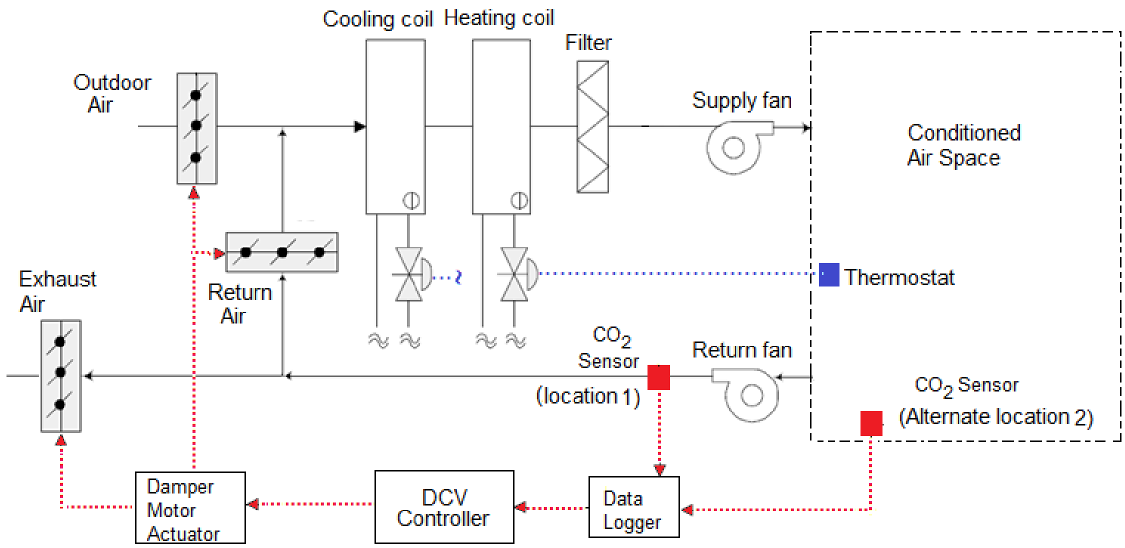
1. CO_2-sensor (or group of sensors) to detect the presence of people by virtue of metabolic carbon dioxide concentration.
2. A control system that receives signals from sensor(s) and process the signal in response to need.
3. A conventional ventilation system, usually an air-handling system, which employs dampers to regulate the amount of supply air.

DCV control can be applied to constant air volume (CAV) and the variable air volume (VAV) systems.

Constant Air Volume (CAV) System

Constant Air Volume (CAV) systems deliver a constant volume of air while varying the temperature of supply air. These systems bring outside air into air handling units (AHU) where a central fan typically draws air across heating or cooling coils and discharges into the conditioned space via a ductwork. An extract or return fan returns the air from the individual zones back to the AHU where it is re-circulated or exhausted outside.

In a CAV air handler without active damper control, the damper positions are fixed during installation of the air handler. The quantity of outdoor air introduced into the building thus remains the same. With active damper control strategy, the outdoor air dampers are typically modulated between the base minimum and design ventilation flow rates. For example, when the high level of CO_2 is detected in the space, the outside air damper will fully open to let more outdoor air into the air handler and vice versa. The figure below shows a standard control arrangement with red lines for DCV and blue for space temperature control.



CAV SYSTEM WITH CO_2 CONTROLLED VENTILATION

The DCV controller accomplishes two functions:

1. It calculates the "actual" needed outdoor air cfm using inputs from the multipoint CO_2 sensor using equation:

$$V_O = \frac{N}{(C_S - C_O)}$$

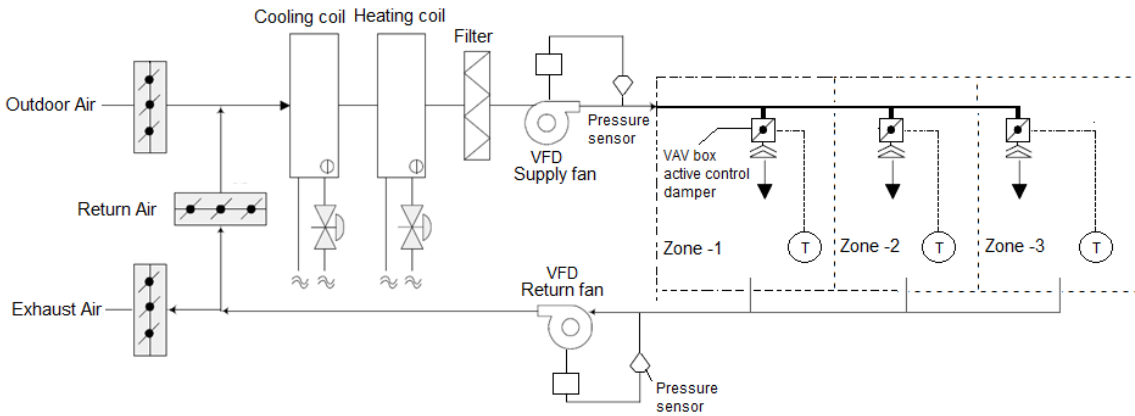
Where,

- V_o = outdoor airflow rate, cfm/person
- C_s = CO_2 concentration in the space, ppm
- C_o = CO_2 concentration in the outdoor air, ppm
- N = CO_2 generation rate, cfm/person

2. It compares the "desired" outside air cfm (setpoint) with the "actual" outside air cfm (controlled variable) and provides signal to the exhaust, return air and outdoor air dampers to modulate between the minimum position (base ventilation rates) and the maximum position (DVR).

Variable Air Volume (VAV) Systems

VAV systems vary the amount of air supplied to a given area, while maintaining the air at a constant temperature. These are applied to multi-zones and offer a very energy-efficient system. The simplest VAV systems use variable frequency drive (VFD) fan motor and VAV terminal boxes to adjust the supply airflow.



Conventional VAV system with VAV boxes

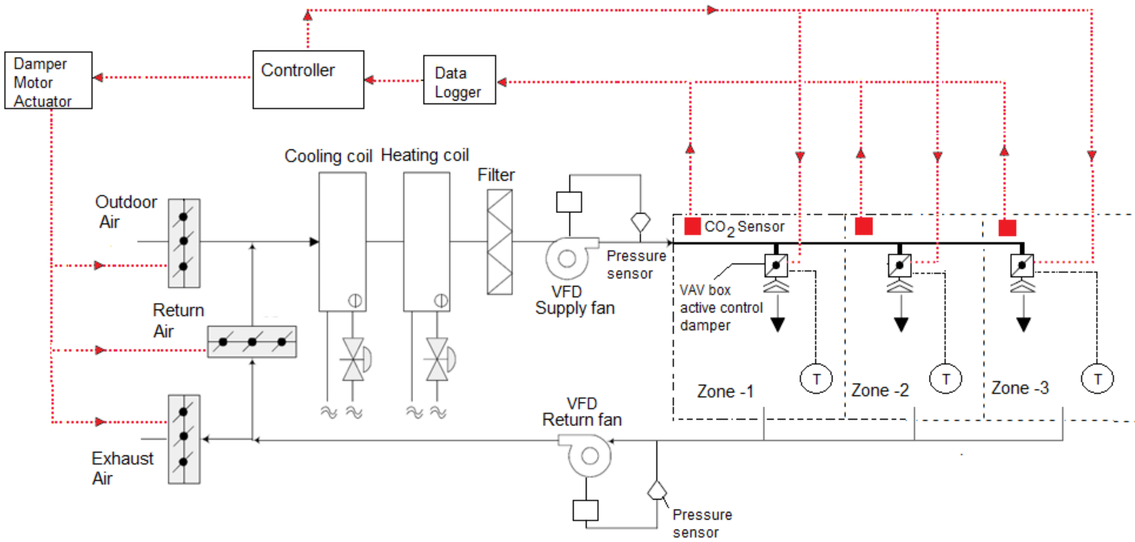
Control of the outdoor air damper in a variable air volume system is very similar to that of a constant volume system. However, setting the minimum damper position in a variable air volume system is significantly more complicated than in a constant volume system.

