ENERGY AND VENTILATION PERFORMANCE ANALYSIS FOR CO_2-BASED DEMAND-CONTROLLED VENTILATION IN MULTIPLE ZONE VAV SYSTEMS WITH MULTIPLE RECIRCULATION PATHS

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ABSTRACT

CO_2-based Demand-Controlled Ventilation (DCV) has been proved to be energy-efficient by altering ventilation rates according to surrogated indications of CO_2 levels. Numerous studies have implemented CO_2-based DCV strategies for single zone Heating, Ventilation, and Air-conditioning (HVAC) systems and multiple zone single-path Variable Air Volume (VAV) systems. However, DCV for multiple zone VAV systems with multiple recirculation paths is still untapped, including VAV systems with Series Fan-Powered Terminal Units (SFPTUs), Parallel Fan-Powered Terminal Units (PFPTUs), and Dual-Fan Dual-Duct (DFDD) system.

In this paper, energy and ventilation performance of the DCV strategies with system- and zone-level dynamic resets for these three systems are evaluated. The DCV control sequences for three systems are first briefly summarized. A co-simulation combining EnergyPlus with CONTAM is adopted. All the ventilation related control strategies satisfy ASHRAE standard 62.1 and other control sequences (e.g., local terminal unit controls) follow ASHRAE Guideline 36. DCV controls for three systems are implemented in Energyplus Energy Management System (EMS) module. New EnergyPlus actuators (e.g., zone minimum airflow) are added and a customized Energyplus is built to achieve the DCV control for each system. An office building is used as a case study to demonstrate the benefits from DCV strategies for multiple recirculation path systems in four different climates. For each system, two baselines for ventilation requirements (ASHRAE 62.1, California Title 24) are considered. The simulation results show that the DCV control logic could lead to 7-14%, 7-21%, 1-8% HVAC source energy savings for SFPTU, PFPTU, and DFDDTU systems respectively compared with the baseline ASHRAE 62.1 approach. Three systems have similar ventilation performance and could achieve a good compliance with the ventilation requirements in Standard 62.1.

INTRODUCTION

Demand controlled ventilation (DCV) is defined as a system that achieves 'automatic reduction of outdoor air intake below design rates when the actual occupancy of spaces served by the system is less than design occupancy (ASHRAE 2019).' ASHRAE Standard 90.1 mandates the DCV system for densely occupied spaces since the 1999 version and also requires the DCV system be in compliance with ASHRAE Standard 62.1 (ASHRAE 2019), which stipulates that the minimum outdoor air intake be based on the sum of ventilation rates required to dilute pollutants generated by occupants (e.g., bioeffluents). Measuring zone carbon dioxide (CO_2) concentration could be an indirect approach to monitor the concentration of bioeffluents generated by occupants since, in general, CO_2 generation rate is proportional to odorous bioeffluent generation rates (Lin and Lau 2014).

CO_2-based DCV has been popular in the heating, ventilation and air-conditioning (HVAC) industry for years. Using CO_2-based DCV in simple single zone HVAC systems is relatively well understood and ASHRAE Standard 62.1 User's manual (ASHRAE 2019) has provided a detailed procedure on how to apply CO_2-based DCV in such simple system since 2004. To be specific, the Standard 62.1 User's Manual includes an appendix showing the underlying theory and a control scheme for using carbon dioxide (CO_2) concentration for DCV in accordance with the Ventilation Rate Procedure (VRP) of ASHRAE Standard 62.1.

There also exists a few CO_2-based DCV control strategies for multiple zone HVAC systems in literature and practice including: 1) Supply air CO_2-based DCV control strategy (Warden 2004), and 2) DCV control strategy from California Title 24 (California Energy Commission 2008). Apart from these early studies, Lin et al. proposed the theoretical equations required to use CO_2 concentration as an indicator of realtime occupant-related ventilation requirements (Lin, Lau and Yuill 2013) and developed control logic that provides potential energy savings while ensuring compliance with Standard 62.1 (Lin and Lau 2015). However, their control sequences are complex and iterations used in their control logic algorithms cannot be implemented into real control systems. To address this limitation, O'Neill et al. developed control sequences that are practical and implementable in typical single-duct VAV systems with Direct Digital Control (DDC) systems (O'Neill, Li, Cheng et al. 2017). They tested the proposed logic in realistic simulations that account for varying occupancy and concurrent cooling loads (O'Neill, Li, Cheng et al. 2019). In addtion, they also assessed the propsosed sequences in a well instrumented test facility at Iowa Energy Center's Energy Resource Station.

