# Integrated Systems Demand Control Technology

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## ABSTRACT

The peak electrical demand of office building VAV systems will be reduced by ≈1.2 Kw/1000 $Ft^2$ by employing an Integrated Systems Demand Control Technology (ISDCT) sequence to reduce peak intake flow by ≈56%. Supply, return, and exhaust fan energy decreases with reduced airflows and pressures; and chiller system energy is saved by reduced cooling coil loads.

The ISDCT sequence continuously computes zone contaminant concentrations allowing compliance with reference standards. The office computer system identifies active personal computers and transfer this and manually inputted zone occupancy data to the building automation system—eliminating the need for $CO_2$ sensors.

All buildings require dynamic building pressurization control to adjust for substantial wind and stack boundary conditions—which can actually reverse intake and exhaust flows. Building differential pressure measurement should be used to calibrate and dynamically reset the building pressurization flow setpoint.

Pressurization flow accuracy was evaluated using 2% and 6% repeatable airflow sensors; the 2% sensors allowed 10 steps of control near minimum pressurization versus 3 steps for the 6% sensor. Given a minimum controllable building pressure setpoint of ≈0.0005 inches water column, a sensor repeatability of 0.00005 inches water column will provide 10 steps of control.

## KEY WORDS

Demand Limiting, Building Pressurization, Indoor Air Quality, Energy Savings

## BUILDING PRESSURIZATION

Since Integrated Systems Demand Control Technology (ISDCT) varies ambient exchange rates, it is part of the building pressure control loop. The primary objective of building pressurization control is to prevent damage to the building envelope in the summer through infiltration and in the winter through exfiltration--both of which cause high humidity in exterior walls. Secondary objectives are to minimize energy consumption by minimizing pressurization flow and to improve air quality by minimizing infiltration of unconditioned air.

Building pressurization is determined by the difference between total intake and total exhaust flows of all air handling units and exhaust fans. Large variations in intake and exhaust flows due to wind and stack effect was documented in constant speed exhaust fans and air handling unit intakes (with fixed damper positions) and exhausts as long as thirty five years ago and has been the central issue of later studies[2,4,5,7,8]. Flow variations can be so large that exhaust and intake flows can be reversed[4,7,8].

Refer to Figure 1. During the minimum outdoor air cycle, air handling unit intake and recirculation dampers are modulated in sequence to maintain intake flow setpoint, and exhaust fans are controlled to maintain building pressure. During the economizer cycle, air handling unit exhaust is controlled along with exhaust fan speed to maintain building pressurization. The differential pressure sensor across the exhaust damper is used during switchover between minimum intake and economizer cycle—to stabilize wind and stack effects.

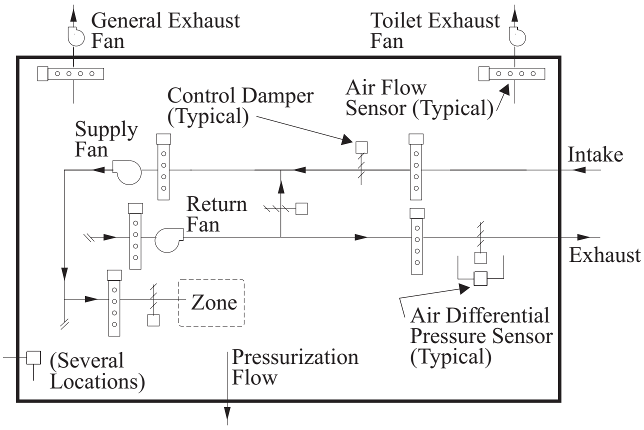


Figure 1: Building Pressurization Control

Supply and return airflow sensors, which are traditionally used to control building pressurization during the economizer cycle, cannot control building pressurization during the minimum intake cycle due to excessive errors. Table 1 lists intake, exhaust, supply, return, and pressurization flows for a 400,000 ft$^2$ tight building. Table 2 lists the pressurization flow measurement errors associated with 2% and 6% repeatability using: (1) measurement of all air handling unit intake flows and all exhaust fan flows; and (2) measurement of all exhaust fan flows and all air handling unit supply, return, and intake flows. The supply/return flow technique has excessive errors (>42% of pressurization flow with 2% repeatability sensors).