During operation, the VAV air-handling unit delivers a mixture of outdoor air and recirculated air to the multiple spaces it serves. The design ventilation rate (DVR) for a VAV system is the summation of ventilation requirements of all the zones served. There will be times when one zone is fully occupied and therefore calling for high ventilation rates while other zones may be unoccupied calling for minimum ventilation rate. The base ventilation flow adjustment of dampers at a central air handler alone thus won't serve a particular zone well. A VAV system will require adjustment to both the individual VAV box damper serving a particular zone as well as the main outdoor air damper at the main central air handling unit.

Without active ventilation control, the volume of conditioned air that enters the space is controlled by a space temperature sensor. The temperature sensor modulates the damper of the VAV box to lower limits during part load but also restricts the ventilation air to that zone.

With active ventilation control (DCV), the percentage or 'richness' of the ventilation air in the total supply air must be maintained. This is significantly more complex.

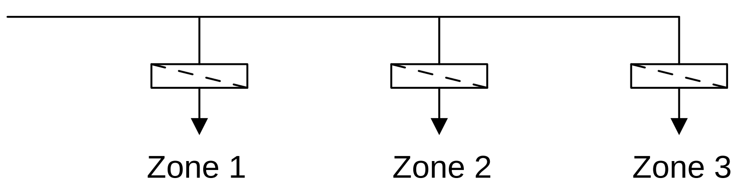
One approach to optimizing ventilation in a multiple-zone VAV system is to have one CO_2 sensor and one temperature sensor (or one combined sensor) in each zone or at least for the most critical zones that are densely occupied and experience widely varying patterns of occupancy (such as conference rooms, auditoriums, or a lounge area). The building automation system (BAS) then determines how much OA needed in each zone, and sets the building outdoor air to meet critical zone. Thus, less critical zones will always be little over ventilated, but no zone is under-ventilated. The concept will be understood with the example below.



Conventional VAV system with VAV boxes

To understand how multi-zone demand ventilation control with multiple CO_2 sensors works, let's consider an example.

Say, the total volume flow rate Vs and the outdoor air flow rate Vo at design conditions for three zones are as shown below. Based on these requirements, the percentage of outside air, OA, at design conditions is 35%.



Design	Z1 (cfm)	Z2 (cfm)	Z3 (cfm)	Total (cfm)
Supply air (Vs)	2,000	4,000	4,000	10,000
Outside air (Vo)	500	1,000	2,000	3,500
Percent outside air OA%	25%	25%	50%	35%

During part load conditions say the required supply air and outside air requirements are as indicated below:

Lean Period (Part Load)	Z1 (cfm)	Z2 (cfm)	Z3 (cfm)	Total (cfm)
Supply Air (V_s)	1,000	3,000	3,000	7,000
Outside air ($V_{O,REQUIRED}$)	200	600	800	1,600
OA%	20%	20%	27%	
Outside air ($V_{O,ACTUAL}$)	1,000 x .27 = 270	3,000 x .27 = 810	3,000 x .27 = 800	1,890

The building controller (automation system) will determine that the zone 3 requires a higher fraction of outdoor air than the other zones; thus zone 3 is the critical zone and the fraction outdoor air for the entire building is set at 0.27. Although the actual outdoor air supplied to zones 1 and 2 exceeds the minimum requirement, all three zones meet the outdoor air requirement. In addition, heating and cooling energy use is reduced because the quantity of outside air introduced into the building (1,890 cfm) is less than the quantity of outside air if the dampers were fixed at the design fraction outdoor air (0.35 x 7,000 cfm = 2,450 cfm).

What's difference between VAV and DCV?

In essence, VAV systems also operate as DCV systems; i.e. varying the amount of air, but there is still a difference.

DCV has been related to indoor air quality control, while VAV systems are related to thermal comfort control. Not all VAV systems operate as DCV systems. Only the VAV systems where the airflow rate varies according to thermal comfort control, as well as by the requirements of air quality, are considered as DCV systems.

ECONOMIZER CONTROL

In both CAV and VAV systems, if the air-handling unit is equipped with the "Economizer" to allow for free cooling, it will interact with the DCV control loop. Whichever control loop is calling for the highest amount of outdoor air will take priority. For example, if the economizer loop is driving the outdoor air dampers open to provide free cooling, the dampers should not close because zone CO_2 concentrations are below set point. In contrast, the outdoor air dampers should be driven toward the DVR, if any zone CO_2 concentration exceeds setpoint.

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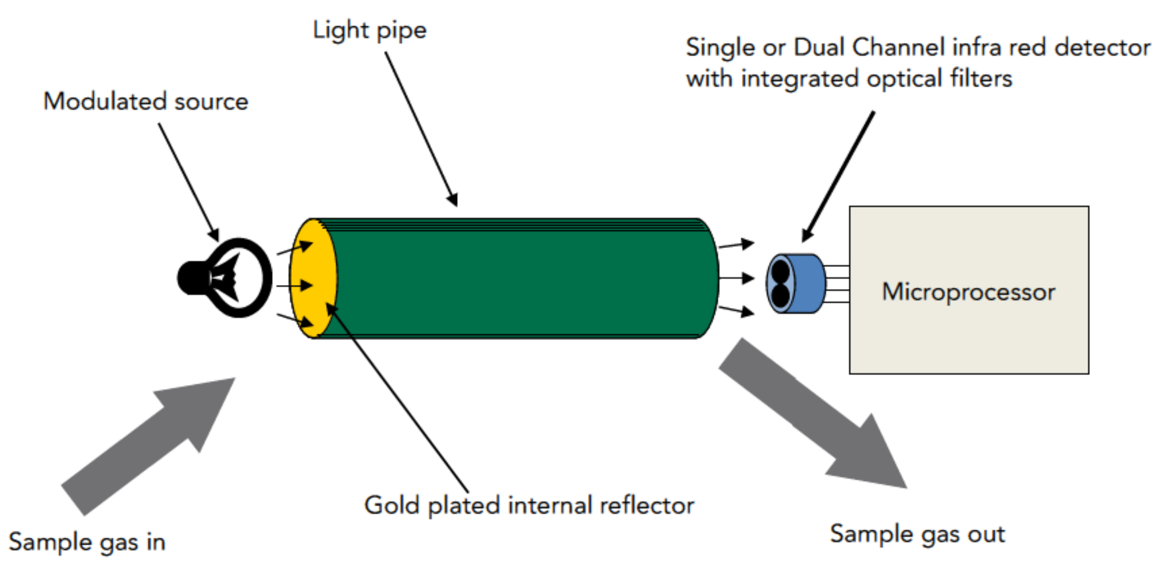
SECTION - 5: THE CO_2 SENSING TECHNOLOGY

A key component of DCV is the carbon dioxide sensor. CO_2 sensors in HVAC applications are based exclusively on the Infrared (IR) absorption principle. This is because different gases absorb infrared energy at specific and unique wavelengths in the infrared spectrum.

