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Time-Averaged Ventilation for optimized control of Variable-Air-Volume systems

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Abstract

Typical Variable Air Volume (VAV) terminals spend the majority of time at their minimum airflow setpoints. These are often higher than the minimum ventilation requirements defined by code, resulting in excess energy use and a risk of over-cooling the spaces. We developed and tested a Time-Averaged Ventilation (TAV) control strategy in an institutional building on the UC Berkeley campus to address this issue. Whenever a zone does not require cooling, TAV alternates the VAV damper between partially open and fully closed so that the average airflow matches a predefined ventilation setpoint. Compared to the existing, base case scenario using single-max VAV logic, this strategy reduced the mean zone airflow fraction from 0.44 to 0.27 during the intervention period. The corresponding reductions in average heating, cooling, and fan power were 41%, 23%, and 15% respectively. In addition to being programmed directly in a native control system, TAV may be applied via sMAP as a low-cost retrofit strategy in any building that has a BACnet network and direct digital control (DDC) to each VAV terminal.

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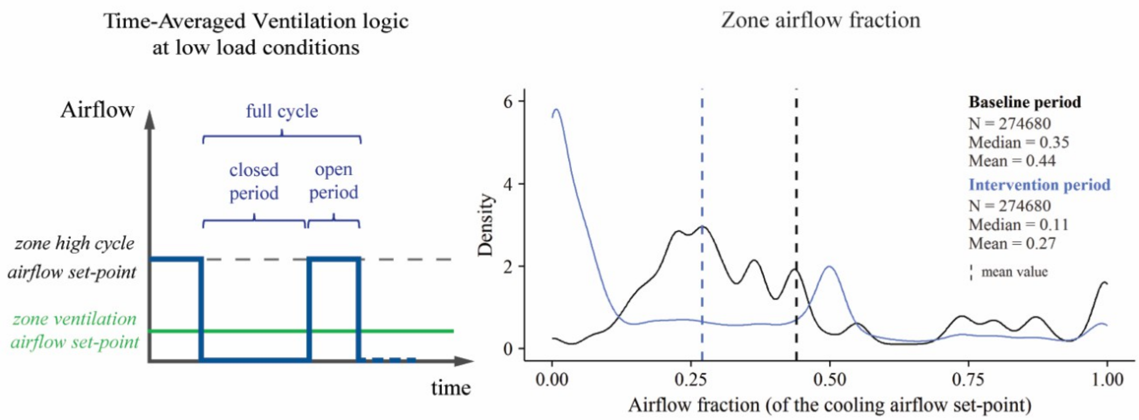
Graphical abstract



Highlights

- We developed a control sequence that alternates each zone VAV damper between open and closed.
- We tested it in a Variable Air Volume system building with single-max control logic and high minimum flow rates.
- Hourly average airflow accurately met the required ventilation rates in each zone.
- Results show a reduction in fan, reheat, and chilled water energy.

Keywords

Air conditioning, Air distribution, VAV system, Control algorithms, Minimum airflow setpoints, Ventilation

Nomenclature

Acronyms	Description
CP	*closed period of zone total cycle time*

OAT_{range}	*range of outdoor air temperature selected*
OP	*open period of zone total cycle time*
PCT	*zone partial cycle time (OP or CP)*
PCT_{min}	*minimum value for zone partial cycle time (180s)*
Q_i	*ith quartile of distribution*
TAV_{ratio}	*zone time-averaged ventilation ratio*
$TAV_{ratio,\ adjusted}$	*zone adjusted time-averaged ventilation ratio*
TCT	*zone total cycle time*
TCT_{max}	*maximum value for zone total cycle time (1800s)*
TCT_{min}	*minimum value for zone total cycle time (900s)*
V_{stpt_high}	*zone high cycle airflow setpoint*
V_{stpt_vent}	*zone ventilation setpoint*

1 Introduction

1.1 Background and motivation

Variable Air Volume (VAV) systems are widely used in commercial buildings in the US, so improving the efficiency of these systems can have a significant impact on energy use. Modern VAV terminals use a feedback controller to adjust the airflow supplied to each thermal zone of a building in order to control the space conditions [1]. At each zone, the VAV damper modulates the supply airflow to respond to changes in cooling or heating load, while also maintaining a minimum airflow for ventilation. As discussed below, the minimum airflow setpoint is often much higher than the required ventilation airflow for the zone, which has a significant adverse impact on energy consumption and occupant thermal comfort.

1.1.1 Benefits of low minimum flow rates

Minimum airflow setpoints that are greater than ventilation requirements waste fan and cooling energy at the air handling unit, and heating energy at the zone reheat coil. When the minimum airflow is larger than needed to satisfy the zone cooling load, the zone temperature is driven down to the zone heating setpoint (or colder if there is no zone reheat). This is one of the main reasons why spaces in many office buildings are typically too cool, even during summer time [2].

ASHRAE Research Project 1515 (RP-1515) [2] tested high (30-50% of the cooling airflow setpoint) and low (10-15%) minimum airflow setpoints in a 91,000 m^2, six building campus facility and a county office building in California. On the campus, low minimum airflow setpoints reduced heating use by 12.2% and cooling and fan energy by 14.6%. In the county office building, heating, cooling, and supply fan energy decreased by 6.1%, 28.8%, and 42.6% respectively. Low minimums also improved occupant thermal comfort due to reduced over-cooling in the summer (dissatisfaction rate dropped by 47%). Similarly, IAQ dissatisfaction dropped by 32% and 62% in the two different study sites, which suggests sufficient air mixing for low minimum setpoint. The study also performed laboratory testing of air change effectiveness with overhead diffusers and showed that ventilation air is well mixed down to 10% flow.

Hoyt et al. [3] used parametric simulations to assess the effect of lower minimum airflow setpoint values on energy building consumption. They demonstrated an average of 31% energy savings for commercial buildings across the seven ASHRAE climate zones by reducing minimums from 30% to 10%.

1.1.2 Barriers to reducing VAV minimum flows

Design guidelines recommend setting the minimum airflow setpoint to the larger of the minimum ventilation requirements and the lowest controllable airflow setpoint allowed by the box controller [1] [4]. However, identifying this minimum controllable airflow setpoint can be a challenge as it depends on attributes of both the VAV terminal and the VAV controller, which are generally provided by separate manufacturers. This is problematic as the manufacturers may not be known until after a project is bid. Additionally, ASHRAE Standard 62.1 [5] ventilation requirements can be complicated to understand and use, which leads many designers to opt for the default minimums listed in ASHRAE Standard 90.1 [6]. Similar to the California Title 24 building energy code (Title 24) [7], this option prescribes minimums of 20% and 30% of the zone cooling airflow setpoint for zones with and without DDC when in deadband, and 50% in heating [6].

Low supply airflow could potentially cause poor indoor air quality (IAQ) due to insufficient air mixing, and could also potentially cause thermal discomfort by triggering cool air dumping and localized draft sensations. However, as described above, the ASHRAE RP-1515 project comprehensively demonstrated that neither is the case in field and laboratory studies at low flow rates [2].