Although these studies demonstrated a considerable energy saving and a good ventilation compliance implementing DCV, they were limited to single-duct VAV systems. There is a gap for researchers and practitioners to implement DCV following ASHRAE standard 62.1 for multiple zone VAV systems with multiple recirculation paths such as VAV systems with fan-powered terminals and dual fan/dual duct VAV systems. DCV control sequences for these systems have not been developed yet in part due to the mathematical complexity of CO_2 mass balance equations and Standard 62.1 requirements for these systems. In addition, there is a lack of building simulation or field test-beds to evaluate the performance of the proposed control sequences due to the system intricacy. In this context, this paper is intended to address this shortcoming: expanding the DCV system to include multiple zone VAV systems with multiple recirculation paths, such as systems with fan-powered terminals and dual duct systems. First, the simulated systems and corresponding control sequences are briefly illustrated. Next, the development of simulation-based testbed and case description of an office building using four typical climate zones is presented. In the next section, we discuss the energy and ventilation performance of the DCV control sequences followed by some conclusions.

THE STUDIED SYSTEMS AND DCV CONTROL SEQUENCES

Description of the Studied Systems

The studied systems include multi-zone VAV systems with series fan-powered terminal unit, parallel fan-powered terminal unit, and the dual fan dual duct, asshown in Figure 1 (a)-(c).

Local controls (terminals, and air handling units (AHUs) for these three systems comply with ASHRAE Guideline 36 (ASHRAE 2018). Since Guideline 36 offers multiple variations and sequences of control for common terminal types, the following terminal control sequences are considered: parallel fan-powered terminal unit with a constant volume fan; series fan-powered terminal unit with a constant volume fan; dual fan dual duct VAV terminal unit – snap acting control.

The key sensors for the proposed control logics are:

- System-level CO_2 sensors:

  For the fan-powered terminal unit (FPTU) system:
  - Supply air CO_2 sensor at the AHU level.
  - Return air CO_2 sensor at the AHU level.

  For the dual fan dual duct (DFDD) system:
  - Return air CO_2 sensor at the AHU level
  - Outdoor air CO_2 sensor at the AHU level.
- Densely occupied zones with CO_2 sensors.
- Sparsely occupied zones with no additional controls.
- Other zones with occupancy sensors.
- Inlet air differential sensor within the primary airstream for Fan-powered terminal units (FPTU); Dual airflow sensors for dual fan dual duct VAV terminal units, one at the hot airflow inlet, one at the cold airflow inlet.

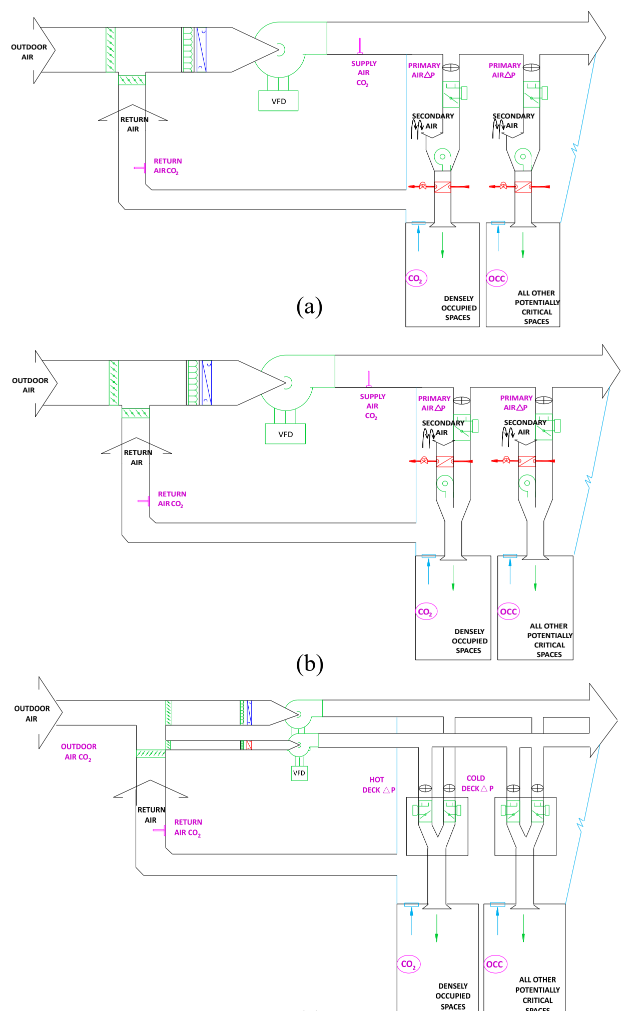


Figure 1 Control schematic of VAV system with (a) series fan-powered terminal units (b) parallel fan-powered terminal units (c) dual fan dual duct system

Summary of the DCV Control Sequences

The sequences developed for multiple zone VAV systems with multiple recirculation paths is similar to sequences developed in the ASHRAE RP-1747 (O'Neill, Li, Cheng et al. 2017), in which the logic is broken into zone level and system level calculations so as to reduce network traffic.