There are two types of sensors to measure CO_2 concentration with the help of the IR absorption method: "Non-dispersive infrared" and "Photo-acoustic". Both have distinctly different operational characteristics.

Non-Dispersive Infrared (NDIR) Detection

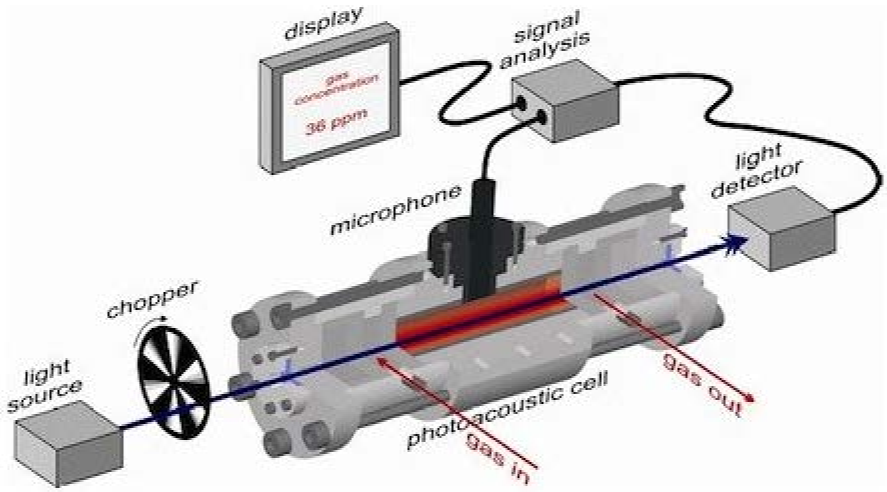
Sensors based on non-dispersive infrared (NDIR) detection search the net increase or decrease of light that occurs at the wavelength where CO_2 absorption takes place. The light intensity change depends on the concentration of carbon dioxide.



The main components of a commercial sensor are an infrared source (lamp), a gas cell/path, a wavelength selection device, and some optical components (lenses or more usually mirrors) to couple the radiation from the source through the gas cell to the detector. The sensor accuracy can be affected by the accumulation of particles in the sensor and the aging of the light source.

Photo-Acoustic (PA) CO_2 Sensors

Another type of infrared technology used to measure CO_2 is called photo-acoustic sensing. This technology also exposes the gas sample to infrared light. However, unlike absorptive infrared, the technique is based on the detection of sound waves that are generated due to the absorption of modulated light. The amplitude of the acoustic wave is directly proportional to the product of laser power, the concentration of the molecules in the gas sample, and the sensitivity of the PA detector.



The sensor accuracy can be affected by vibration and atmospheric pressure changes but these are non-sensitive to dirt and dust.

Sensors for HVAC Applications

Both non-dispersive infrared and photo-acoustic CO_2 sensor technologies are subject to drift (± 100 ppm / year) and inaccuracy (± 100 ppm). Both uses digital electronics, with output signals communicated on digital serial bus formats (e.g., BACNet) or else converted to standard analog output of 0-10 VDC or 4-20mA. This corresponds to 0-2000 ppm CO_2 concentration.

Non-dispersive infrared technology is preferred over photo-acoustic for two reasons:

1. Photo-acoustic sensors have increased error at low humidity (i.e. less than 25% RH).
2. Photo-acoustic sensors generate a relatively "noisy" output signal.

Typical Non-Dispersive Infrared CO_2 Sensor

Below are the generic CO_2 sensor specifications that are appropriate for the HVAC industry:

1. Range: 0 - 2,000 ppm
2. Accuracy: +/- 50 ppm
3. Temperature dependence of CO_2: < 5 ppm per °C
4. Stability: <5% Full Scale for 5 years
5. Linearity: +/- 2% Full Scale
6. Voltage: 230 VAC, electrically isolated power supply unit
7. Indicator Lights: LED green: up to 1000 ppm, LED amber: from 1000 ppm CO_2 to 1500 ppm CO_2, LED red: over 1500 ppm CO_2 (flashing)
8. Acoustic warning signal: Every 5 minutes above 1500 ppm CO_2.
9. Housing: Standardized concealed box
10. IP rating: IP 20 (shock hazard protection for indoors)
11. Safety class: II
12. Manufacturer recommended minimum calibration frequency: 5 years

How to evaluate CO_2 Sensor Location and Quantity?

Sensor location and quantity are not explicitly defined in ASHRAE or any other code. The exact criteria will vary between different buildings and system types. The key is to select a location where the sensor can accurately measure the CO_2 concentration and is representative of the area or zone served. A special consideration for CO_2 sensor placement is to ensure it is not located in an area where people might be directly breathing on the sensor (e.g., near water cooler/coffee service areas).

Sensor Coverage

According to California Title 24 Energy Code, if in a given zone the design occupancy density is greater than 25 people per 1,000 ft^2, the space would be considered a likely candidate for a DCV, and should receive its own sensor. If Title 24 is not applicable to the project, then you may consider using fewer sensors, and lowering the threshold set point to account for less CO_2 sampling, and increased dilution of air within the space.

Zones that are served by one air handler that are not loaded to the same level or frequency should have their own sensors, provided DCV shows opportunity for worthwhile ventilation airflow reduction. The table below provides general guidelines:

Building Arrangement	**CO_2 Sensors**	
	One HVAC Unit	**2 or More HVAC Units**
Single space, single zone Area < 5000 ft^2	One	One sensor per unit
Single space, single zone Area > 5000 ft^2	Quantity as required. The area covered by each sensor shall be less than 5000 ft^2	One sensor per unit (up to 5000 ft^2 each)
Multiple spaces, single zone Area < 5000 ft^2	One. Locate sensor in space that is most ventilation sensitive	One sensor per unit
Multiple spaces, single zone Area > 5000 ft^2	One sensor per space	One sensor per unit
Multiple spaces, multiple zones	One sensor per zone	One sensor per zone

Rule of Thumb: Generally one sensor can serve up to 5,000 sq. feet.

Sensor Location

1. **Duct Mounted Sensors -** Duct mounted sensors are typically located in the return airstream of an air-handling system. This approach is best applied where the ventilation system operates continuously and where all the zones served by the air handler have similar levels of activity and occupant densities, occurring at the same time. A duct mounted sensor is not recommended where the system serves a number of areas with diverse occupancy. A duct mounted sensor in the return air duct of a system that also incorporates a ceiling return plenum may be subject to error because of building infiltration or supply duct leakage.

2. **Wall Mounted Sensors (Zones Measurement):** The local wall mounted sensor(s) are recommended for multi-zone applications. Criteria for placement of wall-mount sensors are similar to those for temperature sensors. Because people breathing on the sensor can affect the reading, find a location where it is unlikely that people will be standing in close proximity (2 ft. to the sensor). Avoid installing in areas near doors, air intakes or exhausts or open windows. One sensor should be placed in each zone where occupancy is expected to vary. Sensors can be designed to operate with VAV based zones or to control larger areas up to 5,000 ft^2 (if an open space).

