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Simulation Results and Analysis of Eight Ventilation Control Strategies in VAV Systems

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ABSTRACT

This paper presents eight ventilation control strategies and their annual energy and indoor air quality simulation results for an academic building. The results show that without tempering at the terminal boxes, no ventilation strategy could satisfy the outdoor air requirements when the thermal loads are low, and the fixed outdoor air percentage method is the worst one. From an economic perspective, strategies using optimization techniques minimize the operation energy demand and consumption. Supply air temperature and primary airflow rate are the two proper optimizable parameters on the air side of heating, ventilating, and air-conditioning (HVAC) systems.

INTRODUCTION

Indoor air quality has become a concern in the building engineering community and great effort has been devoted by researchers and engineers to this issue in recent years. Many publications discuss either determining the outdoor air (OA) quantity for a space or per person (ASHRAE 1989) or control equipment, instrumentation, and algorithms to ensure the required OA quantities (Elovitz 1995; Levenhagen 1992; Mumma and Wong 1990). Nevertheless, the literature does not contain information on the possible control strategies that might be employed to systematically vary the OA quantities applied to a single building. The multiple spaces equation (Equation 6-1) of ASHRAE Standard 62-1989 relates the zonal and systematic OA requirements. The equation expresses the relationship of the OA flow rate and the primary airflow rate since the OA is usually delivered by the primary airflow. However, the original equation limits the applications to systems without secondary (local) recirculation air. In addition, the original equation is based upon volumetric flow rates. Hence the actual mass flow rates of OA are dependent upon air density or temperature and relative humidity. The density of air for summer or winter design days can be around ±18% that of air at standard indoor design conditions. As a result, a volume-based OA ventilation standard can result in either excessive energy utilization in the winter or underventilation in the summer. Therefore, the ventilation rates from the Standard 62 table were converted to mass flow rates at standard indoor design conditions, as well as all other flow rates in this research.

To take into account the benefit of local recirculation in fan-powered variable-air-volume (VAV) systems, Ke and Mumma (1996) generalized the multiple spaces equation:

$$Y = \frac{X}{1 + X - F} \tag{1}$$

Nomenclature

- F = $(Z + S \cdot X)/(1 + S)$, fraction of total OA in primary supply air to satisfy critical zone;
- S = Mp_c/Ms_c, ratio of air drawn from the plenum to the primary air in the critical zone;
- X = Mo_n/Ms_n, the uncorrected ratio of OA to the total primary supply;
- Y = Mo_Y/Ms_n, the corrected ratio of OA to the total primary supply;
- Z = Mo_c/Ms_c, the ratio of required OA to primary air into the critical zone;
- Mo_c = OA flow rate the critical zone needs;
- Mo_n = $\sum_{i=1}^{n} Mo_i$, sum of the OA flow rates each zone needs;
- Mo_i = the OA flow rate zone i needs;
- Mo_Y = total OA flow rate corrected to account for recirculation;
- Mp_c = airflow rate recirculated from plenum directly to the critical zone i;
- Ms_c = primary airflow rate supplied to the critical zone;
- Ms_i = primary air entering zone i;
- Ms_n = $\sum_{i=1}^{n} Ms_i$, total primary airflow rate.

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TABLE 1
Design Airflow Rates in Control Strategies 1, 2, 7, and 8

Ct'l.	Floor	Ground		First		Second	
1	OA Ratio	0.69276		0.27186		0.40521	
		Outdoor Air	Primary Air	Outdoor Air	Primary Air	Outdoor Air	Primary Air
2	lbm/h	10645	15366	7524	27677	14035	34636
	gm/s	1341	1936	948	3487	1768	4364
7 and 8	lbm/h	9373	15949	7497	27314	12051	34749
	gm/s	1181	2010	945	3442	1518	4378

As the OA flow rate, Mo_i, for each room is determined according to Standard 62, the systematic total OA flow rate, Mo_Y, can be calculated from Equation 1. Applying Equation 1 ensures each space will have adequate unused OA with a minimum Mo_Y.

Based on the above generalized multiple spaces equation (GMSE), this paper first describes eight control strategies for OA flow rates in HVAC systems. The main components of the HVAC system of an existing library building were modeled. Then, all eight control strategies and the system model were programmed to perform annual simulations for Harrisburg, Pennsylvania. Finally, comparisons were made among different strategies.

VENTILATION CONTROL STRATEGIES

Several control methods are used by HVAC engineers to determine the quantity of minimum OA drawn into an air-handling unit (AHU). These methods may or may not satisfy ventilation requirements. To attempt to achieve ventilation requirements with minimum energy, eight control strategies for VAV systems have been proposed. They may be grouped into two categories: with and without local tempering capacities for the first and last four strategies, respectively. A brief description of eight common control strategies follows.

1. Fixed Minimum Outdoor Air Percentage

This control strategy without summer tempering employs a fixed ratio, Y value, of OA directly from the outside to the total primary supply air except when the economizer mode is activated, which usually draws more OA flow than the minimum up to 100%. This strategy is commonly achieved by a fixed minimum position OA damper. Literature has reported that this scheme results in OA deficiency during off-design conditions (Mumma and Wong 1990). The fixed minimum percentage used in the simulations was selected to be the Y value determined by the GMSE at 15:00 on July 21 (Table 1). This was assumed to be the hour of peak cooling load with the peak primary airflow rate. From Equation 1, the peak Mo_n (sum of zonal OA requirements) alone cannot determine the maximum Y; neither can the peak Ms_n (sum of primary airflow rates). However, using the latter is a conventional procedure to determine the design percentage.

2. Fixed Minimum Outdoor Airflow Rate to Buildings Without Regard for Distribution

This control strategy without summer tempering maintains a constant induced OA flow rate independent of the primary airflow rates as long as the system is not in the economizer mode. These fixed amounts, listed in Table 1, were determined by using the GMSE at 15:00 on July 21. They are equal to the product of Y times the total primary airflow rates at that hour. It is possible, however, for the fixed minimum not to be met if the primary supply airflow rate falls below the fixed minimum OA setting. Furthermore, the fixed minimum OA settings at 15:00 on July 21 might not be the lowest throughout an entire year for the same reason stated in strategy 1.