At the zone level, actual zone ventilation needs are dynamically determined using CO_2 (or occupancy) and airflow sensors for occupant component, adjusted for zone air distribution effectiveness based on supply air temperature. For the zones with CO_2 sensors, the required breathing zone outdoor airflow (V_{bz}) is calculated using the readings from CO_2 and airflow sensors in different locations for three different systems. The zone primary airflow minimums are then reset using Trim & Respond (T&R) logic. If the primary air is rich with outdoor air due to an economizer operation, zone minimums are reduced. Otherwise, the zone minimums are increased for critical zones only to ensure the required outdoor airflow rate at AHU is never above the design rate. At the system level, the required AHU outdoor air intake (V_{ot}) is dynamically determined using the ASHRAE Standard 62.1 VRP. This value is then the input to the economizer control to maintain the supply air temperature by adjusting the outdoor air and return air damper positions. On top of that, the outdoor air ratio (OAR, the ratio of actual outdoor airflow to the required AHU outdoor air intake) is calculated to determine whether the outdoor air is rich and sufficient. The OAR rich binary value is broadcasted to the zone controllers to adjust minimum setpoints in the T&R logic.

The rationale behind the logic is that when primary air is mostly outdoor air due to economizer operation, zone airflow minimums can be very low, all the way down to the zone minimum ventilation rate when supply air is 100% outdoor air. When the economizer is disabled, the zone minimums in the critical zones go the other way: they are increased to induce more primary airflow in the critical zones. It is possible for the minimum airflow in the critical zone to increase to its maximum airflow.

For the detailed descriptions of the DCV control sequences for these three multizone recirculating systems, please refer to the updated ASHRAE guideline GP36 to be published. Since the developed sequence is similar to the one developed in the ASHRAE RP-1747 (O'Neill, Li, Cheng et al. 2017), these documents could also be referred (O'Neill, Li, Cheng et al. 2017, O'Neill, Li, Cheng et al. 2019). The major differences lie in (1) Calculation of required breathing zone outdoor airflow V_{bz} using CO_2 mass balance; (2) Definition of the critical zones in different systems; (3) Calculation of the system ventilation efficiency E_v (or) for different systems using the ASHRAE Standard 62.1 Ventilation Rate Procedure. Apart from these details, the general ideas behind the logics are the same.

SIMULATION DETAILS

Simulation Platform

Co-simulation (Dols, Emmerich and Polidoro 2016) of EnergyPlus and CONTAM (Walton and Dols 2016) through Functional Mock-up Unit (FMU) is conducted as demonstrated by the team in RP-1747 (O'Neill, Li, Cheng et al. 2017). EnergyPlus is used for the energy simulation while CONTAM is used for the airflow simulation. The outputs from CONTAM program provide the zone infiltration flow rates and zone mixing air flow rates to EnergyPlus. These variables overwrite the counterparts in the zone contaminant calculation in EnergyPlus. On the other hand, EnergyPlus takes care of the zone contaminant calculation, demand control ventilation, and building energy simulation. It sends back CONTAM boundary conditions such as zone air temperatures, outdoor environmental parameters, system level air flow rates, and the outdoor air fractions. The results from EnergyPlus are also used to evaluate the energy savings potentials from the proposed practical DCV control strategies and verify whether ASHRAE Standard 62.1 required ventilation rates can be satisfied in each space.

Case Description

The candidate building for simulation is the former Iowa Energy Center Energy Resource Station (ERS), as shown in Figure 2. This building was the testbed for ASHRAE RP-1747. Simulation inputs including building geometry, internal heat gains, and dynamic occupancy schedules are exactly same with those in the EnergyPlus model of RP-1747 (O'Neill, Li, Cheng et al. 2019). HVAC equipment and system in original RP-1747 model are replaced with the system of interests for this work. There are eight test rooms, four offices, two classrooms, and a media center. The proposed DCV control strategies are applied to eight test rooms (office West A, office West B, office South A, office South B, office Interior A, conference East A, conference South B, and conference Interior A). Three conference test rooms install CO_2 sensors. Four office rooms install occupancy sensors and one open office West A does not install either CO_2 sensor or occupancy sensors. Two variable air volume (VAV) air handling units (AHUs) serve all the rooms. One serves the eight test rooms and the other serves the rest of rooms. The cooling source is the electric chiller and the heating source is the natural gas boiler. The terminal units are VAV boxes with hot water reheat coils in reality.

Simulation are conducted in the following ASHRAE Climate Zones to assess the performance of the DCV control sequences: 1A (Miami), 3A (Atlanta), 3C (Oakland), and 5A (Chicago). In the following, we will use 3A (Atlanta) as an example to show the simulation setup as well as the detailed results in avoidance of repetition. For the assessment of the energy and ventilation performance, we will show the result comparison of four different climate zones.