These are some of the reasons why, for most building systems, HVAC designers commonly set VAV minimum airflow setpoints to higher values than necessary (usually 30% to 50% of the cooling airflow setpoint) when compared to the ventilation rates required by codes and standards.

1.1.3 Research on stability and accuracy concerns

VAV flow probes and controllers have been evaluated for stability and accuracy under low flow conditions in laboratory tests and field studies. Dickerhoff and Stein [8] tested VAV terminals from two manufacturers and controllers from four manufacturers. They concluded that boxes can achieve acceptable accuracy and stability with minimum airflow setpoints as low as 10-20%. Liu et al. achieved similar results in an extended scope of the work in 2014 [9].

1.1.4 Shift in building code and control practice from single-max to dual-max control logic

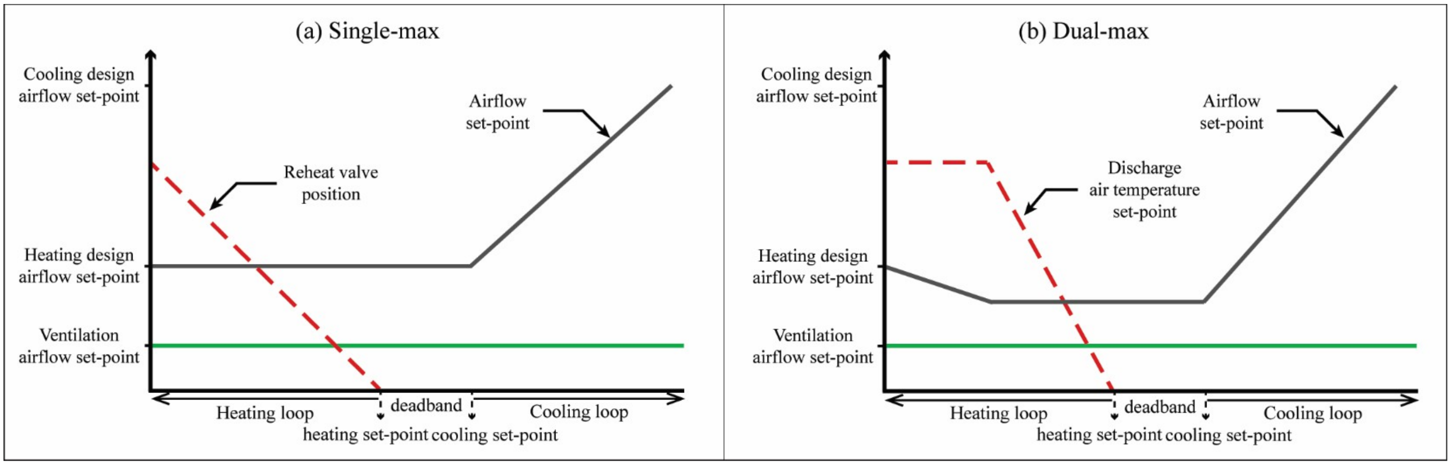


Figure 1.1: (a) single-max control logic and (b) dual-max control logic

Single-max is the predominant zone airflow control logic in existing VAV systems. Figure 1 .1.a shows single-max control logic, where the VAV terminal provides maximum airflow at maximum cooling load. As the cooling load decreases, the supply airflow drops to the minimum airflow setpoint. For single-max zones, this minimum airflow setpoint is equal to the heating design airflow setpoint. In the dual-max control logic (Figure 1 .1.b), the airflow increases in heating mode once the discharge air temperature reaches a certain temperature. Therefore, the dual-max approach has two maximum airflow setpoints; heating and cooling. Because dual-max decouples the airflow required in heating from that required under low load conditions, this logic allows for much lower minimum airflows in deadband compared to the heating design setpoint approach (e.g. 10-20% vs 30-50%).

Dual maximum control logic is prescriptively required in energy codes and standards (Title 24 since 2008 [10], and ASHRAE 90.1 since 2013 [6]) for all new construction with a minimum flow no larger than 20%. For existing construction, retrofitting from single-max to dual-max controls can sometimes be done cost effectively by modifying control programming.

1.2 Problem statement

Given current VAV terminal capabilities, is it possible to control VAV terminals to the minimum ventilation rates prescribed by the current California energy code? Recent research has shown that it is feasible to control to as low as 10-20% of the cooling airflow setpoint, particularly in newer VAV terminal controllers. However, minimum ventilation requirements can be lower than this threshold.

An additional question is whether it is feasible to control older Direct Digital Controls VAV terminals to effectively use dual-max logic. Older VAV terminals may not be able to inherently control to these low

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flows. This may be partly due to low resolution of the analog to digital converter in older controllers or configurable controllers that use hard-coded single-max logic.

1.3 Proposed solution

Our present study proposes a Time-Averaged Ventilation (TAV) method that alternates the position of the VAV damper between closed and open when in deadband mode, as well as in heating mode in some cases. Consequently, the airflow supplied to the zone varies between zero and a predefined positive value that is significantly higher than the VAV terminal's controllable minimum limit—thus avoiding concerns about controllability and accuracy at low flow. Over a period of time, TAV controls the average airflow to the actual ventilation setpoint for each zone as opposed to an arbitrarily high minimum airflow setpoint, achieving significant energy savings.

Even though allowed by ASHRAE Standard 62.1-2013 [5] (section 6.2.6.2. Short Term Conditions) and by Title 24 2013 [7] (exception 2 to 120.1(c)1), to the best of our knowledge, this averaging method has not been implemented in overhead VAV systems in buildings. There are commercial products for sale that use the concept of TAV (such as some underfloor air distribution diffusers), demonstrating that this is a viable approach for ventilation in the market today.

2 Description of the demonstration site

2.1 Building description

Figure 2.2: Sutardja Dai Hall (source: http://www.berkeley.edu/)

We implemented TAV in Sutardja Dai Hall (SDH), an institutional building on the campus of the University of California, Berkeley. We selected the building out of convenience and ease of access to the system controls. The seven floor, 13,000 m^2 building was completed in 2009. It primarily consists of open and private offices, though a few classrooms, an auditorium, a cafe, a data center, a nanofabrication laboratory and a few research laboratories are also in the building. The primary HVAC system serving the non-lab portions of the building uses two large, single duct VAV air handling units (AHU). These AHUs operate with current industry best practices for duct static pressure and supply air temperature setpoint controls. The setpoints dynamically change based on requests for increased pressure or lower supply air temperature from each zone. Both AHUs supply a common central duct, providing conditioned air to a total of 138 zones, of which the majority (110) have hot water reheat coils. The building uses single-max VAV control logic and initially did not have discharge air temperature sensors. We installed discharge air temperature sensors in each VAV reheat terminal to allow us to estimate reheat power at the zone level.

The building has a wider range between the zone heating and cooling setpoint temperatures than usual due to the implementation of "Comfy" [11], a socially-driven HVAC optimization software that adjusts temperature setpoints to improve occupants comfort while saving energy. In a typical office building, the setpoints are 21.1 °C (70 °F) and 23.3 °C (74 °F); in this building, they are 20 °C (68 °F) and 23.9 °C (75 °F).