   Compared to duct mounted sensors, this method will be more expensive since additional sensors, wiring, and control points must be installed. In addition, the control sequences will require additional programming. This is the recommended control option for variable air volume (VAV) air handling systems.

Duct vs. Wall Mount

Generally, the wall mounted sensors shall be used for VAV installation and even preferred for CAV installation. Sensors in the occupied space are preferred over location in ductwork. This is because return air tends to be an average of all spaces being conditioned and may not be representative of what is actually happening in a particular space. This means that some spaces could be highly under ventilated, and others over ventilated.

The principal driver for use of duct-mounted sensors is to reduce costs by reducing the number of sensors required for a job. In the past few years, CO_2 sensor pricing has dropped dramatically meaning that the cost difference between using duct-mounted and multiple space-mounted sensors is a minimal portion of the job cost. Hence individual zone wise wall mounted CO_2 sensors are the better choice.

Sensor Wiring

Sensor wiring, voltage, power and control requirements are similar to those ones commonly used in thermostats. There are two types of sensors: wired and wireless. Data from wireless sensors is delivered with the use of signal communications. Wireless sensors have self-contained power supply. Such sensors are used on-board power controls to alert a building operator when battery charge is low and needs to be changed.

The sensors output can be interfaced to any Building Management System (BMS) for retrieving the sensor status via an isolated opto-coupled relay output, which indicates various stages of CO_2 concentration. These digital outputs are suitable for remote management of the HVAC dampers via the BMS systems

Sensor's Calibration

Most CO_2 sensors available in the market today are self-calibrating and require no maintenance over their rated life of 15 years. The self-calibrating feature used by these sensors is based on the fact that when buildings are unoccupied, inside concentrations of CO_2 will typically drop to outside levels which are typically around 400 ppm. The CO_2 sensor is programmed to look for these low points that might occur over a 3-week period. If the sensor sees that it is out of adjustment with the lowest concentration measured over three weeks, the sensor automatically adjusts its calibration. To ensure optimum operation of this self-calibration feature, it is highly recommended that the control sequence of the system include a periodic per occupancy purge of the space to ensure that the sensors see true outside/background levels.

Important!

When you are working towards green building design and for Green Certification, the LEED™ rating system is very specific about the location of sensors. It requires placing sensors between 3 and 6 feet above the finished floor in what is known as the "breathing zone." This is the space in a room where people inhale and exhale. Previous standards of practice put sensors in the return air duct, something unacceptable under LEED because this location does not sense actual room conditions that humans will experience.

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SECTION - 6: KEY DESIGN ISSUES AND CHALLENGES

Two important criteria for any CO_2 control strategy are that the target per-person ventilation rate is met at all times, and that during periods of changing occupancy the lag times as prescribed in ASHRAE standard 62.1-2004 are met. It is possible to determine the number of sensors and to select types of sensors, when a control strategy is chosen correctly. Here are few areas that need attention.

Effect of Variable Respiration Rates and Non-Steady Conditions

Implementing CO_2 based DCV is a matter of estimating the CO_2 generation rate of the occupants (*N*), measuring the concentration difference in the space versus outdoors (*Cs* – *Co*), and then using this difference to determine the rate at which ventilation air (*Vo*) on a per-person basis is delivered to the space. The mass balance equation for determining the target cfm/person assumes the steady state equilibrium condition. The steady state condition means that everyone in a building will be seated; is of the same size, sex, health; and is consuming the same diet. Can all the other factors that influence respiration rates be held constant? The answer is No.

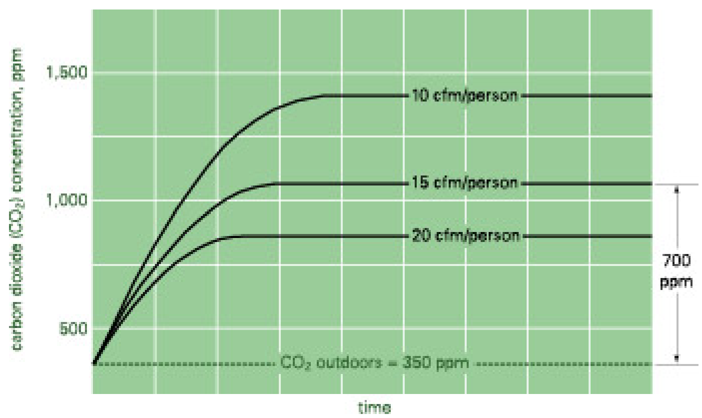
If the level of activity was more strenuous than that of typical office work, then the metabolic rate would go up with a corresponding increase in the CO_2 generation rate. For the same ventilation rate then, this increased level of activity would result in a higher build-up of CO_2 concentrations.

When we examine the range of activities and their impact on respiration, we find that the respiration rate can easily vary to 0.50 for walking, 0.60 for Light machine work and 0.90 for the upper threshold of "light activity". All of these are cited in the Appendix to ASHRAE Standard 62-1999. When these are used to calculate the amount of ventilation per person, we get with the following range of rates:

Respiration Rate	Actual Outside Air	Activity
N	CFM / person	
0.30	14.5	Seated
0.40	19.4	Office Work
0.50	24.2	Walking

0.60	29.0	Light Machine Work
0.90	43.5	Upper threshold of “light activity”

The figure below shows the typical pattern of buildup of CO_2 in a space with office type activity (1.2 MET). The chart assumes a steady-state condition where a constant occupancy is present and the ventilation rate is constant. Once people enter a room, CO_2 concentrations will begin to increase. These levels will continue to increase until the amount of CO_2 produced by the space occupants and the dilution air delivered to the space are in balance. This is called the equilibrium point.



Indoor CO_2 concentrations at various ventilation rates

Where each curve levels off, the rate of CO_2 generation (occupant activity) in the space balances the rate of CO_2 removal from the space. The amount of time required to reach the steady-state condition depends on the population density, the volume of the space, and the air circulation rate. It can be as short as a few minutes for a densely occupied space with a low ceiling height, or as long as several hours for a space with a high ceiling and few occupants.

Concerns over Insufficient Ventilation of Non-Human Pollutants

According to the ASHRAE Journal the single most important issue preventing greater use of DCV is the concern around non-human pollutants. During periods of low occupancy, DCV can reduce ventilation levels low enough that potential building contaminant concentration can increase to the point of causing occupants to complain. These contaminants can be created by off gassing from new furnishings or construction materials, or increased levels of air contaminants from cleaning materials, high particle or dust levels or other episodic occurrences such as spills of odorous liquids or volatile organic compounds (VOCs). ASHRAE has tried to address the issue of non-human pollutants by recommending a minimum area component of the outdoor air ventilation requirements that is typically 60 cfm per 1000 square feet. However, in many cases this airflow level may be insufficient to eliminate complaints, especially during periods of low occupancy. The user may have to exercise changing the upper and lower setpoints by trial and error based on real spot test measurements.