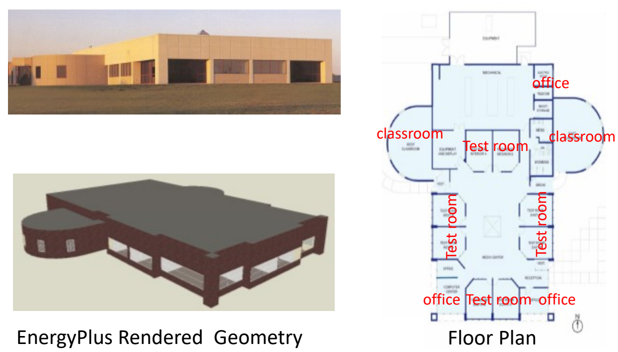


Figure 2 EnergyPlus Render Geometry and Floor Plan of ERS Building

EnergyPlus Models

EnergyPlus models are developed for three systems (Faris, Int-Hout and O'NEAL 2017, Sardoueinasab, Yin and O'Neal 2018) using the ERS building in four different climate zones. We use the version 9.0.1. The internal load schedules and occupancy schedules for each zone are taken from RP-1747. Figure 3 (a)-(c) show the air loop configurations in EnergyPlus for three systems. The VAV terminal local controls for these three systems are constant volume series fan-powered VAV zone control, constant volume parallel fan-powered VAV zone control, and snap-acting dual duct VAV zone control (ASHRAE 2018). Please note that although the reheat coil locations in Figure 3 (b) for PFPTU system are different from ones in ASHARE Standard GP36, the simulation results will not be changed. The economizer control is using a fix dry bulb temperature following ASHRAE 90.1. The high-limit shut off temperature is 66 ˚F, 66 ˚F, 75 ˚F, and 70 ˚F for Miami, Atlanta, Oakland, and Chicago, respectively.

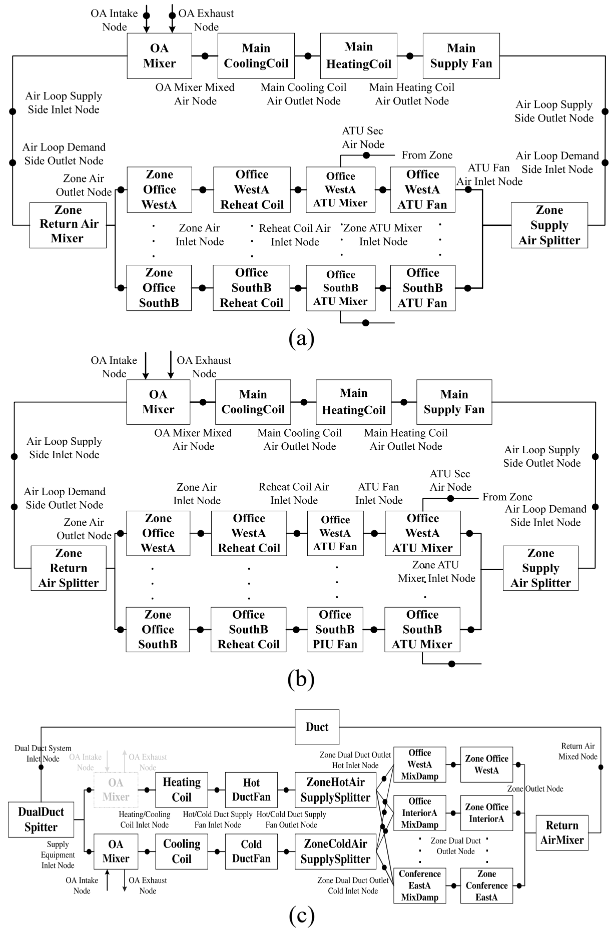


Figure 3 Air Loop Configuration of (a) SFPTU System (b) PFPTU System (c) DFDD System in EnergyPlus

To implement the proposed control sequences including the DCV control and local terminal unit controls, EnergyPlus Energy Management System (EMS) module is adopted. EMS sensor module will get the CO_2 concentrations from individual zones, and then pass the information to the EMS subroutines with the control logics. The DCV control requires a dynamic reset of zone and system level air flow minimum setpoints (minimum primary airflow for FPTUs, and zone minimum airflow for DFDDTUs) and this is achieved by using actuators in EMS that override actuator's variable inside EnergyPlus whenever it needs to be reset. However, the standard EnergyPlus package does not have available actuators for zone level air flow minimum setpoints. Therefore, a customized EnergyPlus needs to be compiled to have the actuator of 'Minimum Primary Air Flow Fraction' in *AirTerminal:SingleDuct :SeriesPIU:Reheat* Module and *AirTerminal:Single Duct:ParallelPIU:Reheat* Module, as well as 'Zone Minimum Air Flow Fraction' in *AirTerminal: DualDuct:VAV* Module. The system-level AHU minimum outdoor air flow is dynamically reset by manipulating 'Minimum Outdoor Air Schedule' in *Controller:OutdoorAir* Module, which is already existing in the actuator list from a standard EnergyPlus. On top of that, actual control logics such as Trim & Respond logic to reset the zone minimum setpoints with "requests" are modeled using EMS subroutines and such logic only involves simple and straightforward mathematical equations. In the field testing, RP-1747 control sequence uses a trim and response ratio of 5% with an updating frequency of 1 minute. Due to the limitation of EnergyPlus building envelope conduction heat transfer algorithm, it is not feasible to have a time step of 1 minute in EnergyPlus. Simulation time steps in both EnergyPlus and CONTAM were set as 5 minutes. The trim and response ratio is set as 12.5% for all simulation studies presented in this paper.

