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INTEGRATED DEMAND CONTROLLED VENTILATION FOR SINGLE DUCT VAV SYSTEM WITH CONFERENCE ROOMS

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0. ABSTRACT

Single duct variable air volume (VAV) systems are widely used in office buildings to achieve energy savings. It supplies proper amount of conditioned air to satisfy both the load and the ventilation requirements of each individual zone. To obtain acceptable indoor air quality (IAQ), the overall outside air (OA) intake ratio has to consider the demands from all the zones with the method provide by ASHRAE 62. Some high-ventilation required rooms make it difficult to use a low OA intake ratio.

This paper presents a new integrated demand controlled ventilation (IDCV) methodology which can ensure acceptable IAQ and energy savings with lower OA intake ratio. The requirement on hardware and software is simple and the implementation is easy. One office building model is applied to demonstrate the energy saving and show how the indoor air ventilation be satisfied under different circumstance. The IDCV VAV methodology can be generalized to other similar buildings where the occupancy of critical zones is rare and clear.

1. INTRODUCTION

Single duct VAV system supplies proper amount of conditioned air to satisfy both the load and the ventilation requirements of a building. Terminal boxes are the main components, in a VAV system, controlling the indoor environment by modulating conditioned air temperature and amount to the dedicated rooms. In a commercial building, big amount terminal boxes are designed and installed. The design and operation of the terminal boxes may not only influence the comfort condition of the individual zones which they are serving but also the overall energy saving performance from the AHU side.

In a VAV system, although the terminal boxes and the AHU are controlled separately, they are actually linked together by factors even since the design phase, such as the OA intake ratio. A lower minimum OA intake ratio from AHU side benefits the energy saving but it might sacrifice the ventilation of some terminal boxes at the same time. The ASHRAE 62.1(ASHRAE standard. 2004) proposed a calculation method to determine the overall OA intake which uses the individual space ventilation requirements and allows the designer to take credit for "unused" ventilation air returning from the over-ventilated spaces in the system. Based on this method, the critical zone still has strong influence on the system's OA intake ratio limit, and therefore, the energy consumption. In a typical office building VAV system, conference rooms, because of their variable occupancy and high occupancy densities, normally become the singularities influencing the system. Their high ventilation requirement leads to two issues in fact: the AHU side struggling minimum OA ratio set point and the terminal side over-ventilation when they are unoccupied or lightly occupied.

Few concerns have been paid to the VAV system design for conference room due to the reason mentioned above. The lower minimum ventilation rates under the design occupancy conditions represent the higher OA ratio limit, particularly for interior conference rooms. It is suggested to use either a VAV box with a CO_2 sensor resetting the zone minimum airflow rate or a serial fan power box with zero minimum airflow set point (California energy commission. 2003). Further more, demand-controlled ventilation (DCV) was also proposed for conference room VAV design and operation (Demand-Controlled Ventilation: A Design Guide, 2003). The basic idea is to adjust outside ventilation air based on the number of occupants and the ventilation demands that those occupants create. CO_2 sensors, occupancy sensors, and real time data are used to count occupants or know with high reliability when people are occupying a space. (Erik, etc. 2006)

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One drawback of the DCV is the implementation may be too complicated for daily operation and require proper hardware and software to carry out the function. The other is that the location and quantity of the CO_2 sensors, or the real time data also create another barrier to the accuracy of the control algorithm if the system is to change the OA intake ratio from time to time. So far there are no definitive answers applying throughout.

Meanwhile, the minimum airflow rates for the terminal boxes were also found to use the same set points for occupancy and un-occupancy, which range from 30% to 90%. (Zhu Y, etc. 2000). This can cause huge amount energy consumption when the space is lightly occupied or unoccupied, and it is not dealt with the existing DCV methodology.

This paper presents the basic engineering analysis regarding to the single duct VAV optimization of conference rooms and proposes an IDCV methodology for the system design and operation. It is shown that an IDCV logic for system with critical zone is practical, energy saving and prevents wasteful over-ventilation by lessening the impact of critical zones on system overall OA intake. This logic resets both the minimum airflow rate and the maximum airflow rate of the terminal box based on the conference room occupancy. The requirement on the hardware and software for the implementation is simple. Comparing to the DCV method, it is more reliable without much data transportation, and will not cause excessive cost on the investment. The analysis is conducted for the full occupancy condition, partial occupancy condition, interior zone condition and exterior zone condition. The energy saving quantitative comparison to the other control methods based on BIN data is presented.