The TAV implementation excluded 24 critical zones identified by the building manager (e.g. the auditorium, the data center, zones with malfunctioning dampers, etc.). We also removed 5 zones for which the ventilation rate requirements were greater than the minimum airflow setpoints. As a result, we implemented TAV in 109 zones out of 138. Although the AHUs operate continuously, we constrained our experiment and the analysis of results to occupied hours (6am to 8pm) in order to ensure the results are comparable to more typical office building operation.

2.2 Building control systems

The building uses DDC down to the zone level, managed by a Siemens Apogee Building Management System (BMS). In addition, the Simple Measurement and Actuation Profile (sMAP) software package runs on a small fanless PC connected to the BMS BACnet network. sMAP is an open source software program developed by UC Berkeley's Electrical Engineering and Computer Sciences department to store and access time-series data. The sMAP open source architecture and the pybacnet package allow the actuation of devices through the BACnet protocol [12]. With these tools, we can easily implement new control strategies in the building independently from the proprietary building management system software. Also, the applications can be run on any building that uses BACnet with no additional hardware or physical modification to the HVAC system. This makes the implementation scalable to other buildings and BMS systems, and therefore presents significant retrofit opportunities.

2.3 VAV terminal airflow fraction before intervention

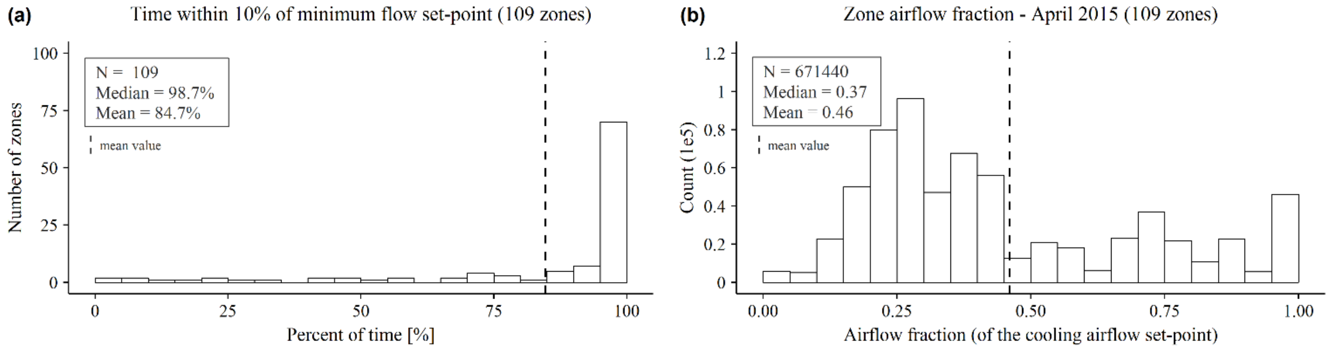

Figure 2.3: Distributions showing (a) the percentage of time each zone spends within 10% of the minimum airflow setpoint and (b) the airflow fraction distribution for 109 zones during occupied hours in April 2015.

Figure 2 .3.a clearly demonstrates that zones spend a large portion of time at their minimum flow setpoint in this building. During occupied hours in April 2015, about 66% of the zones (72 out of 109) spent over 95% of the time at the minimum airflow setpoint. The average zone airflow fraction is about 46% (Figure 2 .3.b), which is a little higher than the average minimum airflow setpoint (41.4% of the cooling airflow setpoint, see Figure 3 .5), due to the small amount of time a zone cooling load requires more than the minimum airflow.

The minimum airflow setpoints in this building are high because the existing controls use single-max VAV logic where the minimum is typically determined by the larger of the design heating airflow or 30% of the design cooling airflow (a common rule of thumb). Actual cooling loads are often a small fraction of the design cooling load [2] and much less than the required peak heating airflow.

The large amount of time that the zones spend at the minimum airflow setpoint illustrates the importance of the minimum airflow setpoint in the design of VAV terminals. Given that zone cooling loads are typically low and the ventilation requirements are usually lower than the minimum airflow setpoints, there is an opportunity to save energy with lower minimums, primarily by reducing over-cooling.-

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3 Methods

3.1 Time-Averaged Ventilation principle of operation

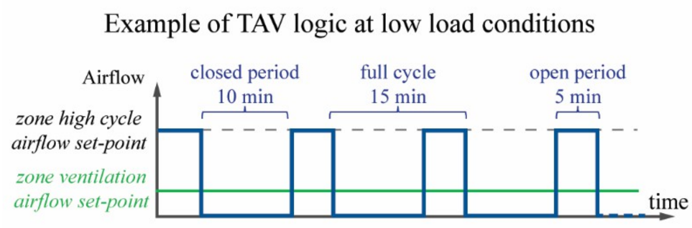

Figure 3.4: Example of TAV logic at low load conditions

TAV alternates the zone damper between partially open and fully closed to achieve a particular average zone airflow. When the zone enters the *high* part of its cycle, we command the zone minimum airflow setpoint to a predefined *high cycle airflow setpoint* (V_{stpt_high}), release the damper command to automatic control, and the damper opens. When the zone enters the *low* part of the cycle, we command the damper closed and also command the zone minimum airflow setpoint to zero (in order to avoid PID loop integral windup issues). We control the length of the high part of the cycle to maintain an average airflow that matches the zone *ventilation setpoint* (V_{stpt_vent}). The example in Figure 3 .4 achieves the ventilation rate required of one-third of the high cycle airflow setpoint so the damper is partially open for one-third of the total cycle time.

3.2 Description of zone ventilation and TAV airflow limits

3.2.1 Determination of the high cycle airflow setpoint (V_{stpt_high})

For demonstration purposes, and to avoid stratification at the heating design conditions, we defined each zone's high cycle airflow setpoint (V_{stpt_high}) equal to 50% of the cooling airflow setpoint for the occupied zones with reheat capability and 20% for the zones without reheat.

3.2.2 Determination of the ventilation setpoint (V_{stpt_vent})

The ventilation rates required by Title 24 are based on the area and the number of occupants in the space. This ventilation rate is the larger of the following:

- 15 cfm (7 L/s) times the number of occupants in the space
- the space program-specific ventilation rate (provided by the code) times the space floor area, 0.15 cfm/ft^2 (0.07 L/s) for most space types

We referred to the Advanced Variable Air Volume (VAV) System Design Guide [1] to determine expected occupancy rates for each occupancy type and construction documents for floor areas. The

Design Guide provides estimates of the resulting ventilation requirements for a few occupancy types, with all values rounded up to the closest 5 cfm (2.35 L/s).