3. Hourly Minimum Outdoor Air from the GMSE

This control strategy without summer tempering utilizes the GMSE to correct the minimum required OA quantity continually as long as the system is not in the economizer mode. That is, the minimum OA flow rate setting is reset according to the primary airflow rate dynamically. Similar to the problem in control 2, the primary airflow rates, which are determined by the thermal loads, cannot be adjusted without tempering. Therefore, a zone might be underventilated if its primary airflow rate is less than the required OA flow rate.

4. GMSE with Supply Air Temperature Reset

This control strategy without summer tempering is similar to control 3 except that it also allows supply air temperature (SAT) reset. The SAT is increased to the highest allowable bounded by either ductwork capacity or space humidity. Raising the SAT increases the chiller's coefficient of performance (COP) and increases the hours of operating in free cooling (economizer) mode (Ke and Mumma 1997b). Every time the SAT changes, the minimum OA is recalculated according to the GMSE. Similar to strategies 2 and 3, the flow rate of OA entering the systems cannot exceed that of the total primary airflow rate.

5. GMSE with Primary Airflow Optimization

It may be observed in Equation 1 that by lowering the critical zone's F value, the system OA quantity can be reduced. Ke and Mumma (1997a) proved that the critical zone's F value can be lowered by increasing the primary airflow rate to the critical zone. This control trades off the reduced OA load on the chiller with fan, tempering, and zone cooling energy increases (Mumma and Bolin 1994). This conflict creates an ideal opportunity for optimization, or energy minimization. Since the objective of optimization is to minimize the total energy consumption, the objective function should include at least the main energy items: cooling (chiller), heating, humidification, tempering, and fan.

The objective function follows Ke and Mumma's algorithm (1997a) to redistribute the primary airflow rate (Ms_i) in the GMSE by using tempering. The goal is to reduce the energy used to condition the OA as well as the overall energy used in the system. Another benefit of using tempering is to satisfy the ventilation requirements completely all the time. When the thermal loads are too low to require sufficient primary airflow rates to carry adequate OA, the VAV box minimums are reset to deliver more primary air so that the ventilation requirements are achieved. Meanwhile, tempering of the primary air is used to prevent overcooling. The optimum is achieved when the total power demand has been minimized, with or without tempering, and the ventilation just satisfied. Since the OA requirements and thermal loads change dynamically, the optimizing algorithm must execute continually with time.

6. GMSE with Combined Primary Airflow and SAT Reset Optimization

Ke and Mumma (1997b) showed that increasing the SAT to the highest allowable value might globally consume more energy than no reset, but an optimized SAT saves energy under many conditions. Therefore, in addition to manipulating the primary airflow rate in control 5, this strategy also allows SAT reset. Unlike conventional SAT reset controls, which raise the SAT to its bounded limits, e.g., control 4, this strategy increases the SAT only when the system energy consumption will be reduced or minimized. Consequently, this strategy was expected to have benefits of both optimized variables, the primary airflow rate and SAT.

7. GMSE with Outdoor Air Limits (*F*-Limit)

Stanke (1993) presented a Z-limit method to determine the OA flow rate at design conditions for sizing smaller AHU coils in shut-off VAV systems with local reheat. This method ensures ventilation sufficiency if the pre-set OA capacity is large enough. As shown in Figure 1, an AHU should operate on the solid curve or below. If the thermal load-based operational point is at *A*, or any point above the horizontal line of the fixed upper OA limit, this control strategy will move it toward a point (*B* in Figure 1) on the fixed upper OA limit line by increasing the primary airflow rate of the critical space and tempering. If the thermal load-based operational point is at *C*, where the primary airflow rate is less than the OA flow limit, the primary airflow rate is reset to equal the required OA flow rate, i.e., the system operates on the $Z = 1$ line. Since the Z value dominates the required OA flow rate in the multiple spaces equation, it is the appropriate variable use when determining the design OA flow rates for shut-off VAV systems. However, it is invalid for fan-powered VAV systems. To implement Stanke's concept in the GMSE, the parameter F must be used to determine the OA flow rates instead of Z. Whether Z or F is used, the final goal of Stanke's concept is to limit the OA flow rates. The concept is implemented in this strategy.

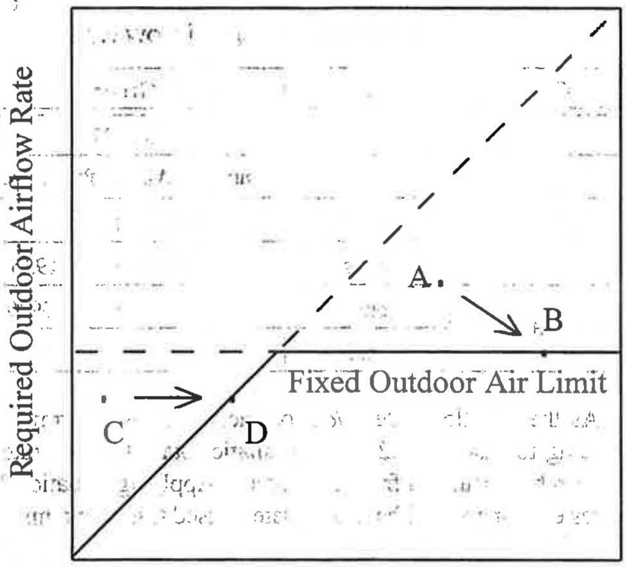


Figure 1 *F- or Z-limit method.*

This control is basically a much improved version of control 2 since it allows the system to operate below the OA flow limit. Because the OA quantity required at the design conditions greatly exceeds that required at other times, control 2 uses excessive energy. In addition, the AHU employing control 2 must be equipped with very large heating and cooling coils to handle the large quantities of OA if there is no tempering. Thus, it might be a good idea to limit the OA flow rates within prescribed limits. The limits were selected to be the optimum values at design hours, i.e., the OA flow rates of control 5 at 15:00 of July 21, as listed in Table 1. If the required OA is higher than the limit, the primary airflow rate must be increased and redistributed appropriately to induce the proper quantity of OA.