2. IDCV THEORY AND PRINCIPLES

ASHRAE 62 is a ventilation standard that explains how to design and construct a space that has an acceptable quantity of ventilation air. The appendix A stated the calculation for the outdoor air intake airflow for single supply system, where all the ventilation air is a mixture of outdoor air and re-circulated air from a single location, e.g., reheat, single-duct VAV, single-fan dual-duct, and multi-zone. The basic equations are given as 1-3:

$$X = \frac{\sum Q_{oa,i}}{\sum Q_i} \qquad (1)$$

$$Z = \max\left\{\frac{Q_{oa,i}}{Q_i}\right\} \qquad (2)$$

$$Y = \frac{X}{1 + X - Z} \qquad (3)$$

With real-time data known regarding the Q_i of each zone, the system total outdoor airflow is determined for any time. The ventilation requirements for each individual space, once it is occupied, should always be satisfied although the supply air amount varies with thermal load. The IDCV method also abides by the ventilation standard for an acceptable IAQ stated in the ASHRAE standard.

Figure 1 depicts the basic procedure for an IDCV VAV system design regarding the ventilation aspect and the overall OA intake ratio. Unlike the conventional design method, the IDCV first evaluates the initial OA intake ratio, and lowers it down until a reasonable level by changing the vital set points of the critical zone terminal box. With this logic, the overall AHU side OA intake ratio and the maximum and minimum airflow rates set points for the terminal box are optimized to obtain the goal of saving energy and preventing over-ventilation. Hereby, the $V_{\max}$ is no longer a traditional definition but the maximum constant air flow set point of the terminal box under summer occupancy scenario. The operation of the conference room terminal box is defined:

1. *Summer occupancy scenario:*

Whenever the conference room is occupied in summer, the control is overridden to “Summer Occupancy” status. The primary air damper modulates to provide 1.5 (adjustable) times maximum conditioned air to the room. The reheat valve modulates to maintain the room temperature;

2. *Summer/Economizer un-occupied scenario:*

Deactivate the terminal reheat, set minimum air flow to 0 cfm;

3. *Winter occupancy scenario:*

Whenever the conference room is occupied in winter/economizer season, the control is overridden to “Free Cooling Occupancy” status. The controller reset the minimum flow set point according to the outside air intake ratio range and modulates no less than this amount air to the room. The reheat valve modulates to maintain the room temperature when the air flow hits the minimum set point;

4. *Winter/Economizer un-occupied scenario:*

Treat un-occupancy scenario the same with off hour scenario, set calculated minimum air flow as the flow set point if temperature is too low.