TABLE 1
Building Pressurization Flow

| **Building Pressurization Flow** | | | | |
|---|---|---|---|---|
| 400,000 Ft$^2$ x 0.03 Cfm/Ft$^2$ = 12,000 Pressurization Cfm | | | | |
| **Peak Cooling Operation During Minimum Outdoor Intake Cycle** | | | | |
| | Supply x1000 Cfm | Return x1000 Cfm | Intake x1000 Cfm | Exhaust x1000 Cfm |
| Air Handling Unit #1 | 100 | 77 | 23 | 0 |
| Air Handling Unit #2 | 100 | 85 | 15 | 0 |
| Air Handling Unit #3 | 100 | 72 | 28 | 0 |
| Air Handling Unit #4 | 100 | 77 | 23 | 0 |
| Exhaust Fan #1 | | | | 18 |
| Exhaust Fan #2 | | | | 18 |
| Exhaust Fan #3 | | | | 18 |
| Exhaust Fan #4 | | | | 11 |
| Exhaust Fan #5 | | | | 12 |
| Totals | 400 | 311 | 89 | 77 |
| Press. Flow x 1000 cfm (Total Intake - Total Exhaust) = | | | | 12 |

With 2% repeatability, pressurization flow can be measured within ≈10% of its true value. With 6% repeatability pressurization flow can be measured within ≈29% of its true value. With 29% pressurization flow error, ≈3 steps of control will change building pressure across null pressure, versus ≈10 steps with 10% pressurization flow error.

Given the difference in actual measured leakage in a variety of commercial buildings with and without fans, pressurization flows in commercial buildings will vary by ≈400%[6,9] (≈0.03-0.012 cfm/ft$^2$) Since building differential pressure setpoints are very small when compared to effects of wind pressures, building differential pressure cannot be used as the primary input control variable. Rather, building differential pressure should be used to reset the pressurization flow setpoint based on temperature and wind. The minimum pressurization/depressurization flow is determined when there is no wind and the ambient temperature is the same as the building interior.

Variable speed exhaust fans can control flow to within**Error! Not a valid link.**0.5% accuracy. Air handling unit intake and recirculation dampers can control intake flow within ≈2% of setpoint flow—if they are sized with the range of flow coefficients ($C_O$) required to accommodate wind and stack effects at the intake louver. Air handling unit exhaust flow can be controlled ≈1-2% accuracy by modulating return fan speed.

TABLE 2
Building Pressurization Measurement Errors

| **Building Pressurization Control During Cooling Peak (Minimum Outdoor Intake Cycle)** | | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| | **6% Repeatability** | | | | **2% Repeatability** | | | |
| | Supply x 1000 Cfm | Return x 1000 Cfm | Intake x 1000 Cfm | Exhaust x 1000 Cfm | Supply x 1000 Cfm | Return x 1000 Cfm | Intake x 1000 Cfm | Exhaust x 1000 Cfm |
| Unit #1 | 6.0 | 4.6 | 1.4 | 0 | 2.0 | 1.5 | 0.46 | 0 |
| Unit #2 | 6.0 | 5.1 | 0.9 | 0 | 2.0 | 1.7 | 0.3 | 0 |
| Unit #3 | 6.0 | 4.3 | 1.7 | 0 | 2.0 | 1.4 | 0.56 | 0 |
| Unit #4 | 6.0 | 4.6 | 1.4 | 0 | 2.0 | 1.5 | 0.46 | 0 |
| Exh. Fan #1 | | | | 1.1 | | | | 0.4 |
| Exh. Fan #2 | | | | 1.1 | | | | 0.4 |
| Exh. Fan #3 | | | | 1.1 | | | | 0.4 |
| Exh. Fan #4 | | | | 0.7 | | | | 0.2 |
| Exh. Fan #5 | | | | 0.7 | | | | 0.2 |
| Totals | 24.0 | 18.7 | 5.3 | 4.6 | 8.0 | 6.2 | 1.8 | 1.5 |
| | | | | | | | | |
| Methods: Measured Flows | | Square Root of: [Sum of Individual Errors Squared] | | Cfm Error % of Press. Cfm | | Square Root of: [Sum of Individual Errors Squared] | | Cfm Error % of Press. Cfm |
| Sup/Ret/Exh. Fan | | 15.5 | | 128.0% | | 5.2 | | 42.7% |
| Intake/Exh. | | 3.4 | | 28.7% | | 1.1 | | 9.6% |

## AIRFLOW AND AIR DIFFERENTIAL PRESSURE SENSORS

Airflow sensing technology has significantly advanced in the last decade. Permanently calibrated electronic thermal dispersion airflow sensors, traceable to a national standard, are commercially available with 2% installed repeatability over a full range of velocities[3]. These sensors are used for intake, exhaust, supply, return, and zone flow measurement and for building differential pressure. Permanent calibration eliminates expensive and intrusive recalibration required to protect the building envelope.