8. GMSE with *F*-Limit and Optimized SAT Reset

This control is similar to control 7 except that it also allows optimized SAT reset as discussed in control 6. This control strategy used the same fixed OA limits as control 7 (Table 1). The SAT is the only variable in the objective function of the optimization to minimize the total energy consumption.

SYSTEM MODELING

Descriptions of the Test Building

The test building is a college library located in south central Pennsylvania, which has 13,260, 14,650, and 14,670 ft^2 (1,128, 1,357, and 1,359 m^2) for the ground, first, and second floors, respectively. The building's HVAC system consists of three conventional variable-air-volume (VAV)

TABLE 2
Comparison of Total Energy Consumption of Different Control Strategies

Control	Ratios to the Maximum Total			Total MBtu/h (kW)		% Max
	Gnd Fl.	1st Fl.	2nd Fl.			
1	0.1947	0.2966	0.3765	58.85	(17.25)	86.78
2	0.2354	0.3039	0.4607	67.82	(19.88)	100.00
3	0.1990	0.2965	0.4102	61.42	(18.00)	90.57
4	0.1972	0.3359	0.4289	65.25	(19.12)	96.21
5	0.1814	0.2950	0.3880	58.62	(17.18)	86.44
6	0.1779	0.2847	0.3558	55.50	(16.27)	81.83
7	0.1978	0.3035	0.4066	61.57	(18.05)	90.79
8	0.1944	0.2932	0.3756	58.54	(17.16)	86.32

subsystems serving the three floors. The VAV boxes used throughout the building are series-type fan-powered VAV (SFPVAV) boxes.

The air-handling and cooling capacities of the three AHUs are 7,640, 11,700, and 12,980 cfm (3,605, 5,521, and 6,125 L/s) at 27.8, 42.5, and 52.1 tons, respectively. The VAV AHUs have variable-speed drives with tracking dampers and controls as required to measure and adjust the OA and recirculation airflow rates accordingly. The AHUs are equipped with temperature-based air-side economizers capable of handling up to 100% OA. The outdoor design conditions were 95°F (35°C) dry-bulb (DB) and 76°F (24.4°C) wet-bulb (WB) for cooling and 0°F (−17.8°C) for heating. The indoor design conditions were 75°F (24.4°C) and 50% RH for cooling and 70°F (21.1°C) and 35% RH for heating.

In addition, 48 SFPVAV boxes (12, 15, and 21 boxes on the ground, first, and second floors, respectively) are used in the building. The box flow rates range from 140 to 1,980 cfm (66 to 934 L/s). The SFPVAV boxes have constant-speed fans operating continuously to circulate plenum air with primary air to the zones. In this building the ceiling space of each floor is used as a return air plenum.

Hourly Envelope Load Calculations

The thermal load for each zone served as the basis for most of the simulation and optimization work in this research project. It impacts not only the energy consumption but the quantity of primary air delivered, which is the vehicle for providing OA to the zones. This project used the Building Loads Analysis and System Thermodynamics (BLAST 1993) program with weather data of Harrisburg, Pennsylvania, to perform the hourly load calculations. BLAST was used to generate the zonal occupancy, temperatures, sensible and latent loads, outdoor dry-bulb temperature and humidity ratio, and return plenum temperatures for the system simulations. The ceiling spaces were treated as individual zones in the BLAST modeling so that the plenum temperatures were available.

Mechanical Components Modeling

To perform the simulations and optimizations, each of the components that influence the objective equation, i.e., cooling (chiller), heating and tempering, humidification, and fan energies, must be represented by a computer model. The fan performance was modeled by a second-order polynomial based on manufacturers' cut sheets. Components (such as water pumps and cooling towers) that experience no significant changes in energy consumption under different control strategies are excluded from the simulations. The objective function is the equivalent hourly electricity demand. To set up the objective function, all nonelectrical energy consumption, such as steam heat for the heating and tempering coils and humidifiers, had to be converted to the cost equivalent of electricity. According to a physical plant's report (PSFEI 1995), the costs of electricity and gas were \$16.22 and \$3.43/MMBtu, respectively. Assuming all heat for the system is from gas, the conversion factor of heat to electricity used in this research was 0.21.

RESULTS OF SIMULATIONS

In this paper it is assumed that the OA controls operated with perfect accuracy and response for all eight strategies, i.e., the assigned OA flow rates were drawn without deficiency. The results of the hourly floor-by-floor computer-simulated control systems for the test building are summarized in Table 2. To make an objective comparison, all averages were based on occupied periods (4,872 hrs/yr), except the average of corrected OA flow rates, Mo_Y. The average corrected OA flow rates were averaged over the hours of minimum OA operation, a number that was different for each control approach. From Table 3, the observations discussed below can be deduced.

Total Primary Air Mass Flow Rates, *Ms*

The average values of the first three controls are identical because no tempering and no SAT reset was used. The primary airflow rates for those cases were determined by the thermal loads only. Comparing control 4 with control 3 (with and without SAT reset, respectively), SAT reset results in higher primary

TABLE 3a
Annual Hourly Simulation Summary (English Units)

