Saving Money with Real-Time VAV Control Optimization

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Outdoor airflow into a building is necessary for occupant health and comfort, but conditioning it requires large amounts of energy and makes overventilation an expensive problem. However, in variable air volume (VAV) systems a way exists to reduce this cost by increasing the efficiency of how the ventilation air is distributed. The secret is to oversupply discharge air to certain zones, despite the reheat penalty incurred by doing so. This counterintuitive notion reveals a trade-off in how energy can be consumed in different parts of the system. Advanced control systems can take advantage of this dynamic to optimize system setpoints in real time, yielding significant energy cost savings.

The optimal control scheme in this article was designed for single-duct VAV systems with terminal reheat, and it is applicable to systems both with and without demand-controlled ventilation. It may be suitable for other types of VAV systems, though these have not yet been investigated.

The Core Trade-Off of HVAC

According to the acronym itself, HVAC systems support heating, ventilation and air conditioning. In terms of energy efficiency, though, it is the "and" part that is one of their biggest drawbacks. Trying to satisfy all three of these goals at the same time with a single airstream necessarily creates inefficiencies, such as overventilation and simultaneous heating and cooling. Eliminating these inefficiencies turns out to be a trade-off of energy costs versus capital and maintenance costs. More-efficient designs are typically more complex and thus more expensive to purchase and maintain. For instance, dedicated outdoor air systems attempt to decouple ventilation and heating/cooling requirements, but they do so by duplicating components for each purpose. On the other hand, simple designs like constant air volume systems are cheaper and easier to install and maintain, but they can be energy hogs and therefore are generally not allowed by energy efficiency codes. On both of these

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measures, variable air volume systems are somewhere in the middle. Because of their relative simplicity and good energy performance, VAV systems are a popular choice for applications that have many zones, such as offices and academic buildings.

VAV Drawbacks

VAV systems have a couple of limitations, however. One is that a single air handler is tasked with ventilating multiple zones within the building. This is problematic because not all zones are created equal. Some zones, like lecture halls, have large floor areas and high occupancies; thus, they have high ventilation requirements when in use, while others may be small single-person offices. Since the outdoor air is evenly mixed into the supply air, though, a VAV system cannot direct that ventilation precisely where it needs to go. Inevitably, to meet the requirements of zones that need a higher fraction of outdoor air, zones with lower relative requirements may become overventilated, especially if they have high sensible loads. This means that more outdoor air than necessary is cooled and dehumidified, at considerable expense.

A second drawback is that the air handler can only provide supply air at a single temperature. This temperature must be low enough to wring moisture out of humid air. It also must be low enough to satisfy zones' positive sensible loads without exceeding their maximum designed discharge airflow rates. Cold supply air leads to another challenge, though. Some zones may have low or negative sensible loads. In that case, VAV terminal boxes can clamp down on how much air they deliver to avoid overcooling the zones. However, this is limited by ventilation considerations such that after a certain point, a minimum amount of airflow is provided, and the discharge air is reheated. (Volume is increased again while reheating with dual maximum logic.)[1]

This simultaneous cooling at the air handler and heating at the terminal boxes represent wasted dollars. One goal then for an existing VAV system should be to minimize the impact of these inherent inefficiencies by making improvements to its controls. For instance, supply air temperature reset attempts to address simultaneous heating and cooling by seeking a setpoint that balances both needs.[2] Addressing low ventilation efficiency, though, requires additional creativity. This will lead to a solution that can tackle both.

The Ventilation Calculation

Appendix A of ASHRAE Standard 62.1-2019[3] defines an alternative series of equations for determining how much outdoor air is required at the air handler, based on the ventilation requirements and discharge air volumes at each of the zones. It recognizes that outdoor air is not precisely distributed, and so some air returning from overventilated zones can be considered unused and credited toward the outdoor air requirement as it makes another pass through the system. The result is that the outdoor air volume requirement at the air handler is greater than the sum of the volumes needed by each zone individually, but the outdoor air fraction can be somewhat less than the fraction required by the most demanding zone.

Let's explore the equations to see both why this is, and how that requirement can be manipulated in accordance with the standard.