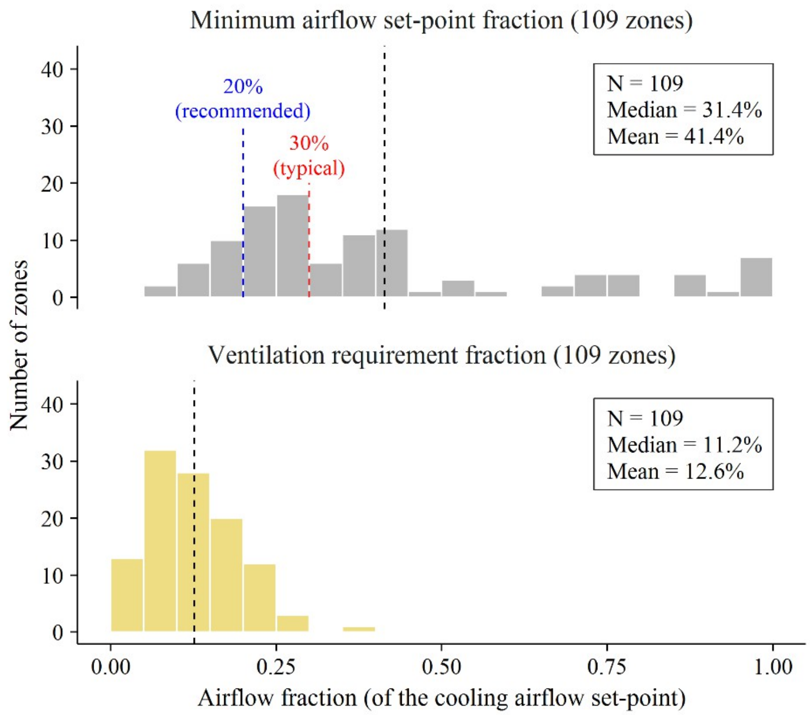

Figure 3.5: Distribution of the minimum airflow setpoints for the building shown as fraction of the zone cooling airflow setpoint (top) and Title 24 ventilation minimum requirements (bottom) for 109 zones.

The top pane of Figure 3 .5 shows the distribution of the minimum airflow setpoints. A minimum setpoint value of 220 cfm (104 L/s) is common in this building even for VAV terminals with high peak flows (1150 to 2400 cfm (543 to 1133 L/s). This explains the 18 zones for which the minimum airflow setpoints are below the 20% recommended. The 30% red-vertical-dashed line illustrates the common practice for VAV terminal minimum airflow setpoint; the 20% blue vertical line corresponds to the maximum setpoint allowed by current energy codes and standards. The bottom pane of Figure 3 .5 represents the distribution of the minimum ventilation rates required by code. For half of the zones, the minimum ventilation requirements are lower than 11.2% of the zone cooling airflow setpoint, which is much lower than both the current median of 31.4%, the conventional minimum of 30%, and the 20% required by code.

3.3 Description of TAV parameters and logic

3.3.1 Determination of the TAV_{ratio} for each zone

We divide each cycle of TAV into two parts: (a) an Open Period (OP) when the damper is partially open, which we also refer to as the *high* part of the cycle; and (b) a Closed Period (CP) when the damper is fully closed (see Figure 3 .4), also referred to as the *low* part of the cycle.

The TAV ratio is given by the ratio of the ventilation setpoint (V_{stpt_vent}) and the high cycle airflow setpoint (V_{stpt_high}) in equation (0).

$$TAV_{ratio} = V_{stpt_{vent}} / V_{stpt_{high}} \tag{0}$$

For TAV to be effective, we need the TAV ratio to be strictly less than 1 ($V_{stpt_{vent}} < V_{stpt_{high}}$¿ . As mentioned earlier, we removed 5 zones that did not fall into this category. California Energy code allows the temporary reduction of the rate of outside air provided to the space for a maximum period of 30 minutes at a time as long as the average hourly rate meets the minimum ventilation requirements [7]. ASHRAE Standard 62.1 [5] (6.2.6.2 Short Term Conditions) also allows design based on averaging conditions for intermittent supply air. According to ASHRAE 62.1 guidelines, the calculations across the 138 zones give *allowed averaging time periods* ranging from 0.2h to 2.2h with a mean *allowed averaging time period* of 1.5h. We decide to follow Title 24 requirements as they are stricter, and so we limited the maximum total cycle time to be no longer than 30 minutes (TCT_{max} = 1800 seconds). For this implementation, we selected a minimum Total Cycle Time (TCT) of 15 minutes (TCT_{min} = 900 seconds). Lastly, we constrained any Partial Cycle Time (PCT) (i.e. OP or CP) to be at least 3 minutes (PCT_{min} = 180 seconds) to account for damper actuator full stroke times (typically 1-2 minutes) and in order to avoid high frequency cycling. Thus, the adjusted TAV ratio for each zone is calculated using equation (0).

$$TAV_{ratio,\,adjusted} = max\left\{TAV_{ratio}\,,\,PCT_{min} / TCT_{max}\right\} \tag{0}$$

In addition, we constrained the adjusted TAV ratio to be greater or equal to the ratio $PCT_{min}/$¿ TCT_{max}, which in our case is 10%. Since the PCT_{min} and TCT_{max} parameters are user-defined, they can be modified as needed for a particular building. In our case, there are nine zones which have an unadjusted TAV ratio smaller than 10% therefore, for those zones, this TAV implementation will supply more airflow than the ventilation requirements.

3.3.2 Determination of OP and CP for each zone

Based on the adjusted TAV ratio for each zone, we determine the Total Cycle Time (TCT) using equation (0).

$$TCT = min\left\{TCT_{max},\; max\left\{PCT_{min}/TAV_{ratio,adjusted},\, PCT_{min}/\left(1-TAV_{ratio,adjusted}\right),\, TCT_{min}\right\}\right\} \tag{0}$$

OP and CP are calculated using the equations (0) and (0).

$$OP = TAV_{ratio,adjusted} * TCT \tag{0}$$

$$CP = \left(1-TAV_{ratio,adjusted}\right) * TCT \tag{0}$$

3.3.3 TAV at the zone level

When a zone does not require any heating or cooling, the zone is considered to be in the deadband mode. TAV is active for a particular zone when it is in the deadband or in heating mode (see Figure 3 . 6), and inactive when it is in cooling mode. Note that TAV may not need to be active in heating mode for systems that use dual-maximum VAV logic.