	Average *Ms*, lbm/h			Average Mo_Y, lbm/h			Average *Qoa*, MBtu/h			
Ct'l	Gnd. Fl.	1st Fl.	2nd Fl.	Gnd. Fl.	1st Fl.	2nd Fl.	Gnd. Fl.	1st Fl.	2nd Fl.	Ct'l
1	11212.0	19204.7	17781.2	8706.9	6914.6	7803.7	15.854	22.665	24.493	1
2	11212.0	19204.7	17781.2	10424.2	7659.6	12777.6	17.300	23.647	26.614	2
3	11212.0	19204.7	17781.2	8652.6	6214.9	10457.0	14.942	21.944	23.479	3
4	11785.3	25903.4	28864.2	8706.1	6250.1	9272.2	8.318	8.740	10.422	4
5	11579.6	19258.2	18458.8	7607.7	5634.8	9143.0	8.678	10.889	13.491	5
6	12004.7	20413.2	21743.8	7624.9	5634.8	7793.2	8.162	9.198	10.554	6
7	11541.3	19496.5	17864.7	7724.8	5780.8	9971.7	8.502	10.939	13.705	7
8	11973.5	20643.4	21326.3	7724.4	5779.7	8945.7	7.963	9.243	10.719	8
	Average *Fbhp*, hp			Average *Qcc*, MBtu/h			Average *Qhc*, MBtu/h			
Ct'l	Gnd. Fl.	1st Fl.	2nd Fl.	Gnd. Fl.	1st Fl.	2nd Fl.	Gnd. Fl.	1st Fl.	2nd Fl.	Ct'l
1	0.8182	1.4538	1.1537	44.023	71.190	81.146	5.147	0.000	0.829	1
2	0.8182	1.4538	1.1537	45.525	72.184	83.292	14.251	1.017	19.918	2
3	0.8182	1.4538	1.1537	43.104	70.471	80.133	7.026	0.523	10.084	3
4	0.8481	3.1492	3.0084	41.779	62.349	67.285	7.304	0.615	20.337	4
5	0.8409	1.4814	1.1756	43.022	70.585	79.653	0.436	0.000	1.941	5
6	0.8588	1.7259	1.4956	41.717	64.416	69.733	0.436	0.000	4.099	6
7	0.8464	1.5007	1.1632	43.677	70.745	79.934	4.362	0.097	8.928	7
8	0.8666	1.7432	1.4778	42.361	64.548	69.993	4.366	0.102	13.785	8
	Average *Qhumd*, MBtu/h			Average *Qtemp*, MBtu/h			Power Consumption, MBtu/h			
Ct'l	Gnd. Fl.	1st Fl.	2nd Fl.	Gnd. Fl.	1st Fl.	2nd Fl.	Gnd. Fl.	1st Fl.	2nd Fl.	Ct'l
1	3.473	6.139	2.778	0.076	0.464	22.441	13.202	20.115	25.536	1
2	6.014	6.479	8.716	0.076	0.464	22.441	15.964	20.610	31.245	2
3	3.920	6.322	5.400	0.076	0.464	22.441	13.496	20.111	27.817	3
4	4.063	6.758	5.241	0.054	0.265	8.850	13.376	22.781	29.091	4
5	2.610	6.186	3.718	2.101	0.766	25.348	12.303	20.006	26.316	5
6	2.606	6.175	2.831	2.060	0.752	19.792	12.063	19.305	24.131	6
7	3.408	6.503	5.058	1.937	2.120	22.872	13.413	20.585	27.574	7
8	3.404	6.493	4.256	1.918	2.084	14.992	13.182	19.885	25.473	8
	Avg. Primary Air [CO_2], ppm			Avg. Plenum [CO_2], ppm			Max. Zonal [CO_2], ppm			
Ct'l	Gnd. Fl.	1st Fl.	2nd Fl.	Gnd. Fl.	1st Fl.	2nd Fl.	Gnd. Fl.	1st Fl.	2nd Fl.	Ct'l
1	377.54	432.65	451.47	678.40	607.93	701.94	1488.95	1319.19	2138.27	1
2	321.61	413.65	326.20	622.46	588.93	576.67	1222.04	1159.87	1255.68	2
3	384.03	442.93	384.08	684.89	618.21	634.55	1222.04	1000.00	1255.68	3
4	393.29	477.19	489.76	678.15	607.73	636.41	1112.62	1000.00	1003.22	4
5	422.35	447.15	435.47	711.98	621.82	667.89	1000.00	1000.00	974.39	5
6	428.52	448.72	485.13	707.26	615.13	681.29	1000.00	1000.00	974.39	6
7	403.37	444.23	392.10	695.20	616.31	640.52	1000.00	1000.77	978.14	7
8	409.86	445.73	441.05	690.73	609.66	647.39	1000.00	1000.77	978.14	8

TABLE 3a (Continued)
Annual Hourly Simulation Summary (English Units)

Ct'l	Average *Ts*, °F			Remarks	Ct'l
	Gnd. Fl.	1st Fl.	2nd Fl.	Descriptions	
1	53.00	53.00	53.00	Fixed Outdoor Air Percentage	1
2	53.00	53.00	53.00	Fixed Outdoor Airflow Rate	2
3	53.00	53.00	53.00	Generalized Multiple-Spaces Equation (GMSE)	3
4	54.27	58.68	62.95	GMSE with SAT Reset	4
5	53.00	53.00	53.00	GMSE with Optimized Primary Airflow Rate	5
6	53.97	54.18	57.43	GMSE with Optimized Primary Airflow Rate and SAT	6
7	53.00	53.00	53.00	GMSE with Outdoor Airflow Limit (*F*-Limit)	7
8	53.97	54.18	58.28	GMSE with *F*-Limit and Optimized SAT	8

TABLE 3b
Annual Hourly Simulation Summary (SI Units)