CONTAM Model

CONTAM model was built for exchanging air flow information with EnergyPlus. It was created through the pseudo-geometry concept, without drawing the actual building floor plans to scale. It is required to have inputs, including the actual thermal zone areas, window sizes, and door sizes, etc. The AHU system is also required for supply air terminals and return air terminals at each thermal zone. The CONTAM model for ERS building is shown in Figure 4. AHU-1 is responsible for the eight test rooms and AHU-2 is responsible for the other thermal zones. Only the performance related to AHU-1 is analyzed in this study.

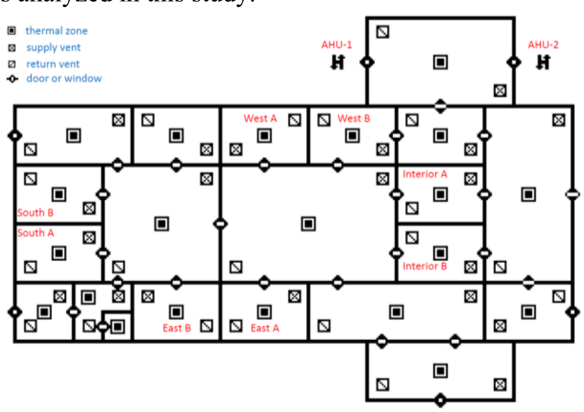


Figure 4 CONTAM Model for ERS Building

Description of Baseline Cases

Two baselines are considered for the three systems. The first baseline is following a simplified compliance approach from Addendum f in ASHRAE 62.1-2016, while the second baseline is following California Title 24. The simplified ASHRAE 62.1 approach is used to replace the current Table 6.2.5.2 in ASHRAE 62.1. The new approach would provide a new method to determine zone ventilation efficiency (E_v) and also determine zone minimum primary airflow as 1.5 times of zone required outdoor air (V_{oz}). The primary goal with this approach, and why it is used as a baseline here, is that it provides a simple and deterministic approach to establish the system level required outdoor air (V_{ot}). The zone air flow minimums (V_{z_min}) for eight test rooms are listed in Figure 5 with zone air distribution effectiveness (E_z) values of 0.8. The system-level outside air minimums are 582.67 CFM and 880 CFM for ASHRAE 62.1 and California Title 24, respectively.

Room	Simulated Area A_z	Zone Population P_z	Diversified Population	Design Breathing Zone Ventilation	Simplified 62.1 Zone Minimum	Title 24 Zone Minimum
Units	ft²	Number	Number	CFM	CFM	CFM
West B	266	2	1	25.96	48.675	39.9
South A	266	2	1	25.96	48.675	39.9
East B	266	1	1	20.96	39.3	39.9
Interior B	320	8	4	59.2	111	120
Interior A	266	2	1	25.96	48.675	39.9
West A	600	4	4	56	105	90
South B	600	20	13	136	255	300
East A	280	14	10	86.8	162.75	210

Figure 5 Zone Air Flow Minimums for Two Baselines

Zone level design maximum air flow rates in cooling mode and AHU maximum air flow rate for four climate zones are also calculated. Based on the above calculations, zone level terminal parameter settings for the two baselines are calculated for SFPTU, PFPTU, and DFDDTU respectively, including maximum air flow rates, minimum air flow rates, constant fan sizing value, and the fan on flow fraction. It is noted that the maximum secondary air flow rate is assumed to be 60% of the cooling design air flow (Titus 2012). Figure 6 depicts the zone minimum air flow settings for two baselines using Atlanta as an example.

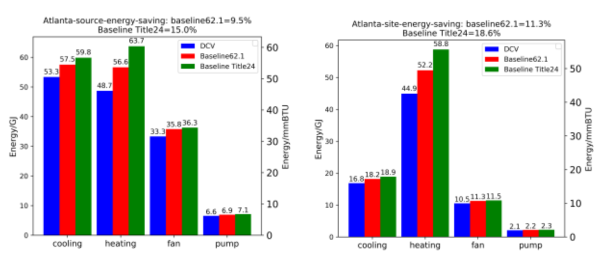
System	Item		Value							
PFPTU	Maximum Primary Air Flow Rate		0.291	0.102	0.192	0.301	0.155	0.243	0.258	0.278
	Maximum Secondary Air Flow Rate/ Constant Fan Sizing		0.174	0.061	0.115	0.180	0.093	0.146	0.155	0.167
	Minimum Primary Air Flow Fraction	T62.1	0.171	0.226	0.120	0.256	0.338	0.076	0.466	0.083
		T24	0.146	0.186	0.098	0.330	0.366	0.078	0.548	0.068
	Fan-On Flow Fraction		0	0	0	0	0	0	0	0
SFPTU	Maximum Primary Air Flow Rate		0.291	0.102	0.192	0.301	0.155	0.243	0.258	0.278
	Minimum Primary Air Flow Fraction	T62.1	0.171	0.226	0.120	0.256	0.338	0.076	0.466	0.083
		T24	0.146	0.186	0.098	0.330	0.366	0.078	0.548	0.068
DFDDTU	Maximum Primary Air Flow Rate		0.291	0.102	0.192	0.301	0.155	0.243	0.258	0.278
	Zone Minimum Air Flow Fraction	T62.1	0.205	0.272	0.144	0.307	0.406	0.092	0.559	0.099
		T24	0.175	0.223	0.118	0.396	0.439	0.093	0.657	0.081