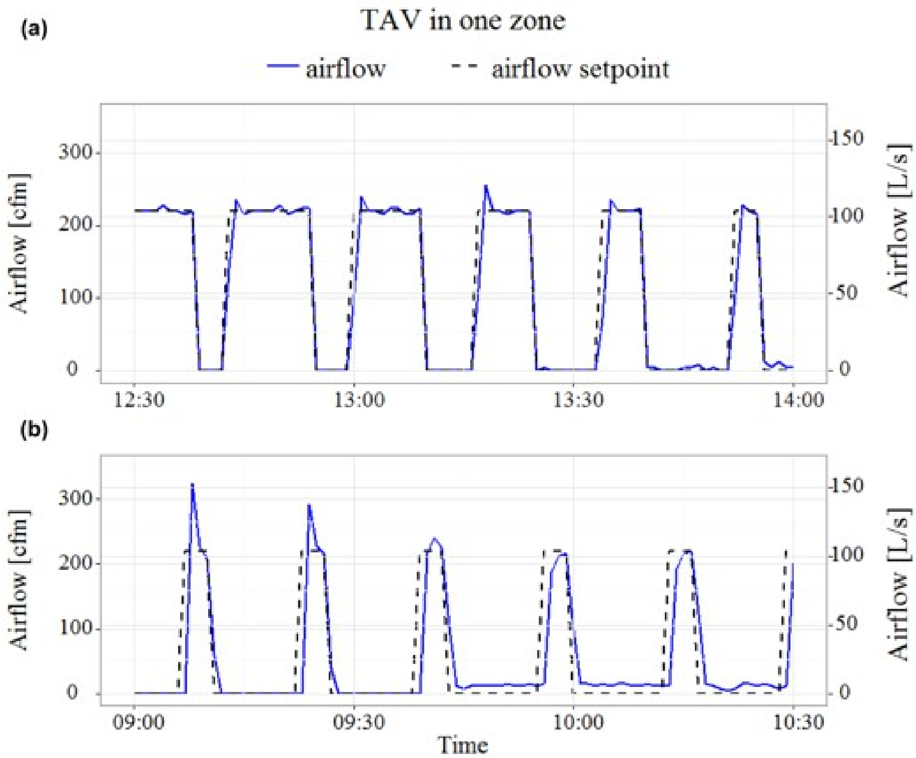

Figure 3.6: Example of airflow and minimum airflow setpoint for a zone when TAV is active in heating mode (a) or in the deadband (b). The measured airflow trend responds to the rectangular profile of the airflow setpoint.

When a zone requires heating (see the first few cycles of Figure 3 .6 (a)), OP and CP are adjusted to account for the need for additional airflow in heating. This prevents air stratification in the room due to high discharge air temperatures. The OP increases proportionally with the heating demand signal for the zone up to the total cycle time so that the damper is open for the full cycle time at the peak heating condition. When a zone is in cooling mode, TAV remains inactive for a period corresponding to the zone total cycle time. This adds hysteresis and prevents the TAV logic from switching on and off quickly when the room temperature is oscillating around the zone cooling setpoint. At a fixed 1-minute interval, the control sequence evaluates whether or not each zone should be in the low or high cycle. For illustration purposes, Figure 3 .6 (b) also shows some particular cases of airflow overshoot (see first cycles) and non-zero airflow measurements when the damper is closed (see last cycles).

3.3.4 TAV at the AHU level

We added a limiter to the sequence to stagger the times at which the dampers change position in order to avoid a large number of zone dampers opening and closing at the same time, potentially causing highly variable flows at the AHU level. This condition may otherwise occur on AHU startup, or in the case of SDH (which operates continuously), also at the onset of nighttime setback when many zones enter the deadband mode simultaneously. The limiter determines the number of dampers allowed to be closed at the same time. There is no constraint when the dampers open to avoid any issue with meeting a required ventilation rate. Equation (0) dynamically calculates this limit as:

$$limit = max\left\{\frac{Number\ of\ active\ zones - \sum of\ TAV_{ratio,\,adjusted}\ for\ active\ zones}{number\ of\ intervals\ per\ TCT_{min}}\right\} \quad (0)$$

Overall, the limiter is used in order to desynchronize the zone cycle. It is based upon the number of zones in which TAV is currently active and the number of intervals over the minimum cycle time. We round the limiter to the nearest whole number of zones.

3.3.5 Field study implementation schedule

We present the results of a field study conducted between April 1st and April 30, 2016, in which TAV was alternatively switched on and off based on weather forecasts using the weather underground website [13]. As part of our measurement & verification methods, and in order to be able to compare periods with similar outside conditions and building conditioning demand, we designed the activation schedule of TAV to obtain similar distributions of the outdoor air temperatures (OAT) between the baseline period and the intervention period. Each period contains ten complete weekdays.

3.4 Outside air temperature (OAT) distribution during the study period

We retrieved the actual outside air temperature data using the building site weather station. Using Tukey's test [14] for each distribution, we removed the outlier day(s) for which the daily mean OAT was not in the range shown in equation (0).

$$\begin{gathered} Q \\ (¿¿3 - Q_1), Q_3 + k * (Q_3 - Q_1) \\ Q_1 - k * ¿ \\ OAT_{range} = ¿ \end{gathered} \quad (0)$$

Where Q_1 is the lower quartile of the distribution, Q_3 is the upper quartile of the distribution and $k = 1.5$.

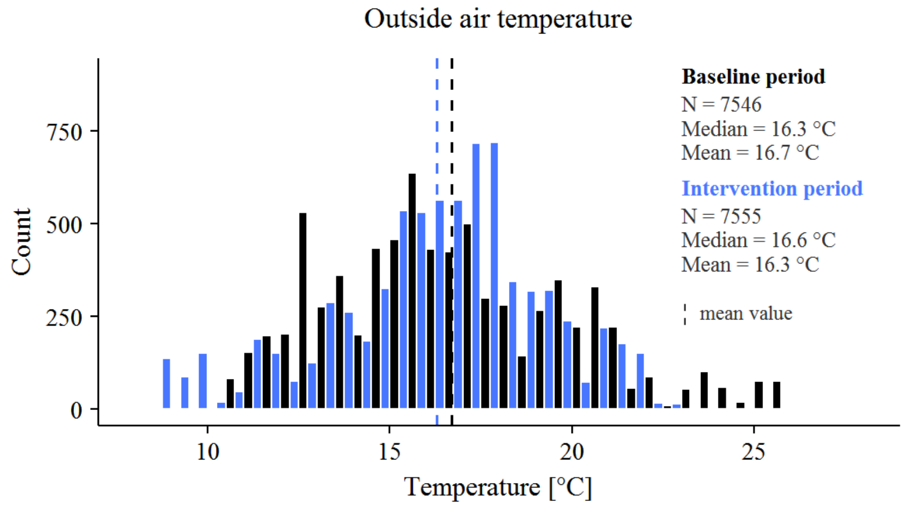

Figure 3.7: Distribution of the outside air temperature during the baseline and intervention portions of field study period (April 1-30 2016)

After removing the outlier days (one day each for the baseline and intervention periods), the distribution of OAT is similar between the baseline and intervention period with the OAT slightly higher (+0.4 °C) on average during the baseline period (Figure 3 .7).

4 Results

4.1 Zone temperature distribution

Since lower minimums reduce over-cooling, we expect the overall indoor temperature to increase with the implementation of TAV. As noted in Section 2.1, this distribution is wider than usual due to the operation of Comfy [11].