Ct'l	Average *Ms*, gm/s			Average *Mo*, gm/s			Average *Qoa*, kW			Ct'l
	Gnd. Fl.	**1st Fl.**	**2nd Fl.**	**Gnd. Fl.**	**1st Fl.**	**2nd Fl.**	**Gnd. Fl.**	**1st Fl.**	**2nd Fl.**	
1	1412.72	2419.80	2240.43	1097.07	871.25	983.26	4.6465	6.6429	7.1785	1
2	1412.72	2419.80	2240.43	1313.45	965.11	1609.98	5.0703	6.9306	7.8000	2
3	1412.72	2419.80	2240.43	1090.23	783.07	1317.58	4.3793	6.4313	6.8812	3
4	1484.95	3263.82	3636.89	1096.97	787.52	1168.30	2.4378	2.5616	3.0546	4
5	1459.04	2426.54	2325.81	958.57	709.99	1152.02	2.5433	3.1914	3.9541	5
6	1512.59	2572.07	2739.72	960.74	709.99	981.94	2.3921	2.6957	3.0931	6
7	1454.20	2456.56	2250.95	973.32	728.38	1256.44	2.4919	3.2061	4.0168	7
8	1508.66	2601.07	2687.11	973.28	728.24	1127.16	2.3338	2.7089	3.1416	8
	Average *Fbhp*, hp			**Average *Qcc*, kW**			**Average *Qhc*, kW**			
Ct'l	**Gnd. Fl.**	**1st Fl.**	**2nd Fl.**	**Gnd. Fl.**	**1st Fl.**	**2nd Fl.**	**Gnd. Fl.**	**1st Fl.**	**2nd Fl.**	**Ct'l**
1	0.8182	1.4538	1.1537	12.9023	20.8647	23.7825	1.5085	0.0000	0.2428	1
2	0.8182	1.4538	1.1537	13.3427	21.1558	24.4114	4.1768	0.2980	5.8375	2
3	0.8182	1.4538	1.1537	12.6330	20.6538	23.4856	2.0592	0.1533	2.9554	3
4	0.8481	3.1492	3.0084	12.2447	18.2735	19.7202	2.1408	0.1801	5.9605	4
5	0.8409	1.4814	1.1756	12.6091	20.6874	23.3449	0.1276	0.0000	0.5690	5
6	0.8588	1.7259	1.4956	12.2266	18.8794	20.4377	0.1276	0.0000	1.2012	6
7	0.8464	1.5007	1.1632	12.8010	20.7342	23.4272	1.2786	0.0284	2.6167	7
8	0.8666	1.7432	1.4778	12.4152	18.9181	20.5137	1.2795	0.0298	4.0401	8
	Average *Qhumd*, kW			**Average *Qtemp*, kW**			**Power Consumption, kW**			
Ct'l	**Gnd. Fl.**	**1st Fl.**	**2nd Fl.**	**Gnd. Fl.**	**1st Fl.**	**2nd Fl.**	**Gnd. Fl.**	**1st Fl.**	**2nd Fl.**	**Ct'l**
1	1.0178	1.7992	0.8143	0.0222	0.1361	6.5771	3.8692	5.8954	7.4843	1
2	1.7627	1.8989	2.5546	0.0222	0.1361	6.5771	4.6789	6.0404	9.1573	2
3	1.1487	1.8528	1.5826	0.0222	0.1361	6.5771	3.9555	5.8943	8.1526	3
4	1.1908	1.9805	1.5361	0.0159	0.0777	2.5937	3.9204	6.6768	8.5260	4

TABLE 3b (Continued)
Annual Hourly Simulation Summary (SI Units)

5	0.7650	1.8129	1.0896	0.6156	0.2245	7.4290	3.6057	5.8634	7.7127	5
6	0.7637	1.8097	0.8298	0.6037	0.2205	5.8007	3.5356	5.6579	7.0724	6
7	0.9989	1.9060	1.4823	0.5676	0.6213	6.7033	3.9311	6.0331	8.0816	7
8	0.9977	1.9029	1.2474	0.5621	0.6106	4.3939	3.8635	5.8281	7.4659	8
	Avg. Primary Air [CO_2], ppm			**Avg. Plenum [CO_2], ppm**			**Max. Zonal [CO_2], ppm**			
Ct'l	**Gnd. Fl.**	**1st Fl.**	**2nd Fl.**	**Gnd. Fl.**	**1st Fl.**	**2nd Fl.**	**Gnd. Fl.**	**1st Fl.**	**2nd Fl.**	**Ct'l**
1	377.54	432.65	451.47	678.40	607.93	701.94	1488.95	1319.19	2138.27	1
2	321.61	413.65	326.20	622.46	588.93	576.67	1222.04	1159.87	1255.68	2
3	384.03	442.93	384.08	684.89	618.21	634.55	1222.04	1000.00	1255.68	3
4	393.29	477.19	489.76	678.15	607.73	636.41	1112.62	1000.00	1003.22	4
5	422.35	447.15	435.47	711.98	621.82	667.89	1000.00	1000.00	974.39	5
6	428.52	448.72	485.13	707.26	615.13	681.29	1000.00	1000.00	974.39	6
7	403.37	444.23	392.10	695.20	616.31	640.52	1000.00	1000.77	978.14	7
8	409.86	445.73	441.05	690.73	609.66	647.39	1000.00	1000.77	978.14	8
	Average *Ts*, °C			**Remarks**						
Ct'l	**Gnd. Fl.**	**1st Fl.**	**2nd Fl.**	**Descriptions**						**Ct'l**
1	11.67	11.67	11.67	Fixed Outdoor Air Percentage						1
2	11.67	11.67	11.67	Fixed Outdoor Airflow Rate						2
3	11.67	11.67	11.67	Generalized Multiple-Spaces Equation (GMSE)						3
4	12.37	14.82	17.20	GMSE with SAT Reset						4
5	11.67	11.67	11.67	GMSE with Optimized Primary Airflow Rate						5
6	12.20	12.32	14.13	GMSE with Optimized Primary Airflow Rate and SAT						6
7	11.67	11.67	11.67	GMSE with Outdoor Airflow Limit (*F*-Limit)						7
8	12.20	12.32	14.60	GMSE with *F*-Limit and Optimized SAT						8

airflow rates. Similar patterns appear when controls 6 and 8 are compared with 5 and 7, respectively. Control 5, which has no SAT reset, has a higher average than control 3 because it minimizes the total energy consumption by increasing primary airflow rates to reduce OA loads. Control 7 also has higher primary airflow rates than control 2 because control 7 uses tempering to increase the primary airflow rates as needed to deliver the required OA when the thermal loads are low.