Figure 6 Zone Level Terminal Parameter Settings for the Two Baselines in Three Different Systems

RESULTS AND DISCUSSONS

EnergyPlus-CONTAM co-simulation was conducted in four climate zones for SFPTU, PFPTU, and DFDDTU systems. Source energy savings potential from the proposed control logic and associated ventilation performance (e.g., OAR) are compared with two baselines. Only eight test rooms and associated HVAC equipment are included in the analysis (the ERS facility and the simulation model also include other spaces that are separated from the test rooms). The ventilation performance is evaluated using under-ventilation and over-ventilation hours compared with the proposed ASHRAE 62.1 simplified approach. We will first discuss the detailed results of the energy and ventilation performance using Altanta as an example. Then, in the discussion section, comparison results will be presented in different climate zones for different systems respectively.

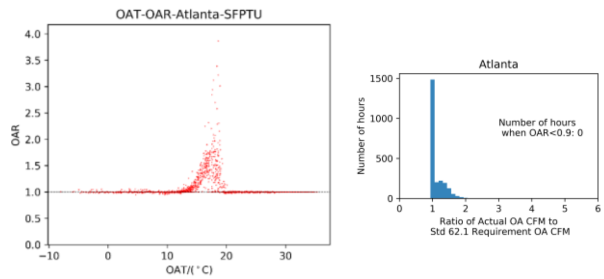
Figure 7 shows the annual HVAC source and site energy saving percentages from the proposed DCV logic compared with two baseline DCVs for the SFPTU system. The site to source energy conversion factors are 3.167 for electricity and 1.084 for natural gas respectively. Baselines from Title 24 (green bar) consumed the most energy compared with others. The proposed DCV control strategy saves 9.5% and 15.0% in terms of source energy compared with the two baselines. It saves 11.3% and 18.6% in terms of site energy compared with the two baselines. This is because the zone minimums from the proposed DCV logic are dynamic and less than those from the baselines. Most of the energy savings come from the heating.



(a) (b)

Figure 7 (a) HVAC Source and (b) Site Energy Consumption by End Use for the SFPTU system in Atlanta

Ratio of actual outside air flow rate (V_{oa}) to system level required outdoor air flow (V_{ot}) is calculated to evaluate the ventilation performance of the DCV control strategy of the SFPTU system. Figure 8 (a) shows the scatter plot of OAR vs. outdoor air temperature (OAT) for the Atlanta case. Figure 8 (b) depicts the OAR distribution in bins. The hourly average data when the system is on (2,340 hours in total) are used. There are no under-ventilated hours (OAR<0.9). The rationale of the selection of 0.9 as a cut-off point is because, in practice, outdoor air flow meters will general exhibit at least 10% measurement error (Lu, O'Neill, Li et al. 2020). As expected, actual outdoor airflow is higher than the required ventilation airflow in the economizer mode.

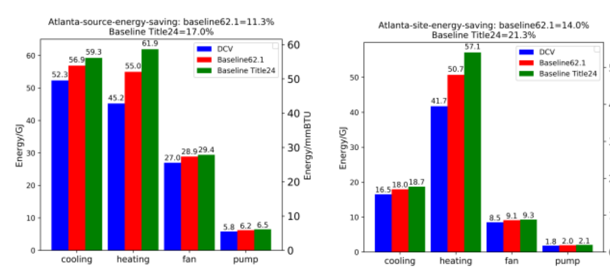


(a) (b)

Figure 8 (a) Scatter plot of OAR vs. OAT (b) Bin plot of OAR for SFPTU system

PFPTU

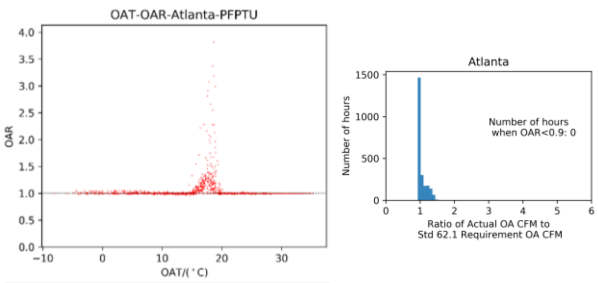
Figure 9 shows the annual HVAC source and site energy saving percentages from the proposed DCV logic for the PFPTU system compared with two baselines. The DCV control strategy saves 11.3 % and 17.0% in terms of source energy compared with the two baselines. It saves 14.0% and 21.3 %in terms of site energy compared with the two baselines. Similar to the results of SFPTU system, most of the energy savings come from the heating. The DCV control strategy for PFPTU system saves a little more energy than SFPTU system because zone primary airflow (V_{pz}) is reset within the range of zone required outdoor air (V_{oz}) and the difference between the cooling maximum air flow ($V_{coolmax}$) and the parallel fan air flow, rather than the range of V_{oz} and $V_{coolmax}$.