Intermittent Occupancies

For most space types, the design ventilation rate is calculated by multiplying the maximum occupancy of the space by the ventilation requirement (cfm/person). The intermittent occupancy provision of ASHRAE 62, Section 6.1.3.4 permits calculation of the design ventilation rate based on the *average* occupancy of the space, rather than the *maximum* occupancy, but only if the duration of maximum occupancy in that space does not exceed three hours. Using the intermittent occupancy provision instead of implementing DCV, sometimes simplifies system control and permits smaller HVAC equipment without sacrificing operating costs appreciably. When considering DCV, it is improper to use this provision to lower the maximum occupancy for the sake of reducing the design ventilation rate.

Demand-controlled ventilation should be sized based on the peak occupancy. It is NOT appropriate to reduce the size (capacity) of the ventilation system when demand controlled ventilation is being used.

Chemical Filtration

When considering CO_2 based DCV, make sure that carbon dioxide is NOT removed from the space by methods such as gas-sorption filtration. When CO_2 is used to indicate occupancy, any means of reducing its concentration (other than dilution with outdoor air) will result in an under ventilated space.

VOC sensors can be substituted for CO_2 sensors

Volatile organic compound (VOC) sensors cannot be simply substituted because VOC sensors:

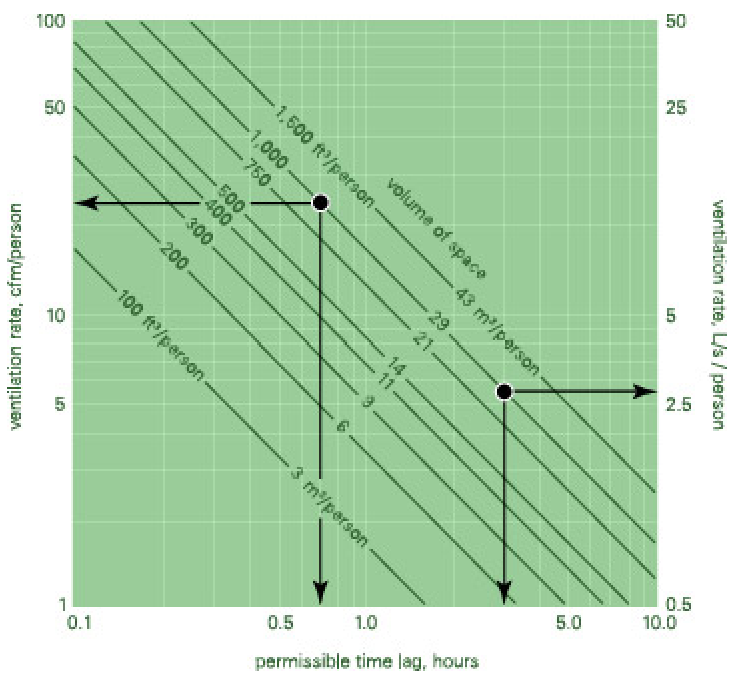
- Cannot measure CO_2;
- React in different ways to different contaminants; and

- Cannot distinguish between a potentially harmful air contaminant and harmless gas (such as perfume or after-shave vs. benzene).

Lag time for ventilation

During the non-steady-state conditions, which are typical of real buildings, the concentration of CO_2 within the space lags behind the actual number of occupants. At the beginning of occupancy the steady-state condition does not yet exist in the space, so the *measured* difference between indoor and outdoor CO_2 concentrations would result in an under-ventilated space. Considerable time can elapse before the space reaches its steady-state condition, if it ever does. (Most spaces never reach equilibrium because of changing occupancy and operation of the HVAC system.)

Depending on the application, ASHRAE Standard 62-2004 [Section 6.1.3.4] allows ventilation to lag occupancy, provided that the ventilation system achieves an acceptable indoor condition within the permissible time frame. The figure below shows the Maximum lag time permissible for ventilation**.**



DCV and Building Pressure

When the outside dew point exceeds 65°F, humidity levels in negatively pressurized building envelopes can exceed 70% RH. High humidity conditions in and near the building envelope will result in mold growth. Some molds may be toxic to humans while others may damage the building structure. The widespread use of DCV has limited the amount of outside air introduced into a building. Without a positive pressurization flow (the difference between the outside air intake flow rates and the total exhaust flow rates), a building cannot be pressurized. Designers must carefully consider building pressurization when utilizing demand controlled strategies (CO_2 or others). Building pressurization becomes even more critical if the energy recovery is used since the differential used to pressurize the building is significantly reduced, even at system design maximums.

Economizer Control

In buildings with an economizer cycle, allow the economizer to override the DCV system at times when the additional ventilation would provide “free” cooling. Select DCV systems that are able to increase outdoor air intake before the building opens in the morning to deal with concentrations of contaminants that may build up overnight.

Equally important, the economizer or DDC system should be properly programmed to accept the sensor's input. Improper programming can negate any potential benefits. For example, if the system is set up to open up the outside air full-open at the first sign of people, it will over-ventilate unless the group of people is normally a very large group.

Outside CO_2 Variability

We know the outside CO_2 is not static and that it varies both geographically and over time throughout a single day, as well as seasonally. But in most locations, the outdoor concentration (*Co*) of carbon dioxide seldom varies by more than 100 ppm from the nominal value. Because of this and in lieu of installing an outdoor CO_2 sensor, most designers use a one-time reading of the outdoor CO_2 concentration at the building site. This simplifies control, lowers the installed cost, and usually increases accuracy because it avoids the potential inaccuracy of an outdoor sensor.

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SECTION - 7: INVESTMENTS AND ENERGY SAVINGS

The overall cost for implementing DCV has dropped substantially in recent years. The average cost of CO_2 sensors is now priced below $200 (compared to over $500 a decade ago). Today's sensors can self-calibrate, so they need far less maintenance than their predecessors. Also, several HVAC equipment manufacturers now offer DCV-ready rooftop units and variable air volume (VAV) boxes. This equipment is shipped with terminals for the CO_2 sensor wires and controls that are preprogrammed to implement a DCV strategy. By limiting installation costs to the cost of mounting the sensor and running wires to the rooftop unit or VAV box (wireless models are available), DCV ready HVAC equipment substantially reduces the cost of implementing DCV.

APPLICATION OF DCV

Which spaces would benefit the most from DCV?

Although no hard and fast rules apply, DCV provides the greatest savings for buildings with:

1. Highly variable occupancy - DCV offers the greatest potential for energy savings in buildings with wide or unpredictable swings in occupancy, such as auditoriums, restaurants, bars, cafeterias, theaters, retail stores, classrooms, and conference rooms. Buildings with highly variable occupancy and buildings that rarely or never reach design occupancy will likely save more energy than facilities with predictable near-design occupancy, such as office buildings or schools.
2. Moderate to extreme heating or cooling climates - Given that DCV can reduce the amount of outdoor air brought in, buildings in climates where a lot of energy is required to heat or cool the outdoor air stand to gain the most, while those in climates where little conditioning is required and where economizer operation is common will save less. Facilities with large refrigeration loads, such as supermarkets, will also benefit from the reduced humidity load that the display cases would otherwise have to remove.
3. Conventional HVAC systems - Buildings that have mechanical air conditioning systems offer opportunities for greater energy savings than do facilities using other cooling systems, such as evaporative cooling. These other systems use

100 percent outside air during normal operation, which means that "ventilation performance" cannot be improved. However, these buildings may benefit from the use of DCV in winter because it will reduce the amount of outside air that must be heated.