It makes little sense to employ an air balance contractor to take pitot tube traverses of system flows given the low cost of accurate permanently calibrated instrumentation. Traditional zone flow sensors (VAV boxes) have significant inaccuracies—as much as 30% at lower flows. Typically, zone flow sensors provide a feedback signal representative of the direction of flow change (increase/decrease) which is used only for the purpose of maintaining zone temperature.

A typical building pressurization after a successful air balance is in the range of 0.0015-0.0350 inches water column. This setpoint can be lowered to ≈0.0005 inches water column by using sensors[3] with repeatability equal to 10% of this minimal setpoint. Heating and cooling energy savings for pressurization flow is ≈$1-$2.5/Cfm-Yr.

## INDOOR AIR QUALITY

For the purposes of this paper, the building indoor air quality setpoint is determined by the design setpoint limit concentrations of particulate and gaseous contaminants. This definition of indoor air quality is similar to the internationally used *decipol* index in that it is directly tied to contaminant concentration. This definition is different than the *decipol* index in that it is not limited to only those contaminants that people can detect. ISDCT can be used with *ASHRAE Standard 62-2001 Ventilation Rate* and *Indoor Air Quality Procedures* and international decipol-based Standards including *CR 1752 Ventilation for buildings- Design criteria for the indoor environment* since all indoor air quality standards are tied to contaminant concentration.

Programs for zone contaminant concentration incorporating the following parameters have been commercially available since 1988 (CONTAM87)[1]: (a) volume/mass of air in space; (b) ambient intake flow rates; (c) zone flow rates; (d) supply flow rates; (e) contaminant concentrations of ambient air; (f) emission rates of building, people, cleaning, and other sources; (g) transfer fan flow (conference rooms); (h) zone ventilation efficiency; and other data.

The building automation system is integrated with the office computer system. A software patch provides the building automation computer zone occupancies and scheduled emissions periods (cleaning and other). The office computer system determines zone occupancy based on status or activity of personal computers and through manual input of conference and assembly room schedules. Manual input to the automation system establishes base emission rates for building carpet, furnishings, cleaning, construction activity, and abnormal ambient contaminant concentration. Gas phase filtration can be factored in by the building automation system along the lines Yu and Raber presented in 1991[10].

## SEQUENCE OF CONTROL

Refer to Figure 2. Based on the demand controlled ventilation modeling by Carpenter[1] intake flow rates can be decreased to near zero for approximately three hours, which is on par with the duration of an expected peak electrical demand period. During Period 1 ambient air intake is increased to dilute contaminant concentrations below the building indoor air quality setpoint limits—to a point that will minimize total energy cost (peak demand reduction and energy consumption). Historical trend data will establish the expected duration of future peak periods. Intake, supply, and zone airflow sensors will establish zone entering contaminant concentrations. During Period 2 intake is flow is reduced to a setpoint that will allow the contaminant concentrations to coast up to their setpoint limit at the end of the peak period.