(a) (b)

Figure 9 (a) HVAC Source and (b) Site Energy Consumption by End Use for the PFPTU system in Atlanta

Figure 10 (a) shows the scatter plot of OAR vs. outdoor air temperature (OAT) in Atlanta for PFPTU system. Figure 10 (b) depicts OAR distribution in bins. The average hourly data when the system is on (2,340 hours in total) are used. The number of under-ventilated hours (OAR < 0.9) is 0.



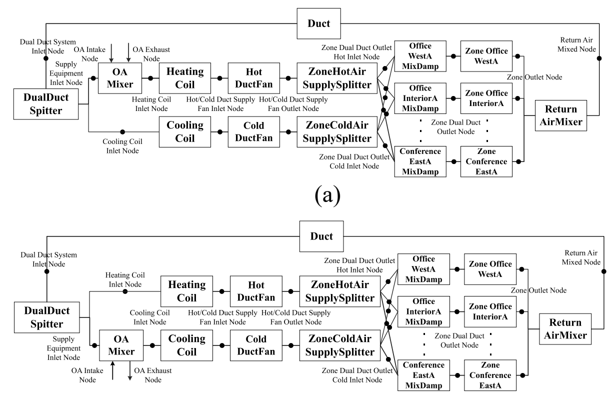
(a) (b)

Figure 10 (a) Scatter plot of OAR vs. OAT (b) Bin plot of OAR for the PFPTU system

DFDDTU

The snap-acting control of the DFDDTU system normally has outdoor air provision in the cold air duct. In the heating season, the cold air duct damper is closed therefore there is no outdoor air coming to the zone which may cause a ventilation problem. In this case, there is no inner zone that requires cooling and hence there is no OA source. To mitigate this issue, we assume that there is also an OA intake in the hot air duct. Therefore, if there is a need, the OA will come from the hot air duct in the heating season while the OA will come from the cold air duct in the cooling season. Due to the limitation of Energyplus, the air loop configuration cannot be changed during the simulation and thus we cannot simulate this proposed OA control through single simulation. Therefore, we built two configurations of the system for the heating and cooling season respectively, as shown in Figure 11. We ran the annual simulation for both configurations and we post-processed the results by merging the results for the heating configuration (Nov. to Mar.) and cooling configuration (Mar. to Nov.).

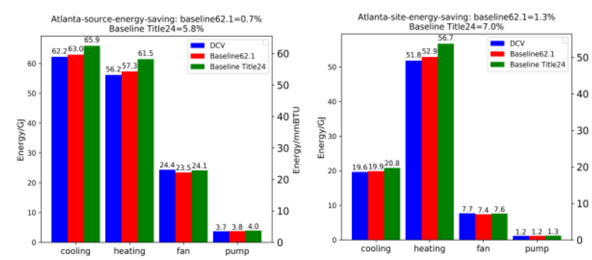
Figure 12 shows the annual HVAC source and site energy saving percentages from the proposed DCV logic for the DFDDTU system compared with two baselines. The DCV control strategy saves 0.7% and 5.8% source energy compared with the two baselines. It saves 1.3% and 7.0 % site energy compared with two baselines. Compared to the two baselines, the proposed DCV control for the DFDDTU system does not have a modest energy saving in Atlanta. The reason lies in the energy penalty for the simultaneous heating and cooling. When the critical zone is under-ventilated, the minimum zone discharging airflow will be set to a higher value. The zone may become overcooling/overheating, and the snap-acting control will turn into the opposite state to maintain the zone temperature setpoint. This energy penalty mostly occurs when the critical zones are in the deadband mode or when the thermal load is small while ventilation demand is huge. Apparently in Atlanta, this condition is often the case. For this reason, the energy saving ratio of the proposed DCV control in Atlanta is not modest.