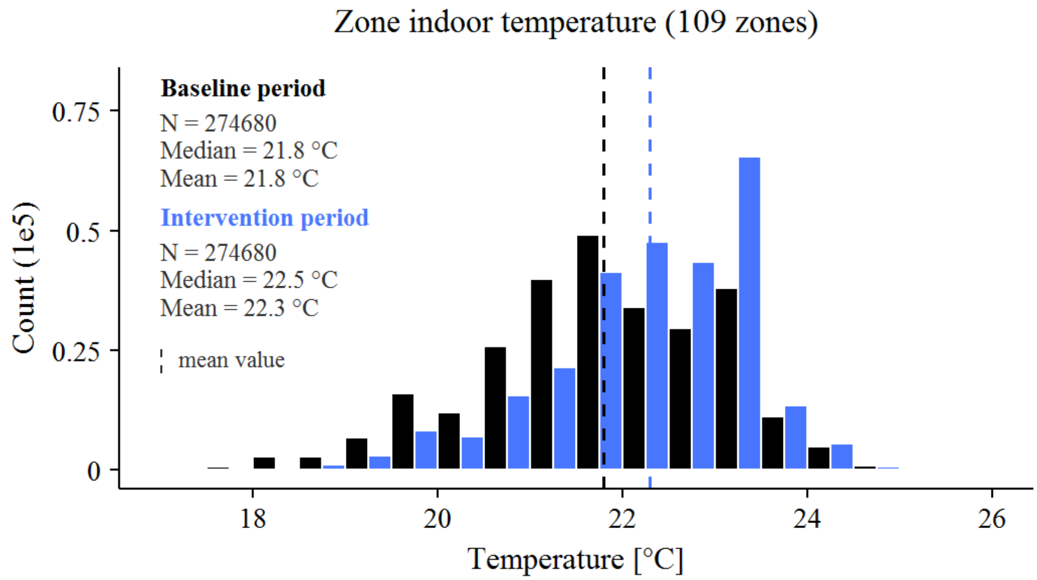

Figure 4.8: Zone temperature distribution for zones selected for TAV only

The distribution of the zone temperatures is quite similar between the baseline and intervention periods, though the mean temperature increases by 0.5 °C with TAV (Figure 4 .8). These results also reflect that TAV does not interfere with the zone temperature control loops and that zone temperature are maintained within similar temperature ranges.

4.2 Airflow distribution

4.2.1 TAV performance

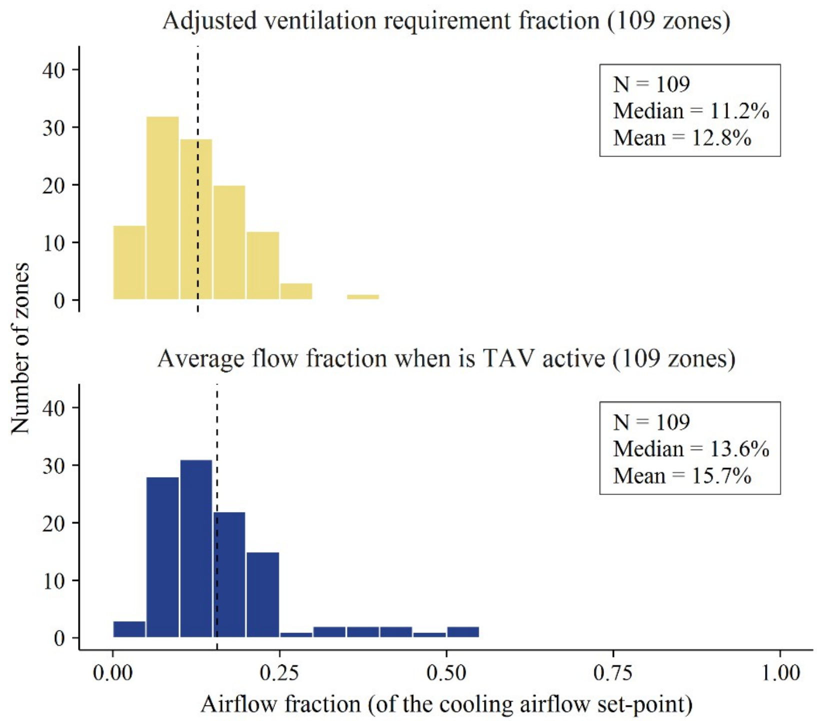

Figure 4.9: Distribution of adjusted TAV ventilation setpoint (top) and zone average airflow fraction when TAV is active (bottom) shown as a fraction of the cooling airflow setpoint.

The top graph of Figure 4 .9 shows the distribution of the ventilation setpoints used in the TAV implementation. For the bottom chart, we averaged the measured airflow fraction for each zone for the time where TAV was active. In order to discard the effects of leaking dampers or potentially inaccurate measurements at very low flows, we assumed 0 cfm airflow whenever the damper is commanded to be closed. The similarity in distributions confirms that TAV is generally operating as expected in maintaining target ventilation values on average.

For each zone, TAV is activated when the zone is either in heating or at low load conditions. Considering the season, very few zones were in heating during this period, or if so, for a very short period of time. The majority of zones were at low load conditions when TAV was active, and knowing that most of the zones spend more than 95% at the minimum (see Figure 2 .3.a), the similarity in airflow distribution is not surprising. The slightly higher values for the actual airflow fraction distribution (15.7% versus 12.8% on average) can be explained by heating periods and that the limiter keeps some zones in the high part of the cycle for longer than the designed OP when many dampers try to close at the same time.

4.2.2 *Airflow fraction distribution*

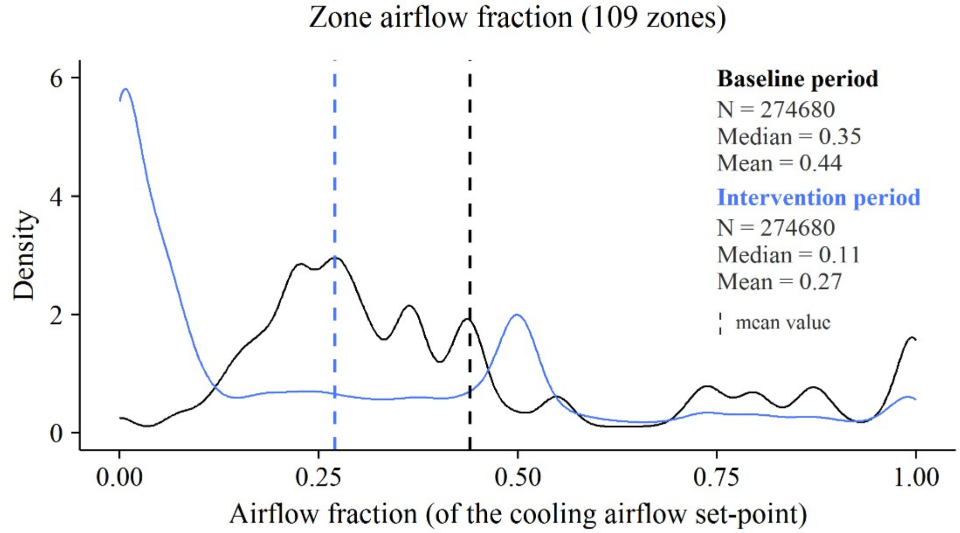

Figure 4.10: Airflow fraction distribution in the building

The distribution of the airflow at the zone level is very different between the baseline and intervention periods. Figure 4 .10 shows the distribution of airflow fraction (airflow supplied to the zone as a fraction of the cooling airflow setpoint) across all zones throughout the study period. Since the average minimum is lower during the intervention period, the mean airflow fraction decreases, from 44% to 27% (38.6% reduction). The median of the airflow fraction decreases from 35% to 11% (68.6% reduction). As expected, we see a peak during the intervention period for the lower values where the airflow is near zero as well as a second spike at 50% of the cooling airflow setpoint, which is specific to the design of our current experiment that sets the high cycle airflow setpoint (V_{stpt_high}) to be 50% of the cooling airflow setpoint for all zones with reheat.