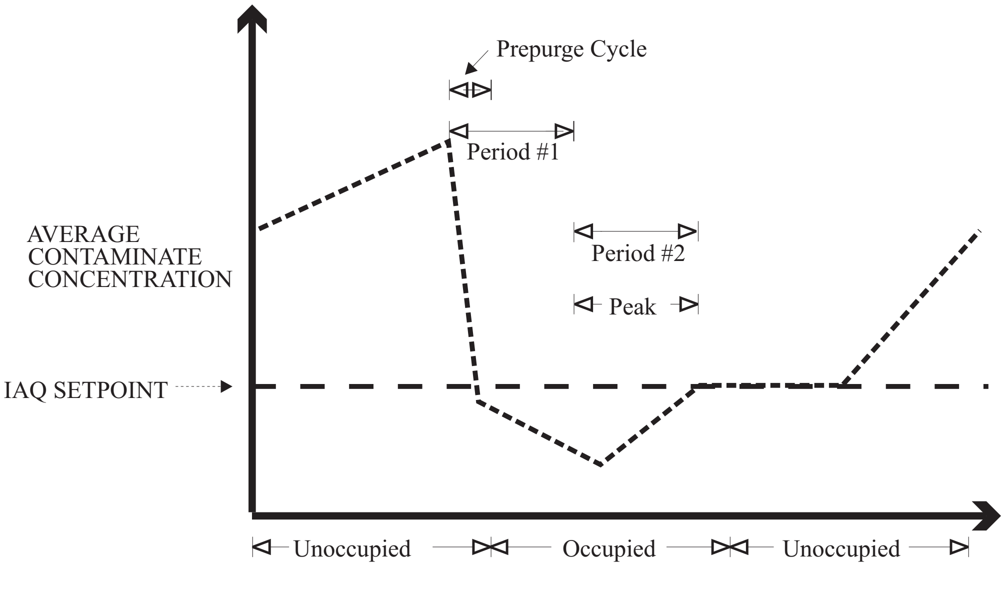


Figure 2: Daily Contaminant Concentration

## ENERGY SAVINGS

Tables 3 and 4 show the energy savings associated with reducing the ambient intake airflow by 57% in the sample building. ISDCT operation reduces supply airflow requirements by decreasing the cooling coil leaving air temperature setpoint by 2°F and by raising the space temperature setpoint by 1°F. Since the load on the cooling coil is substantially decreased, the leaving air temperature can be lowered. Since lower leaving coil temperatures will lower space humidity, space temperature may be slightly increased.

Supply flow is reduced by the ratio of the space air temperature differential (20/24), a 17% flow reduction. Supply flow is further reduced another 2% because supply and return fan horsepower (cooling load) decreases with decreased flow. Total peak electrical energy savings for the chiller system and fan system are ≈1.2 Kw/1000 $ft^2$.

TABLE 3
Peak Air Handling Unit and Exhaust Fan Savings

| | | x 1000 cfm | Inwc | Eff. | Kw | Total Kw | Total Kw Savings | Total Kw savings/1000 $ft^2$ |
|---|---|---|---|---|---|---|---|---|
| Normal Peak | Supply Fan | 400 | 4.0 | 0.60 | 420 | | | |
| | Return Fan | 311 | 1.7 | 0.60 | 139 | | | |
| | Exhaust Fan | 77 | 2.0 | 0.60 | 40 | | | |
| | Pressurization | 12 | | | | 599 | | |
| Peak ISDCT | Supply Fan | 324 | 2.9 | 0.60 | 248 | | | |
| | Return Fan | 285 | 1.5 | 0.60 | 109 | | | |
| | Exhaust Fan | 27 | 0.3 | 0.60 | 2 | | | |
| | Pressurization | 12 | | | | 358 | 240 | 0.60 |

TABLE 4
Peak Chiller System Savings

| | Normal Peak | ISDCT Peak | |
|---|---|---|---|
| Supply airflow | 400,000 | 324,000 | Cfm |
| Ambient enthalpy | 46.0 | 46.0 | Btu/Lb |
| Ambient intake fraction of supply airflow | 0.22 | 0.12 | |
| Ambient intake airflow | 89,000 | 39,000 | Cfm |
| Pressurization airflow | 12,000 | 12,000 | Cfm |
| Exhaust airflow | 77,000 | 27,000 | Cfm |
| Return air temperature | 75 | 76 | ºF |
| Return air enthalpy | 28.0 | 27.5 | Btu/Lb |
| Cooling coil entering air enthalpy | 32.0 | 29.7 | Btu/Lb |
| Cooling coil leaving air temperature | 55 | 52 | ºF |
| Cooling coil leaving air enthalpy | 23.2 | 21.4 | Btu/Lb |
| Space sensible air temperature differential | 20 | 24 | °F |
| Total sensible cooling load | 723 | 703 | Tons |
| Total cooling load | 1318 | 1006 | Tons |
| Peak cooling savings | | 311 | Tons |
| Chiller system coefficient of performance | | 0.81 | Kw/Ton |
| Peak chiller demand savings | | 252 | Kw |
| Peak chiller demand savings | | 0.63 | Kw/1000 $ft^2$ |

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