(a)

(b)

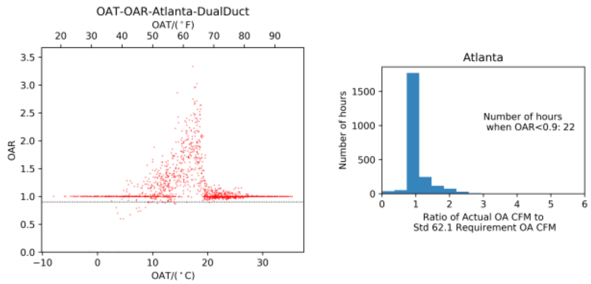
Figure 11 Two configurations of the DFDDTU system for (a) heating and (b) cooling in EnergyPlus



(a) (b)

Figure 12 (a) HVAC Source and (b) Site Energy Consumption by End Use for the DFDDTU system in Atlanta

Figure 13(a) shows the scatter plot of OAR vs. outdoor air temperature (OAT) by climate zone. Figure 13(b) depicts the OAR distribution in bins. The average hourly data when the system is on (2340 hours in total) are used. The under-ventilated hour (OAR<0.9) is 22 and occupies 0.94% of the total system operation hours. The low OAR values mostly occur in the hours during the heating hours when the cold duct (where outdoor air comes from in the cooling configuration) is closed. This is due to the Energyplus limitation that the air loop configuration cannot be changed during the simulation as mentioned. In these hours the OA should also be provided in the hot duct.



(a) (b)

Figure 13 (a) Scatter plot of OAR vs. OAT (b) Bin plot of OAR for the DFDDTU system

DCV Performance Comparison in Four Climates

The energy and ventilation performance in four different climate zones are shown in Table 1, in terms of the source energy saving ratio, source energy saving absolute value, and OAR compliance rate between 1 hour and 5 minute time step data in four climate zones (total 2340 operation hours).

Table 1 Comparison of Energy and Ventilation Performance in Four Climate Zones

Source Energy Saving (Relative Ratio)					
System	**Baselines**	**Miami**	**Atlanta**	**Oakland**	**Chicago**
SFPTU	62.1	7%	10%	14%	10%
	24	11%	15%	22%	17%
PFPTU	62.1	7%	11%	21%	14%
	24	11%	17%	29%	20%
DFDDTU	62.1	4%	1%	8%	2%
	24	9%	6%	13%	7%
Source Energy Saving (Absolute Value, GJ)					
System	**Baselines**	**Miami**	**Atlanta**	**Oakland**	**Chicago**
SFPTU	62.1	11.1	14.1	11.4	22.4
	24	18.3	23.7	19.0	38.9
PFPTU	62.1	10.9	15.7	15.5	27.4
	24	18.2	25.3	23.1	44.1
DFDDTU	62.1	5.9	1.0	7.1	4.7
	24	13.9	8.6	11.9	17.3
OAR Compliance Rate when $V_{oa} \geq 90\%\ V_{ot}$					
System	**Time Step**	**Miami**	**Atlanta**	**Oakland**	**Chicago**
SFPTU	1 hr	100%	100%	100%	99.9%
	5 min	98.5%	97.9%	97.8%	97.8%
PFPTU	1 hr	100%	100%	100%	100%
	5 min	98.6%	98.3%	98.3%	98.2%
DFDDTU	1 hr	99.3%	99.1%	97.2%	98.8%
	5 min	97.9%	97.4%	90.2%	96.5%

In terms of the energy performance in four climate zones, the simulation results show that the DCV control logic could lead to 7-14%, 7-21%, 1-8% HVAC source energy savings for SFPTU, PFPTU, and DFDDTU systems respectively compared with the baseline simplified ASHRAE 62.1 approach. The DCV control sequence in Oakland outperforms than the other zones regarding the source energy saving ratio for all the three systems. This is because the cooling/heating load is small for most of the time, and the actual zonal primary supply air flow rate can be decreased with the reset of the minimum zone air flow setpoints. However, in terms of the absolute energy-saving amount, Chicago saves the most source energy since it has a significant thermal load over the whole year. The energy saving performance of the FPTU systems is slightly better than the DFDD system mainly due to the energy penalty for the simultaneous heating and cooling. In addition, the energy savings in Atlanta and Chicago for DFDDTU systems are relatively modest compared to others.

In terms of the ventilation performance in four climate zones, we can see that the DCV control sequences could achieve good compliance with ventilation requirements in Standard 62.1.

CONCLUSIONS

This paper investigates the energy and ventilation performance of DCV sequences for the multi-zone VAV systems with multiple recirculating paths. Three terminal systems are studied: parallel fan-powered terminal unit with a constant volume fan; series fan-powered terminal unit with a constant volume fan; dual fan dual duct VAV terminal unit – snap acting control. For each system, two baselines for ventilation requirements (ASHRAE 62.1, California Title 24) are considered. An office building is used as a case study to demonstrate the benefits from DCV strategies for multiple recirculation path systems in four different climate zones. The simulation results show that the DCV control logic could lead to 7-14%, 7-21%, 1-8% HVAC source energy savings for SFPTU, PFPTU, and DFDDTU systems respectively compared with the baseline simplified ASHRAE 62.1 approach. Three systems could achieve good compliance with the ventilation requirements in Standard 62.1. The future work includes the testing the DCV control sequences in real facility and the validation of the simulation results. Also, optimal controls will be studied and applied to the DCV of these systems.

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