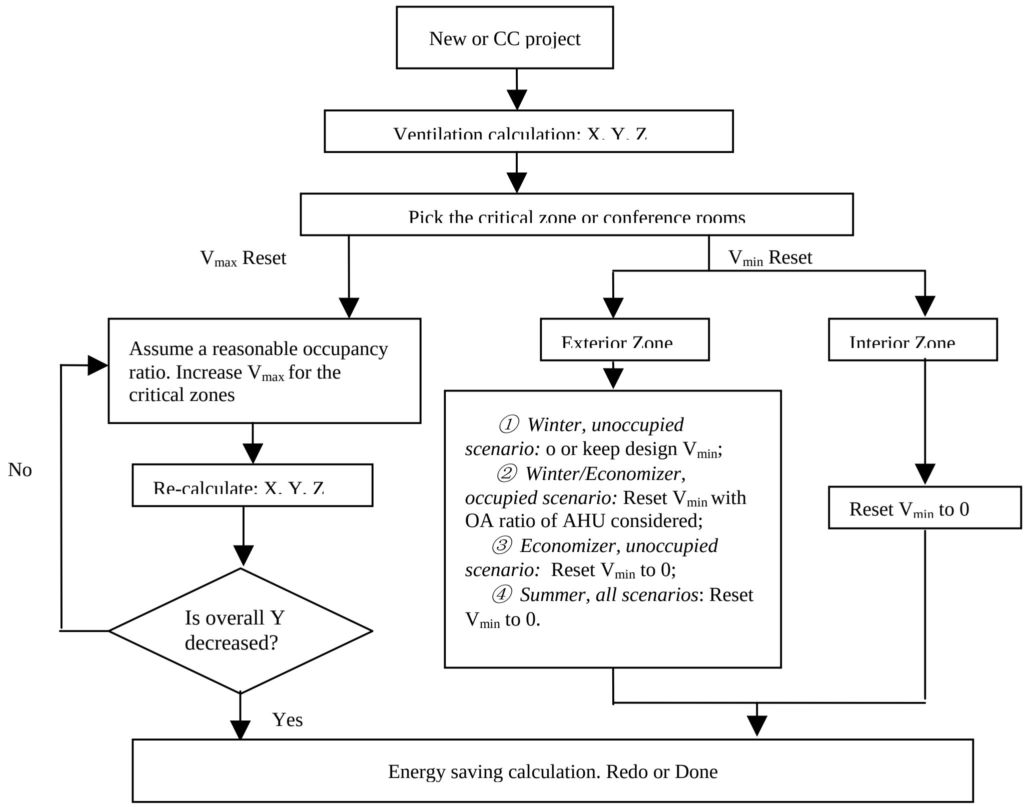

Figure 1: IDCV Design Principles Flowchart

With the IDCV, the terminal box of the critical conference room is overridden by the occupancy sensor to supply the constant reset V_{max} conditioned air during summer season when it is occupied.

Therefore, the ventilation can ensure the high quality IAQ for the conference room.

The detailed operation of the terminal box in the conference room is depicted in Figure 2.

During winter/economizer season and occupancy status, logic resets the minimum flow set point regarding to the outside air intake ratio range, and the controller modulates no less than this amount air to the room. The difference between the scenarios is the terminal box works as VAV for economizer occupancy and as constant air volume for winter occupancy. The reheat valve modulates to maintain the room temperature when the air flow hits the minimum set point.

When the room is un-occupied in summer or economizer season, the VAV box maintains the room temperature to the set point by modulating the airflow which is defined between the V_{min}, which is reset to 0, and the original V_{max}. The re-heat is not very likely to work except the conference room is located in an exterior zone and the building is located in a cold climate.

When unoccupied in winter, the room minimum airflow is determined upon the location of the conference room and the building. The basic idea behind is to ensure the room can always maintain an acceptable temperature while the minimum airflow is changed for energy saving consideration. For hot

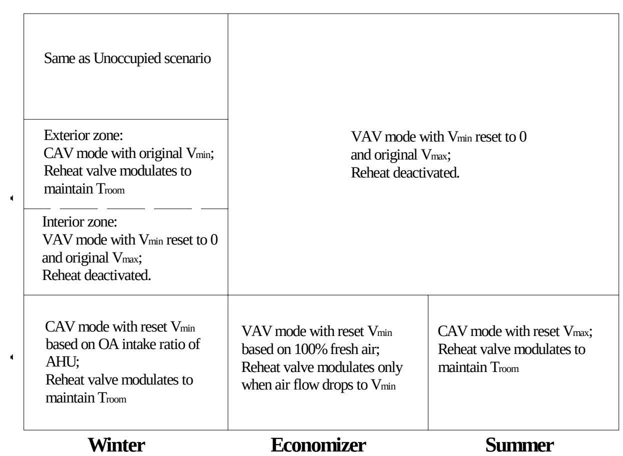

Figure 2: Terminal Box Operation Details with IDCV Logics

climate, the V_{min} can be simply reset to 0 since the conference room is not to have heating load.

3. METHODOLOGY VALIDATION

3.1 Model Construction

To simplify the energy saving calculation and analysis of the IDCV methodology from the system scope, a three-zone building is illustrated in this paper, and this algorithm can be generalized to any real office building. The terminal box dynamic simulation and energy performance in the conference room with the IDCV logic is given in the other paper (Yuebin Yu, etc. 2007).

To calculate the energy saving of a building with the IDCV method, we assume the conference room has 10% occupancy time ratio while the building is with 24/7 schedule. The preliminary design condition (case 1) is listed in Table 1:

Table 1: Preliminary Design Condition

Zones	Zone1	Zone2	Zone3
SA (cfm)	10000	20000	3000
OA (cfm)	1000	2500	1500
X	Y	Z	Overall OA (cfm)
0.15	0.23	0.5	7590

With the calculation method provided by the ASHRAE 62.1, the whole building ventilation parameters can be obtained for both occupancy and un-occupancy situation with the IDCV methodology (case 2). Assuming the V_{max} for the summer occupancy scenario of the conference room area reset to 1.5 times of the original 3000cfm which is 4500cfm, the new ventilation parameters are calculated by the equations (1) to (3) and listed in Table 2.

The total OA intake of IDCV is obtained with the occupancy time ratio considered. The real time distribution of the occupancy and un-occupancy doesn't much influence the overall calculation.

With the amount of OA intake and the local BIN data, the energy saving can be obtained using air model. The energy saving with the IDCV method applied includes two aspects:

1. Coil thermal energy saving due to less OA intake, and

2. Fan energy saving due to different airflow rate.

For the thermal energy consumption calculation, it can be calculated by the equation (4) provided the hourly Q_t is known, which can be obtained with either software load simulation or real trending data. Meanwhile, with the equation (4), the dry coil or wet coil condition should be properly obtained to avoid wrong energy saving calculation.

$$L_t = (h_m - h_s)Q_t \quad (4)$$

In this study, the hourly load is not simulated or measured, therefore the hourly Q_t is not known. An alternative calculation is deduced to acquire the thermal energy saving.