4. Long operating hours - Buildings that are only open for a few hours per day are unlikely to be good candidates for DCV. Those facilities might be better off using timers to shut off ventilation fans during unoccupied hours.

Below are some guidelines on what type of spaces are most suitable for a DCV control strategy.

Recommended	**Possible**	**Not Recommended**
Auditoriums, Theaters Music rooms, Ball rooms Shopping malls, Supermarkets Lobbies and waiting areas Conference rooms Casino & Bowling alleys Churches School classrooms Platforms Commercial Laundries	Dining halls, cocktail lounges & cafeteria Training shops Smoking lounges Specialty shops (barber, florists, furniture, hardware etc) Patient rooms, recovery rooms	Locker rooms Repair and service stations Pet shops Manufacturing areas Warehouses Laboratories Operation rooms and ICU Swimming pools

Most applications indicated as "possible" may be suitable applications, but should be evaluated by the HVAC system designer. Separate factors may govern system selection, such as, mandatory ventilation requirements other than the ASHRAE Standard 62, pressurization between spaces (e.g., between kitchens and dining rooms), regular periodic release of building-related contaminants that are a health hazard to occupants, and extensive requirements for local exhaust.

Caution!

Not all buildings are good candidates for DCV.

1. DCV should be used only in areas where human activity is the main reason for ventilating the space. Industrial or laboratory spaces that are subject to indoor air quality (IAQ) degradation from a wide variety of sources are unsuitable for CO_2-based ventilation control.
2. CO_2-based DCV is suitable only when there is a means of automatically adjusting the ventilation air supply (e.g., variable-speed fans or some variable damper arrangement). If this control is not available, the savings from DCV may justify the modifications to accommodate this degree of control.
3. Thus CO_2-based DCV may not be appropriate, or may require higher target ventilation settings in new buildings or others where there are contaminants not related to human occupancy, as it may not provide sufficient fresh air to dilute those contaminants. The CO_2 sensors used for DCV are not appropriate to monitor CO_2 for medical or industrial purposes that demand precise air quality control.

SAVINGS

If demand controlled ventilation lowers excessive supply outdoor air in a building during heating and cooling seasons, then annual energy expenses for heating and cooling the outdoor air are reduced accordingly. In most locations, the outside air also brings humidity into the space, which burdens the cooling energy consumption substantially. There is significant incentive for building operators to reduce the amount of outside air entering a space when the HVAC system is mechanically cooled or heated.

How much money will a DCV system save me?

Actual occupancy levels in buildings are generally significantly lower than the design occupancy levels. The experience indicates that actual occupancy levels may be 60 to 75% lower in some buildings than the design levels. The saving energy potential using DCV may vary depending on climate, type of a building, hours of use, type of HVAC system, occupancy in the space in which it is implemented, and other operating conditions. The capability of authorized staff to maintain and operate equipment properly may also positively affect savings.

Broadly, the money can be saved on two accounts. First, it saves energy by avoiding heating, cooling and dehumidification of more ventilation air than is needed. Second, lower outdoor air requirements decrease the fan energy expenses to supply or extract air from a building.

Energy Saving Potential

Thermal Energy

When the ventilation is reduced, there is proportionate reduction in the cooling and heating requirements. Refer to the equations below:

The air-conditioning load required for cooling or heating the outside air may be calculated from the following equation:

$q_T = m\ (h_E – h_L)$

$q_s = m\ cp\ (T_E – T_L)$

At standard conditions, where the density of air is 0.075 lb/ft^3 and the specific heat is 0.240 Btu/lb-°F, this equation can be simplified as follows:

$q_T = 4.5\ (Q)\ (h_E – h_L)$ Btu/hr

$q_s = 1.08\ (Q)\ (T_E – T_L)$ Btu/h

Where,

- q_T = Total load (latent + sensible), Btu/hr. [Note the total load accounts for both cooling as well the dehumidification, i.e. moisture removal during humid months].
- q_s = Sensible load, Btu/hr. [Note the sensible load is only the cooling load].
- c_p = specific heat of air at standard conditions, 0.24 Btu/lb-°F
- m = mass flow rate of outside air, lbs/hr
- Q = Airflow, cfm
- h_E = Enthalpy of air entering the air conditioning equipment, Btu/hr-lb
- h_L = Enthalpy of air leaving the air conditioning equipment, Btu/hr-lb

- T_E = Temperature of air entering the air conditioning equipment, °F
- T_L = Temperature of air leaving the air conditioning equipment, °F

Clearly the sensible load and the total load are dependent on the airflow rate and the entering and leaving conditions of air. Let's review the thermal savings using the following example:

Example:

Consider a 10000 sq.-ft. office located in Southern California with the following specifications:

1. 100 occupants
2. Summer design conditions:
   - Outdoor: 89°F dry bulb, 70°F wet bulb
   - Indoor: 75°F and 50% relative humidity
3. Heat gain (not including outside air):
   - Total gain = 56,000 Btu/hr
   - Sensible gain = 40,000 Btu/hr
4. Winter outdoor design conditions:
   - Outdoor: 40°F
   - Indoor: 72°F and 50% relative humidity
5. Heat loss (not including outside air): 45,000 Btu/hr

Assume an outside airflow requirement per ASHRAE Standard 62. In this example it is 15 cfm per person, so the total required fresh air is 1500 cfm.

Solution:

Total heat load: $q_T = 4.5\ (Q)\ (h_E - h_L)$ Btu/hr

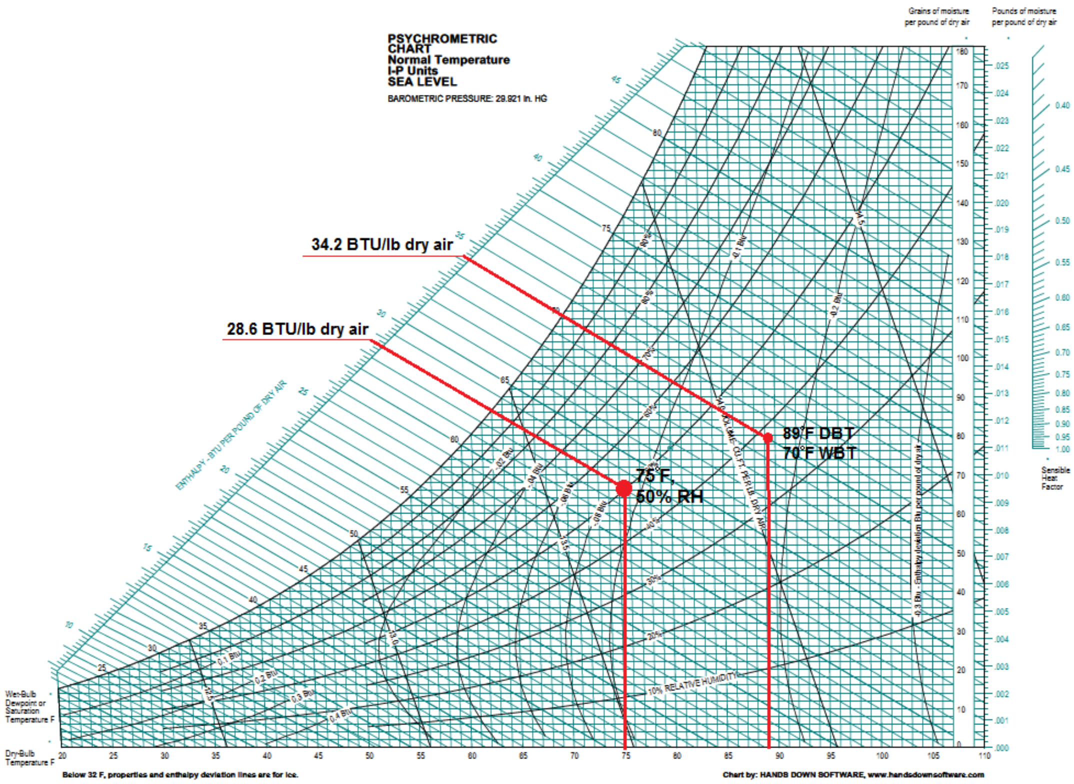
The enthalpy of entering and leaving air is determined from the "Psychrometric Chart"