Each zone has a minimum outdoor airflow requirement, V_{oz}, based on parameters such as occupancy and floor area. The discharge airflow to each zone is V_{dz}, and dividing these two values gives the primary outdoor air fraction, Z_{pz} (referenced here simply as zone fraction):

$$Z_{pz} = V_{oz}/V_{dz}$$

If Z_{pz} is ever greater than one, this would mean that the zone cannot be properly ventilated, as even 100% outdoor air would not be enough to meet its requirement. So, V_{dz} must always be kept greater than or equal to V_{oz}. Among all the zones, the one with the highest Z_{pz} value is known as the critical zone, and we call its zone fraction Z_p. Next, summing up all the V_{oz} values

$$V_{ou} = \sum_{\text{all zones}} V_{oz}$$

gives the uncorrected outdoor intake volume, V_{ou}. If the system could perfectly distribute its ventilation air, this would be the total amount of outdoor air it would need, but since it cannot, the final corrected value will be higher. Likewise, summing all the discharge air values gives V_{ps}, the system primary airflow:

$$V_{ps} = \sum_{\text{all zones}} V_{dz}$$

This is the total amount of supply air flowing through the system. Dividing these latest two values gives the average outdoor air fraction:

$$X_s = V_{ou}/V_{ps}$$

Condensing things slightly, we now use the highest zone fraction value from earlier, that is, the one corresponding to the critical zone, to calculate E_v, the system's ventilation efficiency:

$$E_v = 1 + X_s - Z_p$$

This will be a value between zero and one. It represents how well the system is distributing the outdoor air that it is bringing in, with a value of one implying perfect ventilation efficiency. Finally, the required outdoor air intake volume is found by dividing the uncorrected value by the efficiency:

$$V_{ot} = V_{ou}/E_v$$

This is the minimum amount of outdoor air that must be introduced at the air handler according to the standard, though more than this amount is permissible, such as when economizing. This set of equations supports two use cases. First, system designers may use this calculation process with conservative assumptions for the purposes of system sizing, and second, it can be implemented into the control scheme to use real-time, zone-level data to provide ventilation in accordance with what is occurring within the building during operation. Such real-time reset of ventilation air is now required prescriptively by Standard 90.1-2019 for VAV systems.[4]

Minimizing Outdoor Air Requirements

Given that outdoor air is expensive to condition, it is desirable to minimize the volume required. Walking backward through the ventilation equations shows us how. To start, we can decrease V_{ot} by either lowering V_{ou}, or by raising E_v. The value of V_{ou} can be adjusted according to the principles of demand-controlled ventilation (DCV). This involves monitoring or estimating the population in the zones using equipment such as CO_2 or occupancy sensors (see RP-1547, for example[5]), which can allow ventilation requirements to be lowered, even to zero.[6,7] These principles are well-known, and adding such sensors has a rapid payback.

Instead, let us focus on increasing E_v to get it closer to a value of one, since this is applicable regardless of whether DCV is present in the system. We can do this by raising X_s or by reducing Z_p. First, X_s can be raised slightly by reducing the discharge airflow in any of the non-critical zones, since doing so will shrink its denominator without affecting the numerator. The second, better option is to reduce Z_p by increasing the discharge airflow in just the critical zone. Increasing the discharge airflow to the critical zone can dramatically reduce the outdoor air requirement. This fact is even called out in an informative note in Appendix A of the standard. To illustrate, *Table 1* shows an example for a simple system, where providing an additional 200 cfm (94.4 L/s) of discharge airflow to the critical zone results in a 131 cfm (61.8 L/s) drop in the outdoor air requirement. Readers are encouraged to recreate this example using numbers from their own systems.

TABLE 1 An increase in discharge airflow to the critical zone here results in a 15% reduction in the outdoor air requirement.

	INITIAL STATE			IMPROVED EFFICIENCY		
ZONE NAME	A	B	C	A	B	C
V_{oz}, MINIMUM VENTILATION (cfm)	200	300	200	200	300	200
V_{dz}, DISCHARGE AIRFLOW (cfm)	400	900	1,000	600	900	1,000
Z_{pz}, ZONE FRACTION	0.5	0.33	0.2	0.33	0.33	0.2
CRITICAL ZONE	A			A & B		
Z_p, ZONE FRACTION OF CRITICAL ZONE	0.5			0.33		
V_{ou}, UNCORRECTED OUTDOOR AIR REQUIREMENT (cfm)	700			700		
V_{ps}, SYSTEM PRIMARY AIRFLOW (cfm)	2,300			2,500		
X_s, AVERAGE OUTDOOR AIR FRACTION	0.30			0.28		
E_v, VENTILATION EFFICIENCY	80%			95%		
V_{ot}, OUTDOOR AIR REQUIREMENT (cfm)	870			739		

This example can be taken even further. For instance, once the zone fraction of Zone A has been reduced to 0.33, both Zones A and B together become the critical zones. To continue increasing the ventilation efficiency of the system then, they can have their discharge airflows increased simultaneously, such that their zone fractions drop at the same rate. Once they reach zone fractions of 0.2, Zones A and B will have discharge airflows of 1,000 cfm and 1,500 cfm (492 L/s and 708 L/s), respectively, and the system would have a ventilation efficiency of 100%. At this point, it would only require 700 cfm (330 L/s) of outdoor air.