Total Corrected Outdoor Air Mass Flow Rates, Mo_X

This item in Table 3 lists the annual averages of actual OA flow rates entering the AHUs in the minimum OA mode, which means the OA is equal to or less than the amount required to comply with the GMSE. Depending on the control strategy, a system might be in the economizer mode from a temperature standpoint but be unable to deliver the required minimum OA flow rate, i.e., the total primary airflow rate is less than the minimum OA flow rate. In such cases the system would be placed in the minimum OA mode and summarized accordingly. Basically, control 2 has the highest OA flow rates among the eight strategies because it has fixed minimum OA settings. Control 5 has a lower average than control 3 because it minimizes the total energy consumption by increasing primary airflow rates to reduce OA loads.

Cooling Coil Outdoor Air Loads, Qoa

The OA loads indicate the energy needed to cool and dehumidify OA to the conditions of return air if the cooling coils are on. Accordingly, they are indices of the total OA entering the systems when the chiller is on. As expected, adding tempering and primary airflow optimization (control 5) results in much lower OA loads than a scheme without optimization (control 3). In addition, the SAT reset reduces the OA loads because it increases the potential for free cooling.

Fan Power, *Fbhp*

Fan power presented here refers to that of the AHUs. The SFPVAV box fans are assumed to draw constant power regardless of controls and are not included in the numbers. AHU fan power is a function of the total pressure drop across the systems and volumetric airflow rates. The first three control strategies have exactly the same fan brake horsepower because they are supplying identical primary airflow rates. Similarly, the fan power for the other control strategies follows the same trends as the primary airflow rates. In addition, a lower bound of 0.75 hp on the manufacture cut sheet curve fit data was set for each of the three air handler fans. Since the averages of the first floor are all below 0.86 hp, only 15% higher than the minimum, this implies that the fan system is often operating at minimum power.

Cooling Coil Loads, *Qcc*

The cooling coil loads include both the building thermal loads and OA loads. If tempering is used during the cooling mode to satisfy ventilation requirements, the tempering heat also becomes a part of the cooling coil loads. There is no SAT reset or forced local tempering (i.e., no cooling primary airflow minimum settings) in the first three control strategies. The differences in cooling coil loads among them are mainly from the difference in OA flow rates during cooling. The SAT reset decreases the required local tempering, thus usually reducing the cooling coil loads (Ke and Mumma 1997b). That is why the control strategies with SAT reset have lower cooling coil loads than their counterparts.

Heating Coil Loads, *Qhc*

The first floor has the lowest heating coil loads for all eight control strategies because it is the middle level of the building and has the least exposure to outside. In addition, since higher SATs result in higher heating coil loads but no SAT reset is used in the first three control strategies, the differences in heating coil loads among them can be accounted for by the various OA flow rates during the heating season. As expected, because winter tempering is shifted to the heating coils, controls 4, 6, and 8 with SAT reset have higher heating coil loads than their counterparts, controls 3, 5, and 7, respectively.

Humidification Loads, *Qhumd*

The humidifier is interlocked with the cooling coils, i.e., it is off when the cooling coils are on. Therefore, most humidification occurs in the heating seasons. Basically, this item follows trends similar to those of the heating coil loads.

Tempering Loads, *Qtemp*

The building automation system was designed to set any of the SFPVAV boxes to their minimum primary airflow setpoint when a zone needs heating. The numbers in the rows of the first three control strategies in this item in Table 3 merely reflect the total zonal heating loads of each system. If a zone calls for heating, SAT reset can reduce the tempering energy since more heating comes from the AHUs. This is why control 4 has lower tempering energy than control 3, control 6 lower energy than control 5, and control 8 lower energy than control 7. In addition, the difference is more significant on the second (top) floor than on others because its roof exposure causes the heating requirement to be much higher.

Power Consumption

The total power consumption is the summation of the AHU fan power, cooling and heating coil loads, tempering loads, humidification loads, and chiller power. All loads are converted to equivalent electricity power in Btu/h (kW). The total equivalent electricity power consumption rate constituted the objective function in the optimizing strategies. The item in Table 3 and Figure 2 illustrates that controls 2 and 6 have the highest and lowest power consumption rates, respectively. Control 6 consumes 81.8% of the energy of control 2 (Table 2). Note that the energy difference would have been even greater if tempering were used with control 2 to ensure more adequate ventilation. In general, the SAT reset will reduce the chiller energy consumption but will increase fan energy use and reduce dehumidification capacities. When the systems are operating near the minimum flow rates, the highest SAT reset might be beneficial. Otherwise, the increased fan energy puts SAT at a disadvantage. For example, on the first and second floors, using the control 4, non-optimized SAT reset, there is an energy consumption penalty. From an energy perspective, control 5 and control 8 are about equal among the group with tempering because they optimize different variables—one the total primary airflow rate and the other the SAT. Control 6 is clearly the best because it optimizes both SAT and ventilation airflow/distribution.

Supply Air CO_2 Concentrations

A good ventilation control strategy must not only consume as little energy as possible but satisfy indoor air quality. To compare air freshness among different control strategies, the CO_2 concentration was used as an indicator. Since the transient effects and ventilation effectiveness are beyond the scope of this paper, a steady-state and well-mixed model was assumed. So modeled, the CO_2 concentration at any point in the system was calculated from the mass balance. In system simulations, the OA CO_2 concentration was assumed to be 300 ppm. With assumptions of 1.2 met metabolic rate and an OA requirement of 15 cfm (7.1 L/s) per person, the saturated CO_2 concentration is 1,000 ppm for the ground and first floors. On the second floor, a conference room required 20 cfm (9.4 L/s) per person. When the conference room is critical, its saturated CO_2 concentration is 858 ppm. A surrounding space that requires 15 cfm/person (saturated CO_2 concentration of 1,000 ppm) has a coincident maximum CO_2 concentration of 974.39 ppm. This space is not the critical one, even though its CO_2 concentration exceeds that of the conference room. Carbon dioxide concentration near but not over the saturated values is desirable from an energy minimization perspective. Further, the more uniform the CO_2 concentra-