Based on energy and mass constant law, the equation (5) to (6) can be deduced:

$$Q_t h_m = Q_{rt} h_r + Q_{ot} h_o \quad (5)$$

$$Q_t = Q_{ot} + Q_{rt} \quad (6)$$

Multiple both side of the equation with h_r, it is changed to be:

Table 2: Comparison of Ventilation Parameters for Occupancy and Un-Occupancy

	Occupancy (10%)				Un-Occupancy (90%)			
	Zone 1	Zone 2	Zone 3	Total	Zone 1	Zone 2	Zone 3	Total
SA (cfm)	10000	20000	4500	34500	10000	20000	--	30000
OA (cfm)	1000	2500	1500	5000	1000	2500	0	3500
Z_i	0.1	0.125	0.33	--	0.1	0.125	0	--
X	0.15				0.117			
Z	0.33				0.125			
Y	0.18				0.118			
OA (cfm)	6210				3540			
OA Final (cfm)	3807							

$$Q_t h_r = Q_{ot} h_r + Q_{rt} h_r . \quad (7)$$

Combining the above four equations, the following equation is obtained:

$$\begin{aligned} L_t &= Q_{ot} h_o + Q_{rt} h_r - Q_t h_s \\ &= Q_{ot} h_o + Q_t h_r - Q_{ot} h_r - Q_t h_s \\ &= Q_{ot}(h_o - h_r) + Q_t (h_r - h_s) \end{aligned} \quad (8)$$

For case 1 and case 2, the total energy consumption from indoor air can be regarded as identical or similar, which is presented by the last item of equation (8). Within the equations, the h denotes the energy sign which might be properly calculated by the air enthalpy difference or the temperature difference based on the circumstances.

The thermal energy saving is gained from the OA part of equation (8). Economizer is considered for both the basic condition (case 1) and the IDCV condition (case 2) from 35°F to 65°F.

For the fan energy saving calculation, the building load profile should first be known. The studied building load profile is given in Table 3. With constant supply air temperature, it is assumed that the fan air flow is proportional to the load ratio. 33000cfm conditioned air covers 100% building load in case 1. For case 2, the fan air flow will be 91% percent of that of the case 1 since 30000cfm conditioned air covers 100% building load while the conference room is unoccupied, which composes 90% of the operation time. At the same time, the fan air flow of case 2 under 100% load will be 105% percent of that of the case 1 when the conference room is occupied, which consists of 10% of the operation time.

Table 3: Building indoor load percentage and distribution

Load ratio (%)	10	20	30	40	50	60	70	80	90	100
Distribution based on time (%)	4.27	4.27	6.84	7.69	12.82	17.09	25.64	12.82	5.13	3.42
Hours/ year	374	374	599	674	1123	1497	2246	1123	449	299

The fan energy consumption can be generally expressed as equation (9):

$$\begin{aligned} E_{fan} &= Q_t \times h_f / \eta_f = Q_t \times (h_{f,main} + P_s)/\eta_f \\ &= Q_t \times (h_{f,main} \times \left(\frac{Q_t}{Q_d}\right)^2 + P_s)/\eta_f \\ &= Q_t \times (h_{f,main,d} \times \gamma^2 + P_s)/\eta_f \end{aligned} \quad (9)$$

The corresponding motor power consumptions are calculated:

$$E_{mt} = \frac{E_{fan}}{1000 \times \eta_f \times \eta_b \times \eta_m} \quad (10)$$

To simplify the fan power calculation, set $\eta_f \times \eta_b \times \eta_m = 0.7$ for all the fan speed.

The fan head is set as 5 inch water for 100% airflow while the pressure lost in the main duct is 3.5 inch water. To simplify the calculation, no static pressure reset is considered and it is remained as constant 1.5 inch water.

3.2 Energy Saving Result

This paper assumes the location is in OMAHA. With the IDCV methodology described and the model developed in the above sections, the energy saving can be calculated within spreadsheet. In summer mild weather, such as the night hours, the outside air could be humid and cool, and the outside air may need both dehumidification and reheat. This should be considered in the calculation. The total energy saving for one year is the summation of the product of the energy consumption difference and the corresponding hours.

The thermal energy saving for each hour based on BIN data is listed in Table 4. Value in bracket means negative. Since for both cases, the real OA intake ratio is higher than 23% in the winter and economizer season, there is no energy saving during this period. The dry condition calculation shows when the OA is below 75°F, the dehumidification and reheat might happen.