Diminishing returns to this process exist, however. Going from the state of improved efficiency to one of

FIGURE 1 Diminishing returns of increases in discharge air to the critical zone(s).

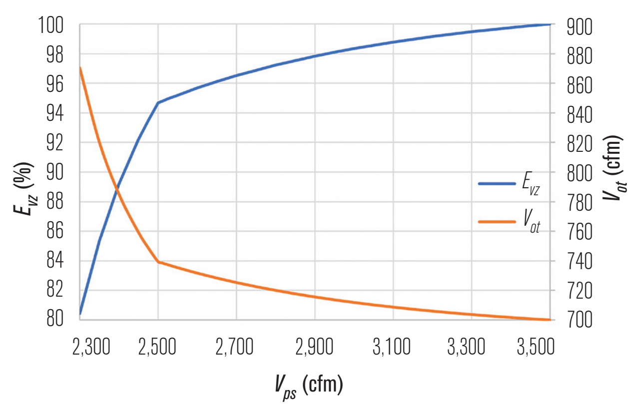
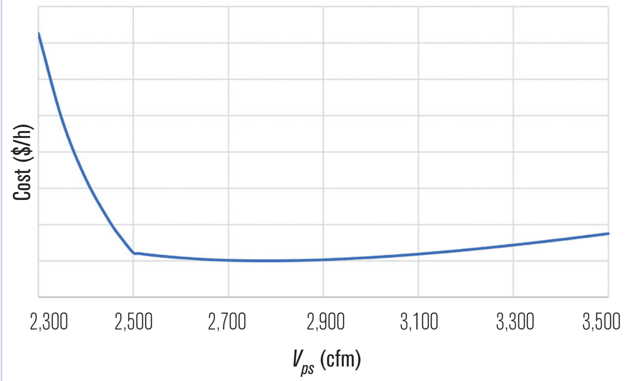


FIGURE 2 Example operating energy cost as a function of overall system airflow.



perfect efficiency, the system primary airflow increases by 1,000 cfm (492 L/s), while the outdoor air requirement only drops by a further 39 cfm (18 L/s), in contrast with the much better rate of reduction we saw before. This process is illustrated in *Figure 1*, with a distinctive kink that occurs once the system moves from having just a single critical zone to having two. In an example with more zones, there would be additional kinks, with one occurring each time a new zone is added to the collection of critical zones.

Putting It Together

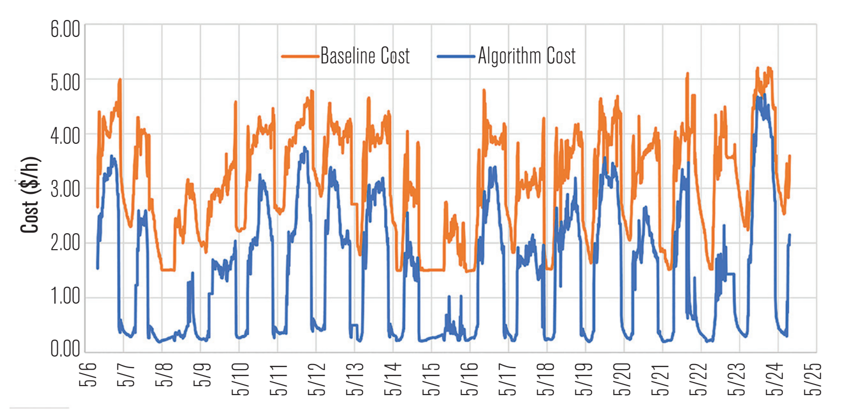
Supplying more air to the critical zone beyond what is necessary to satisfy the sensible load requires that the discharge air be reheated to prevent the zone from getting too cold. This newly required reheat constitutes an energy penalty. At the same time, according to the above analysis, this extra discharge air increases the ventilation efficiency of the system and allows for a reduction in the required outdoor air volume, which reduces the cooling load at the air handler. Starting from the system baseline, this savings generally exceeds the penalty, resulting in a net reduction in energy consumption. (This was recognized and acted upon in RP-1747.[8]) Due to the diminishing returns that were just discussed, though, this is not always the case. At some point while increasing airflow to the critical zone(s), the additional savings that come from the reduction in the outdoor air requirement will be eclipsed by the rise in reheat consumption, fan power for moving the extra airflow, and cooling of the additional return air. *Figure 2* shows how the cost curve might look for the simple example system. In this case, the minimum cost occurs when the overall airflow is 2,800 cfm (1321 L/s), with 720 cfm, 1,080 cfm and 1,000 cfm (340 L/s, 510 L/s and 492 L/s) being discharged to Zones A, B, and C, respectively. The precise location of the minimum depends on many real-time system conditions, such as utility rates and each airstream's temperature and enthalpy. As sensible loads and air parameters change over time, the minimum value will shift somewhere else.