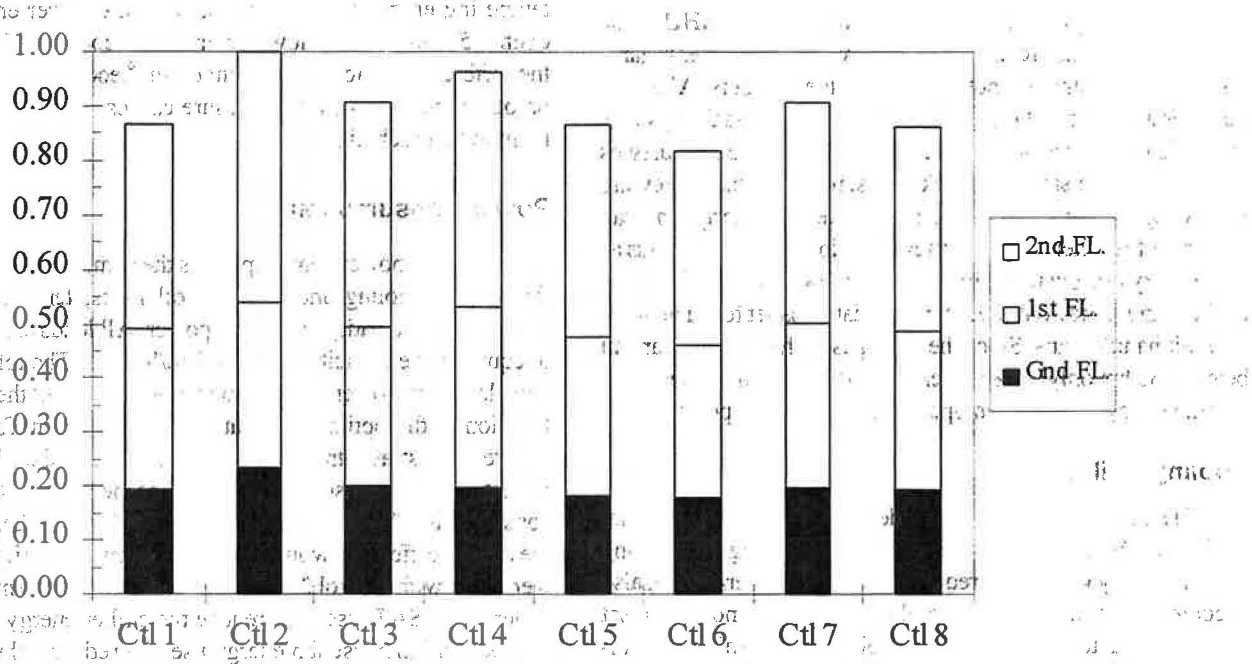


Figure 2 Relative annual total energy consumption.

tions, the better the ventilation air is being distributed and energy use minimized.

Among all control strategies, control 2 has the lowest supply CO_2 concentrations at a considerable energy penalty. This implies control 2 has the highest average OA percentage in the primary air. However, a high percentage does not guarantee good zonal air quality, which depends on both quantities and distributions. Controls 5 and 6 use tempering to manage the distribution of OA, so they tend to have higher primary air CO_2 concentrations. They also are able to keep the zone concentrations just at saturation, thus providing considerable energy savings. Controls 7 and 8 also employ tempering, but they have maximum OA settings, which pull down the average CO_2 concentrations.

SAT reset increases the primary airflow and may decrease the OA flow rates in SFPVAV systems if the ventilation requirements are satisfied as explained previously. This means the recirculation airflow rate at the AHU increases with SAT. Since the return air CO_2 concentrations are always greater than the outdoor ones, it is no surprise that the supply CO_2 concentrations with SAT reset are higher than those without reset because of higher recirculation rates.

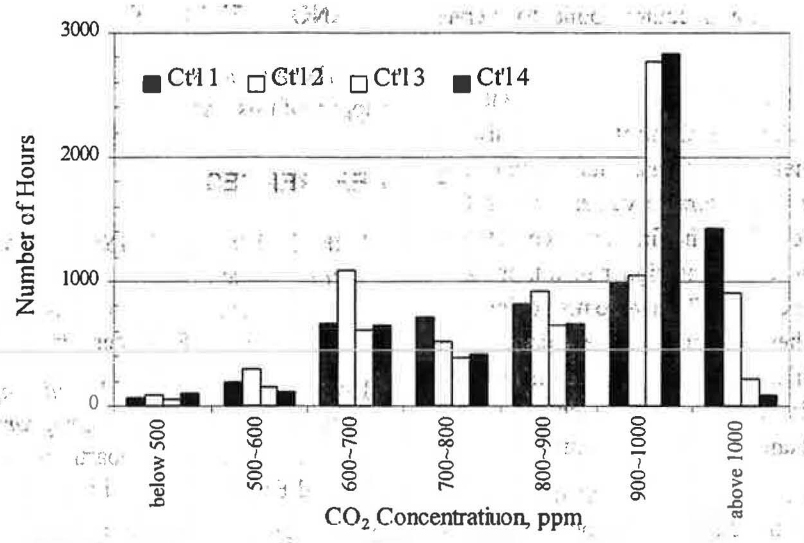
Plenum CO_2 Concentrations

Plenum CO_2 concentrations are the same as those of recirculation and exhaust air due to the well-mixed assumption. They are also the "averages" of zonal CO_2 concentrations because plenum air is the mixture of all zonal return air. Lower plenum CO_2 concentrations mean more unvitiated but conditioned air exhausted, i.e., more energy wasted. Similar to the supply air CO_2 concentrations, control 2 has the lowest one. This does not mean it has the best air quality. Conversely, from an energy minimization point of view, the higher the CO_2 concentrations the better, as long as they are below the saturation values that would result when the specified OA flow rate per person is supplied.