From Table 4, it is shown the thermal energy saving from AHU could be 852 MMBtu/year for the studied case. The amount is estimated to be around 15% of the overall thermal energy consumption of the system.

For the fan energy consumption, the calculation result is shown in Table 5. The fan power unit is converted into Btu for better comparison. As an offset, the terminal box reheat consumes some energy when the conference room is occupied. The calculation result shows that waste is tiny comparing to the fan and coil energy saving with the IDCV logic.

Table 4: Thermal Energy Saving from IDCV Logic

Time period		1-4	5-8	9-12	13-16	17-20	21-24
Average dry bulb Temp (^{0}F)	Dry condition (Btu/Hr)	Energy Saving based on enthalpy difference (Btu/Hr)					
97.00	87387.30	0.00	0.00	0.00	245138.40	197472.60	0.00
92.00	67526.55	0.00	0.00	245138.40	245138.40	245138.40	0.00
87.00	47665.80	153211.50	0.00	231519.60	231519.60	214496.10	197472.60
82.00	27805.05	85117.50	102141.00	197472.60	136188.00	180449.10	214496.10
77.00	7944.30	119164.50	102141.00	136188.00	105545.70	132783.30	180449.10
72.00	(11916.45)	56177.55	57879.90	105545.70	54475.20	86819.85	132783.30
67.00	(31777.20)	8511.75	8511.75	54475.20	10214.10	27237.60	74903.40
Total Thermal Energy Saving		852.8 MMBtu					

Table 5: Fan Power Consumption

	Fan energy
Case 1	312 MMBtu
Case 2	288 MMBtu

From the energy saving example, it is shown the IDCV methodology for the single duct VAV system with critical conference rooms can save about 876 MMBtu year around and most of the energy saving are from the OA intake reset. In the whole process, the building ventilation can meet the requirement of ASHRAE 62 for acceptable IAQ.

3.4 Ventilation Consideration

Based on IDCV operation logic, the IAQ of the critical conference room can always be guaranteed to a better level with more fresh air for the occupants comparing to that of the previous case. The system ventilation accomplishes a dynamic reset with a lower minimum threshold by lessening the influence of the critical room.

Since the minimum airflow rate of the critical room is optimized with the IDCV method, the ventilation might be concerned if it is located in the perimeter zone. During winter season and the conference room is not occupied, the minimum airflow rate should be reset from 0cfm to a safe value once the room temperature drops below a tolerance.

This value should ensure the reheating performance and air distribution while avoiding the buoyant effect and preventing the room from freeze-up when the room is not occupied in winter. When the room is occupied, the V_{min} is dynamically reset for high IAQ according to the AHU side fresh air intake ratio, which might also be reset stage by stage according to the outside air temperature.

4. CONCLUSION

The office building single duct VAV system is analyzed. To meet the ASHRAE IAQ requirement, demand controlled ventilation (DCV) VAV system was suggested by the other researchers. This method is good provided the real time airflow rate of each terminal box is known, the DDC data transmission is reliable and the supervisory controller can deal with the data timely. This requires higher cost for the whole VAV system, and better system operation.

A new IDCV concept and methodology is proposed and analyzed to overcome the drawbacks of the DCV system. This method can be widely used to achieve energy saving and acceptable IAQ for the office building by lessening the influence of critical zones. The hardware and software investment is just like normal VAV system while the energy saving can be huge from thermal part. Occupancy sensor for the conference rooms is required and should sent basic digital input signal to the local and supervisory controller. The terminal box controller has two sets work schedule for occupancy and un-occupancy with simplified dynamic set points.

Energy saving is also analyzed with a concrete model. The calculation result demonstrates the methodology is useful and can saving huge amount of energy while maintain the IAQ to the standard.

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6. Nomenclature:

Q = air flow rate, cfm

X = uncorrected outdoor airflow fraction

Y = corrected outdoor airflow fraction

Z = critical space ventilation fraction

L_t = energy consumed at time t, Btu/h

h = air enthalpy, the subscript s, m, o, and r stand for supply air, mixed air, outside air and return air respectively, Btu/Pound

h_f, $h_{f,main}$, P_s = fan head, pressure drop of the main duct and static pressure set point respectively, inch

γ = flow ratio at time t

η_f, η_b, η_m = efficiency of fan, belt and motor respectively

c = dry air specific heat, 1.05 Btu min/ft^3 ^{0}F h

T_i = indoor air temperature, 75^{0}F

T_s = supply air temperature, 55^{0}F during the summer and economizer season and 65^{0}F during the winter season