The challenge, then, is to be able to determine where this minimum cost point is for any possible VAV system state. Since all states cannot be known in advance, the solution is to create a mathematical model that uses various system parameters and current conditions to calculate the energy cost of a collection of setpoints in real time. These setpoints should include discharge volumes for each of the zones, as well as the supply air temperature, since, in essence, changing it adjusts the shape of the curve in *Figure 2*. Optimizing this model yields the collection of setpoints that achieves the minimum cost while still satisfying all ventilation and thermal requirements. The next section gives an overview of this process.

Optimization Process

For both real systems and modeled ones, operating costs in terms of dollars per hour can be calculated by summing the energy used at the reheat coils (hot water or electricity), cooling coils (chilled water) and supply and return fans (electricity), multiplied by their respective utility rates. Hot and chilled water consumption are estimated by multiplying each airstream's mass flow rate by its change in enthalpy across the coils, while fan electricity consumption is a function of the total amount of air being pushed through the system. This overall cost

FIGURE 3 Algorithm operating cost versus baseline in test building.



represents the objective function to minimize. (For more detail on a similar cost function, see Raftery, et al.[9])

Before optimization begins using the model, information is harvested from the actual system. This includes each zone's ventilation requirement (adjusted for DCV) and its calculated sensible load. From the air handler, the algorithm grabs outside and return air temperatures and humidities, as well as the mixed and supply air temperatures. Providing current utility rates also allows it to adjust for dynamic pricing, including demand response. This data is to be held constant during the optimization process. The only variables that will be manipulated are discharge volumes to each zone, the supply air temperature, and the percentage of outdoor air. Thus, the model consists of these fixed parameters, the variables and the functions and other values that relate them all to one another. (This is much simpler than a full EnergyPlus model, and it can be used across many systems with little additional effort.)

To optimize the model, the algorithm iteratively selects a supply air temperature and percentage of outdoor air to check. These govern the mixed and supply air temperatures and enthalpies. Next, it sets each of the zone discharge volumes equal to their minimum ventilation requirements and then raises them as necessary to satisfy positive sensible loads and to lower the discharge air temperature enough to minimize stratification. It then continues to raise the discharge air volume in the critical zone(s) until the volume of outdoor air meets the requirement from the multizone ventilation equations discussed earlier. Then the cost is calculated for this iteration, and the algorithm moves on to the next one. The algorithm stops once the cost begins to rise between iterations or an infeasible condition is encountered, and the set of variables with the lowest cost is sent to the real system to be applied. When conditions have varied sufficiently, such as fifteen minutes later or if a zone's sensed occupancy changes, the process repeats, and new setpoints are calculated and applied.

This discrete approach sidesteps the difficulties (and differential equations) of a more dynamic model, while being a more complete solution than a trim-and-respond approach like that of RP-1747. However, since it is too complex to be used natively in standard control logic, then until it is added directly into building automation software, it may instead be written in a language like Java or Python and interface with the system as a software plug-in or via standardized communication protocols like BACnet. Once created, it scales easily across both new and existing systems, without the need for additional hardware.

Performance

As part of the development of the above algorithm, it was coded in Java and uploaded as a software add-on to the building automation system of a 50,000 ft^2 (4,645 m^2) office and classroom building in ASHRAE Climate Zone 5A, where it was compared against the building's baseline control sequence. For the early summer period seen in *Figure 3*, it showed a 70% reduction of hot water consumption, 42% of chilled water, and 53% of electricity, leading to an overall operating cost savings of 54%.

This is perhaps an exceptional result, but during the cooling season, such a scheme can be expected to generate savings on the order of 25% to 30%. Even while economizing, when ventilation is much less of a constraint, using the algorithm's calculated setpoints continues to minimize costs.

Conclusions

This article has discussed some of the trade-offs present in HVAC systems in general and in VAV systems in particular. By going through the ventilation efficiency equations from ASHRAE Standard 62.1-2019, it has revealed how providing and reheating extra discharge air at the critical zone creates a trade-off with cooling requirements at the air handler that can lead to overall reduced energy consumption. This opens the way for models like the one presented to be built and implemented to provide significant savings. Therefore, controls engineers and building automation suppliers should consider including such models in their offerings, and end users should express their demand for these cost-saving algorithms.

References

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