Enthalpy of outdoor entering air: h_E = 28.6 Btu/lb

Enthalpy of air leaving the air conditioning equipment: h_L = 34.2 Btu/lb

Therefore, the total heat gain from the outside air is:

q_T = 4.5 (Q) (h_E – h_L) = 4.5 (1500) (34.2 – 28.6) = 37800 Btu/hr



Similarly, the sensible load is:

qs = 1.08 (1500) (89 - 75) = 22680 Btu/hr

And the winter heat loss is:

qs = 1.08 (1500**)** (72 - 40) = 51840 Btu/hr

Now say with DCV, the outdoor air is reduced by 40% to 900 CFM, the new cooling and heating requirements shall be:

q_T (summer) = 4.5 (900) (34.2 – 28.6) = 22680 Btu/hr

q_s (summer) = 1.08 (900) (89 - 75) = 13608 Btu/hr

q_s (winter) = 1.08 (900**)** (72 - 40) = 31104 Btu/hr

Energy Savings

Months	Heat load at Design Ventilation Rates, 1500 cfm (DVR)	Heat load at reduced ventilation rate of 900 cfm using controlled ventilation (DCV)	Energy Savings (DVR – DCV)
Summer total load	37800 Btu/hr	22680 Btu/hr	15120 Btu/hr (~4.43 kWh)
Summer sensible load	22680 Btu/hr	13608 Btu/hr	9072 Btu/hr (~ 2.66 kWh)
Winter heat loss	51840 Btu/hr	31104 Btu/hr	20736 Btu/hr (~6.07 kWh)

For 2 hours per day lean operation, 300 working days, the energy saving will amount to:

Summer total load = 4.43 kWh * 2 hrs * 300 working days = 2,658 kWh per annum

Summer sensible load = 2.66 kWh * 2 hrs * 300 working days = 1,596 kWh per annum

Winter heat loss = 6.07 kWh * 2 hrs * 300 working days = 3642 kWh per annum

The energy budget for heating is significantly higher. The dollar savings can be significant in large air-conditioned spaces.

Savings in Mechanical energy

In any given ventilation system, the energy consumption varies in accordance with fan laws, which state that the fan energy varies directly to the cube rate of the air flow rate.

Doubling the supply air volume requires the energy requirement of the ventilation fans to increase by a factor of eight: $(2)^3 = 8$. Conversely, if the demand of air is halved, the required mechanical energy delivered by the fans would be *reduced* by a corresponding factor of 8. In our example above, a 40 % reduction of ventilated air would lower the mechanical energy demand to $(0.6)^3 = 0.216$ or 21.6 % of the original, thus saving 78.4 % of the fan energy.

Each potential application for DCV must be considered individually, so that the many variables which might affect energy savings in a specific application are weighed appropriately. The real energy savings will vary considerably, obviously affected by many factors such as:

- Building Type - occupancy schedule
- Building Location - heating or cold region
- Space heating and cooling loads
- Ambient temperatures and humidity
- HVAC system type
- Amount of time the system economizes

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SECTION - 8: CODES & STANDARDS

Many building codes in the United States base their ventilation requirements in accordance with ASHRAE 62, which requires that a building brings in a specified minimum amount of fresh air to ensure adequate indoor air quality (IAQ). To adhere to this standard, the choice made in most buildings is to ventilate at the fixed minimum rate per person based on the building type and the assumed occupancy, usually the building's design occupancy. But because the number of people actually occupying the space at any given time can vary widely, the ASHRAE standard allows CO_2 based demand control ventilation with certain rules.

1. ASHRAE 62 – DCV GROUND RULES

DCV as a ventilation control strategy was clarified in 1997 in interpretation of IC 62-1999-33 (formerly IC 62-1989-27). This interpretation identified the ground rules for using CO_2 as a method of controlling ventilation based on real-time occupancy within a space as follows:

I. The use of CO_2 is applied using the Ventilation Rate Procedure of Standard 62, which establishes specific cfm/person ventilation rates for most applications. By definition, ASHRAE Standard 62 says that acceptable indoor air quality is achieved by providing ventilation air of the specified quality and quantity to the space. The standard states: "The Ventilation Rate Procedure described in Table 6.1 (referred in Section 2 of the course) is deemed to provide acceptable indoor air quality, ipso facto."

II. The CO_2 control strategy can be used to modulate ventilation below the design ventilation rate while still maintaining Table 6.1 ventilation rates. Sensor location and selection of the control algorithm should be based on achieving the ventilation rates. The control strategy should also be developed considering inside/outside CO_2 differential.

III. The control strategy must provide adequate lag time response as required in the Standard.

IV. If CO_2 control is used, the design ventilation rate may not be reduced to consider peak occupancies of less than 3 hours (often called diversity). In other words, the variable provision of 6.1.3.4 cannot be applied to lower the estimated maximum occupancy for the purpose of reducing the design ventilation rate while using DCV.

V. CO_2 filtration or bio-effluents removal methods other than dilution should NOT be implemented in the space.

VI. A base ventilation rate should be provided during occupied periods to control for non-occupant related sources.

The ASHRAE 90.1 - 2007 Energy Standard (section 6.4.3.8) also requires that spaces with a design occupancy density greater than 100 people per 1000 ft^2 (i.e.: lecture halls, auditoriums, lobbies) incorporate DCV in the HVAC design.

2. Local Building Codes

There are three major regional building code bodies in the US that establish model code that can be adopted by state, local city, and municipal jurisdictions.

- BOCA – (Building Officials & Code Administrators International) – Northeast
- SBCCI – (Southern Building Code Congress International) – Midwest/South
- ICBO – (International Code Conference of Building Officials) – Western States

Recently these three model code bodies have jointly adopted the International Mechanical Code (IMC) which establishes minimum regulations for mechanical systems using prescriptive and performance related provisions. Like the ASHRAE 62 standard, the IMC also provides provisions for modulation of outside air based on occupancy as long at target cfm per person ventilation rates are maintained. This is addressed in section 403.3.1 of the 2000 International Mechanical Code that states:

"The minimum flow rate of outdoor air that the ventilation system must be capable of supplying during its operation shall be permitted to be based on the rate per person indicated in Table 403.3 and the actual number of occupants present. The IMC has also created a commentary document to provide clarification to the intent of the code. In reference to section 403.3.1, the commentary uses CO_2 control as an example of a ventilation system that can provide a specific “rate per person” based on the actual number of people present. An excerpt from the commentary is provided below.