4.2.3 *Airflow at AHUs*

Similarly to the airflow distribution at the zone level, we found a reduction in the overall airflow measured at the AHUs. Along with a reduced overall airflow requirement, there is some added variation due to TAV.

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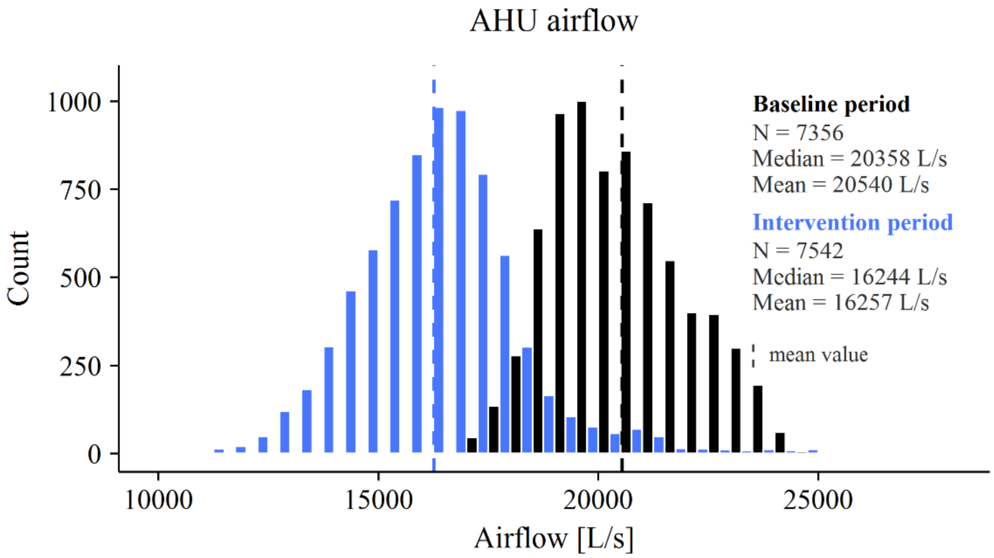

Figure 4.11: AHU level airflow distribution

TAV reduced the mean airflow in the intervention period by 20.9% and the median by 20.2% (Figure 4 . 11). Although these results are similar to those at the zone level, the total airflow also includes the 29 zones in which TAV never runs.

4.3 Energy analysis

The energy use in the building HVAC system can be broken down into fan and chilled water power consumed at the AHU, and the sum of reheat power at each zone. While there are electricity meters for the fans in this building, reheat power consumed in the building is not metered. Reheat power at each zone was therefore estimated based on the density and volumetric heat capacity of air, the measured zone airflow, and the temperature difference between the zone discharge air temperature sensor and the AHU supply air temperature, with a small temperature adjustment to account for duct heat gain, sensor error, and passing valves. This temperature adjustment is dynamically updated whenever the reheat coil has been closed for a period of 15 minutes or longer. There are supply and return temperature sensors and a flow meter for measuring chilled water power consumption at the AHUs. Unfortunately, we identified issues with the chilled water heat meter after the study was complete and therefore, we report estimated chilled water power consumption using the same method as for the reheat power, which only accounts for sensible heat removed by the cooling coil. This is not a significant issue given the observed outside air dewpoint during the study period. Table 4 .1 summarizes the findings, showing that TAV significantly reduced fan, chilled water, and reheat power during the intervention period.

	Baseline period	Intervention period	Comparison
Number of days (excluding outliers)	9	9	

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Mean supply fan power (measured) [kW]	21	16.9	-19.5%
Mean total fan power (measured) [kW]	31.2	26.4	-15.4%
Mean chilled water power (estimated) [kW]	95.9	74.3	-22.5%
Mean reheat power (estimated) [kW]	0.328	0.194	-40.9%

Table 4.1: Energy consumption summary

Table 4 .1 also shows very low reheat power, which can be explained by the time of year in which the study occurred. During this period, the dynamic supply air temperature reset control sequences and the relatively low outside air temperatures allow relatively high supply air temperatures. These higher supply air temperatures, combined with the low zone heating temperature setpoints and relatively high plug loads, mean that reheat requirements are quite low in this particular building at this time of year.

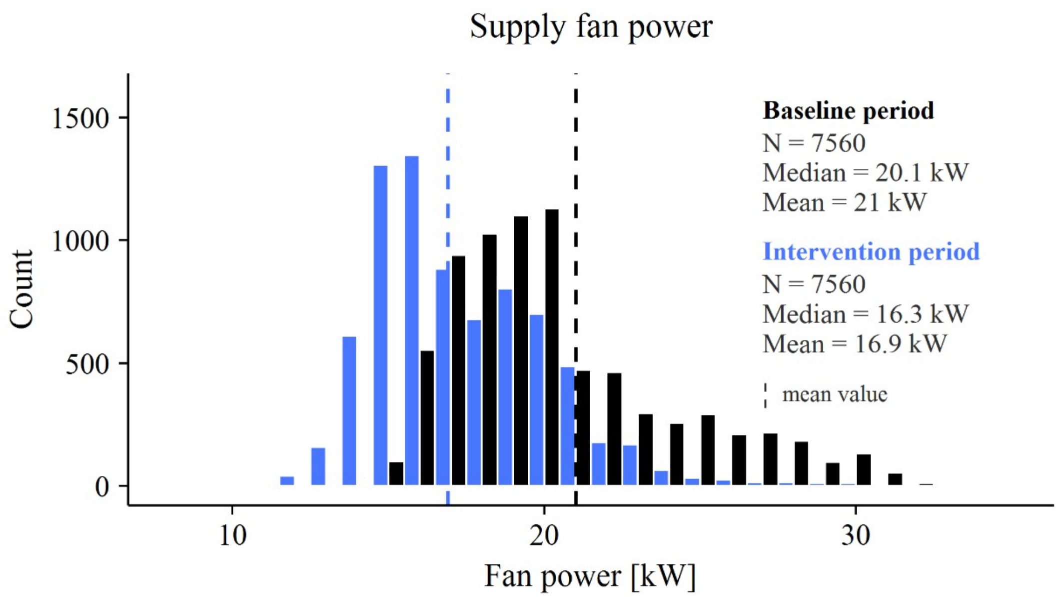

Figure 4.12: Supply fan power distribution

As expected, given the airflow results shown in Figure 4 .11, the supply fan power distribution also shifts to lower values in the intervention period with TAV (Figure 4 .12). Compared to the base case, the mean decreases by 19.5% for the supply fan power and by 15.4% for the total fan power.