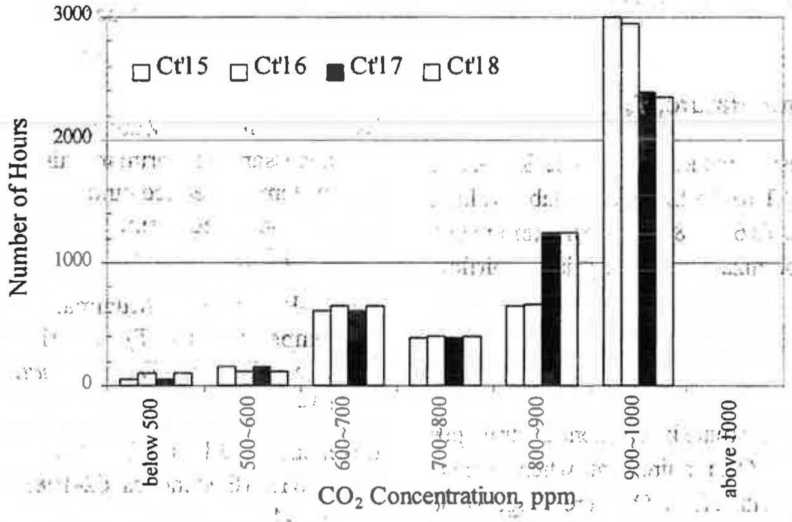
Maximum Zonal CO_2 Concentrations

This item in Table 3 displays the highest CO_2 concentrations for each system in a year. The values for each floor do not necessarily belong to the same zone from control strategy to control strategy. A good control strategy must not allow the CO_2 concentration to exceed the saturated limit of 1,000 ppm. Note that the conference room on the second floor requires 20 cfm (9.4 L/s) per person, which makes the saturated CO_2 concentration equal to 974.39 ppm. Figure 3 shows comparisons of CO_2 bins among different control strategies for the ground floor. This figure displays the frequency of hourly maximum zonal CO_2 concentrations. The hourly maximum zonal CO_2 concentration is the CO_2 concentration of the critical zone(s) since all zones have the same OA requirement per person. Different zones at different hours may exhibit the maximum CO_2 concentration.

As expected, the first four controls without summer tempering cannot comply with the ventilation requirements. Control 1 has the worst air qualities among the four strategies on all three floors. Most of the more than 1,000-ppm hours occur during mild seasons when the cooling loads are extremely low. The



(a) Control Strategies 1 to 4



(b) Control Strategies 5 to 8

***Figure 3** Frequency distribution of CO_2 concentrations for the ground floor.*

SFPVAV boxes in the HVAC system do not have minimum settings for cooling. Therefore, the primary airflow rates may go down extremely low, even to zero. Control 1 brings in the OA at fixed percentages. The OA flow rate is low when the primary airflow rate is low, even with a high OA percentage. Fortunately, SFPVAV boxes recirculate return air from the plenums, which increases as the primary airflow rate is decreasing. If shut-off boxes were employed instead of SFPVAV ones, the maximum zonal CO_2 concentration might be far higher. Unlike control 1, control 2 has fixed OA flow rates. Although it provides better air quality than control 1 under the same tough conditions, the fan systems are not allowed to bring in the setpoint OA flow rates when the cooling loads are low since the spaces would be over-cooled. Even when the OA setpoint is maintained, it does not guarantee that enough OA is distributed to the critical zones. In addition, control 2 consumes more energy to condition the OA than control 1, especially when the actual required OA flow rate is low. On the other hand, controls 3 and 4 had been expected to satisfy the ventilation requirements before this research started since they applied the GMSE to determine the required OA. The Max. Zonal [CO_2] in Table 3 and Figure 3 verify that they do have times of failure. Failure occurs because the primary air, which could not bring in enough OA at low thermal loads, is determined by the thermal loads. This defect can be eliminated with tempering, a situation that could require a boiler to operate during the off season.

All of the last four controls with tempering comply with the ventilation requirement much better. Controls 5 and 6 always satisfy the GMSE, and their annual maximum zonal CO_2 concentrations never exceed the saturated values, i.e., no OA deficiency. In controls 7 and 8, both the first and second floors have four hours of CO_2 concentrations higher than their saturated values. This is because controls 7 and 8 do not achieve the ventilation requirements when the critical zone's primary air reaches maximum but is still not enough to reduce the required OA flow rates to the setpoint. In other words, higher OA flow rate settings should be used under the *F*-limit control.

Besides the OA deficiency during the cooling mode, a situation that does not happen in these fan systems but needs some attention is the minimum setting of VAV boxes for the heating modes. It is necessary that the minimum setting for heating be large enough to deliver adequate OA, since the primary airflow rates are fixed. Otherwise, the OA deficiency could occur in the heating mode.

Average Supply Air Temperature, T_s

Control 4 has the highest T_s values of the three SAT-reset controls since it reset the SAT to the highest available values. The difference between controls 6 and 8 is due to maximum OA flow rate settings and no optimization for the primary airflow rates in the latter.

CONCLUSIONS

Without tempering at the terminal boxes, control strategies 1 through 4 could not satisfy the OA requirements when the thermal loads are low. Control 1, with a fixed OA percentage, is the worst control strategy from a ventilation perspective. Controls 5 through 8 comply with the ventilation requirements completely. Controls 7 and 8 can satisfy the ventilation requirements if the OA capacity of coils is selected properly. This could be a challenge at the design stage if the peak thermal load and ventilation requirement did not occur at the same time, and an error could result in undersized coils.

From an economic perspective, controls 5 and 6 consume the least energy, have the lowest demand, and always meet Standard 62 guidelines. Controls 7 and 8 utilize more energy than controls 5 and 6 due to the fixed OA flow rate settings but may reduce the first cost of coils. The optimized SAT reset always had a beneficial effect on energy use, while the highest SAT reset is not beneficial from an energy perspective and is not recommended. In sum, control 6 (which employed both optimized primary airflow rate and SAT) is the most economical control strategy and control 2 is the worst.

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