“The intent of this section is to allow the rate of ventilation to modulate in proportion to the number of occupants. This can result in significant energy savings. Current technology can permit the design of ventilation systems that are capable of detecting the occupant load of the space and automatically adjusting the ventilation rate accordingly. For example, carbon dioxide (CO_2) detectors can be used to sense the level of CO_2 concentrations, which are indicative of the number of occupants. People emit predictable quantities of CO_2 for any given activity, and this knowledge can be used to estimate the occupant load in a space.”

3. CALIFORNIA TITLE 24 COMPLIANT CO_2 DCV

The 2009 Code of Regulations, Title XXIV Part 6 Energy Code (Title 24), 2005 version has many similarities to ASHARE 62- 2007 with respect to the application of DCV strategies. Section 121 in this energy code explains the requirements of minimum ventilation air and the application of DCV. Some important differences are:

- DCV is required in single zone HVAC spaces that have an economizer and serve a space with a design occupant density, or a maximum occupant load factor for egress purposes in the CBC, of 25 people per 1,000 ft^2 or greater (with a few exceptions)
- Title 24 requires that the ventilation rates must be maintained between a minimum value based on floor area and a maximum value based on occupancy and a ventilation rate of 15 cfm/person. California’s ventilation rates consider human bio-effluents to be the primary contaminant of concern, hence the 15 cfm/person requirement. The ventilation rate of 15 cfm/person would correspond to a CO_2 rise of 700 ppm between indoor and outdoor CO_2 levels. However, Title 24 assumes there will be sensor error, so the indoor CO_2 setpoint is 600 ppm above outdoor ambient. If outdoor CO_2 is not measured, then the outdoor CO_2 level is assumed to be equal to 400 ppm. Without sensor

or assumption error, this setpoint will result in a ventilation rate of 18 CFM/person.

An important requirement to be aware of in Title 24 is that when the HVAC system is operating during normal occupied hours, the ventilation rate (while DCV is active) is not allowed to drop below the values listed in Table 121-A of Title 24, multiplied by the floor area of the conditioned space. The ventilation rate found in this table for a typical office building is 0.15 cfm / ft^2, which results in 15,000 cfm of minimum outside air for the typical 100,000 ft^2 office building presented in the AHSRAE 62 discussion.

Therefore, as a side by side comparison, the following minimum ventilation rates apply:

Comparison of Ventilation Requirements

Code/ Standard	DCV Applicable Building Area	Building Type	Lower Min OA Airflow	Upper Min OA Airflow	Percent OA Reduction
ASHARE 62-2004	100,000 ft^2	Office	6,000 cfm	8,500 cfm	29%
Title 24-2005	100,000 ft^2	Office	15,000 cfm	15,000 cfm[1]	0%
ASHARE 62-2004	100,000 ft^2	K-12 School	12,000 cfm	47,000 cfm	74%
Title 24-2005	100,000 ft^2	K-12 School	15,000 cfm	67,500 cfm	78%

An important concept to note from looking at the table above is that when DCV is applied to a typical office building, it does not offer a large reduction in outside air during times of low occupancy. However, DCV can offer a large reduction in minimum ventilation air to spaces that are designed to be more densely populated such as schools and auditoriums. In fact, the ASHRAE 90.1 -2004 Energy Standard (Section 6.4.3.8) requires that spaces with design occupancy density greater than 100 people per 1000 ft^2 (i.e.: lecture halls, auditoriums, lobbies, etc.) incorporate DCV in the HVAC design.

4. **LEED 2.2 Requirements -** The United States Green Building Council (USGBC) created the Leadership in Energy and Environmental Design (LEED®) program to create a consistent way of allowing owners and designers to design and build an environmentally responsive facility. Within this program are credits that directly discuss CO_2 sensor use and designing an HVAC system that is responsive to indoor carbon dioxide concentrations.

LEED Credits

LEED rewards CO_2 monitoring in two key credits. The principal credit is IEQ Credit 1 – Outdoor Air Delivery Monitoring. The intent of this credit is “to provide capacity for ventilation system monitoring to help sustain occupant comfort and well-being.” This credit has two components within its requirements.

1. First, it requires that all delivery systems have a direct means of outdoor airflow measurement. This is typically accomplished with airflow measuring stations located in air-handling units.
2. Second, it requires room monitoring of carbon dioxide concentrations in all “densely occupied” spaces, so here is where the sensors come in.

LEED Indoor Environmental Air Quality (IEQ) Credit 1 states that when the indoor CO_2 levels rise 10% above the ASHRAE 62- 2007 requirements, then the mechanical control system shall be able to send an alarm to the occupants so that they will be informed and can take corrective action. The spaces that should be included in the application of this credit are all densely populated areas such as those with an occupancy level greater than 25 people per 1,000 square feet (or one person per 40 square feet). This means spaces like classrooms, conference rooms, restaurants, auditoriums, courtrooms, gymnasiums and other assembly areas are usually considered “densely occupied.” This requirement applies regardless of the size of the room; a small conference room and a large lecture hall have the same basic requirements. In these spaces, the credit requires carbon-dioxide monitoring.

LEED is very specific about the location of sensors, requiring them to be between 3 and 6 feet above the finished floor in what is known as the “breathing zone.” This is the space in a room where people inhale and exhale. Previous standards of practice put sensors in the return air duct, which is unacceptable under LEED because this location does not sense actual room conditions that humans will experience.

Although IEQ Credit 1 specifies the installed height of sensors, it does not require the full implementation of a demand control system. Once sensors are installed; however, installing a demand control scheme is usually an easy option, and may potentially be rewarded with additional points under EA (Energy and Atmosphere) Credit 1 – Optimize Energy Performance.

Summary

Most heating, ventilation and air conditioning systems (HVAC) re-circulate a significant portion of the indoor air and admit fixed amount of fresh outdoor air to keep pollutant concentrations below acceptable threshold limits. The amount of fixed air needed for "proper" ventilation largely depends on the population of the building.

Since buildings are rarely at maximum occupancy, significant energy savings can be achieved by reducing the amount of outside air being introduced into the building and then conditioned without compromising air quality. Demand controlled ventilation (DCV) is a method that controls ventilation rates based on the concentration of carbon dioxide (CO_2) of interior air while maintaining proper indoor air quality.

Note that the CO_2-based DCV does not affect the design ventilation capacity; it just controls the operation of the system to be more in tuned with how a building actually operates. It is more of a system control effort as opposed to a new technology development. Earlier, the barrier to widespread implementation was not having a cost effective, simple and reliable sensor. In recent years, advances in sensor technology have shown that demand-controlled ventilation is now both feasible and cost-effective.

The payback from CO_2-based DCV will be greatest in higher density spaces that are subject to variable or intermittent occupancy that would have normally used a fixed ventilation strategy (e.g., theaters, schools, retail establishments, meeting and conference areas). The real energy savings with DCV will depend on the climate being "severe" enough, and the required ventilation rate being large enough, so that the cooling load reduction saves enough energy costs to offset the first cost of the CO_2-sensing equipment. It is important to look into each area individually before incorporating the DCV strategy. Carefully study all of the requirements in ASHRAE IC 62-2001-34 before you adopt a CO_2-based DCV control strategy.

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