5 Discussion

5.1 Outcomes of implementing TAV

5.1.1 Building airflow and energy consumption

The results show that we can use the TAV method to meet ventilation requirements while reducing total fan, cooling, and reheat energy use. SDH uses single-max VAV control logic and thus has higher minimum flow setpoints than in buildings that use dual-max control logic. The potential savings due to TAV will probably be lower in buildings that already use dual-max VAV logic. However, ventilation minimums are often lower than the common practice of using 20% of the cooling design airflow even when using dual-max logic. Additionally, dual-max logic is relatively new and not consistently implemented, even in new construction, despite its requirement in codes. The majority of existing buildings with DDC at the zone level may still continue using single-max for many years to come and would benefit from TAV.

5.1.2 TAV as a functional approximation for dual-max VAV control logic

When a zone is in heating mode, the TAV logic here adapts the duration of the open period (OP) and closed period (CP) according to the zone heating loop output, as described in Section 3.3.3. Measured discharge air temperature (DAT) stays under 32.2 °C (90 °F) when the zone is in heating mode, which prevents thermal stratification issues. Thus, TAV represents a functional approximation of the dual-max VAV control logic, which has significant energy and comfort benefits as a standalone retrofit measure [4]. As TAV can be implemented independently from proprietary BMS software using the sMAP software, there is a large potential for cost-effective (software only) retrofit applications for VAV buildings that use BACnet control at the zone level.

5.1.3 Damper longevity

Compared to normal operation, dampers traveled longer distances (i.e. a much greater fraction of full stroke) during the intervention period when TAV was active (see Figure 5 .13 (a)), which presumably has a negative impact on service life.

Conversely, the damper actuators do not move at all during the closed period when TAV is active, which reduces the number of cycles (i.e. energizing the actuator to change direction) executed by the dampers (see Figure 5 .13 (b)), which presumably has a positive impact on service life.

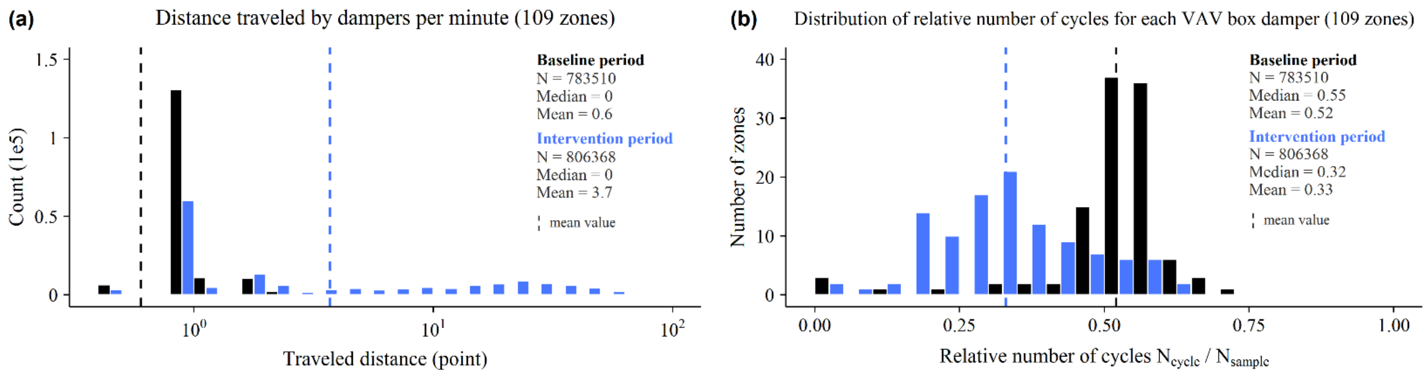

Figure 5.13: (a) Distance traveled by the 109 dampers in April 2016. The distance is given in points corresponding to a percentage opening of the damper. (b) Relative number of cycles by dampers—i.e. fraction of time the damper rotation changes direction over the total number of timestamps.

Further work needs to be done in order to assess the overall impact of TAV regarding mechanical fatigue and damper/actuator service life. In the meantime, we considered several alternatives for TAV in case a negative impact on service life is expected. In the present version, TAV uses a predetermined total cycle time for each zone. Within the total cycle time, OP and CP are calculated using the zone TAV ratio. The present version of TAV uses 15 minutes as a starting point for the total cycle time. However, Title 24 allows a total cycle time to be up to one hour, while the averaging period allowed by ASHRAE Standard 62.1 based on room volume can even be longer than one hour. Using a longer total cycle time could reduce the frequency of damper cycling and the total distance traveled by each damper.
Alternatively, we could use a fixed length of time for the CP, a variable total cycle time, and a fixed damper position during the OP (e.g. damper position fixed at 50%). During the OP, we would then measure the airflow at that fixed damper position and end each cycle when the average airflow over the whole cycle reaches the ventilation setpoint. Using this approach, the damper actuator would move only twice per cycle—a large reduction in damper actuation frequency when compared to typical damper operation (PI loop control). Additionally, in the present version, at the end of the OP we command the damper closed (0%), which explains the short response time (see Figure 3 .6 (b) for an example). However, at the beginning of the OP, we set the airflow setpoint while releasing the override on the damper closed command. The nature of the PI loop control causes a longer response time and yields a small error between the airflow setpoint and the actual airflow provided to the space at the onset of each OP, leading to a slight overestimation in average airflow. Using a fixed CP while monitoring the average airflow to reach the ventilation requirement would resolve this minor issue.

5.1.4 Acoustics

In parallel with the damper longevity, additional damper movement, step changes in airflow, and fully closed dampers might cause acoustical problems. And in fact, there were mechanical issues in some of the zones. A building occupant in one zone described the sound (presumably induced by damper movement) as a “dying whale”. After removing a total of 3 zones (of 138) with defective dampers, we received no further complaints. To investigate this noise issue, we used a SoundTrack LxT sound level meter to conduct A-weighted sound pressure level (Leq in dB(A)) measurements for 28 VAV boxes across 4 floors of the building. In each floor, we selected 6 to 9 boxes that were the closest to the main supply duct of the building. The procedure followed the Performance Measurement Protocols for Commercial Buildings (PMP) guidelines for the basic level [15] with 1 second average measurements. For each measurement, we alternated the damper position between closed and open over a 5 minute interval during a 6 to 20-minute test period. The static pressure of the system was fixed at 425 Pa (1.7 in.w.g) during the measurements. With a mean value of 39.59 dB(A), all average sound levels are well within the PMP recommended range (35-45 dB(A) for open offices without sound masking, 40-50 dB(A) for corridors and lobbies). The maximum value was 45.6 dB(A) obtained in an open office in the Computer Science department surrounded by computer processor noise. We heard barely noticeable clicking sounds associated with the damper movements for 4 of the 28 boxes. Figure 5 .14 shows the time series of the sound pressure level measurements for 3 boxes situated in open office areas.

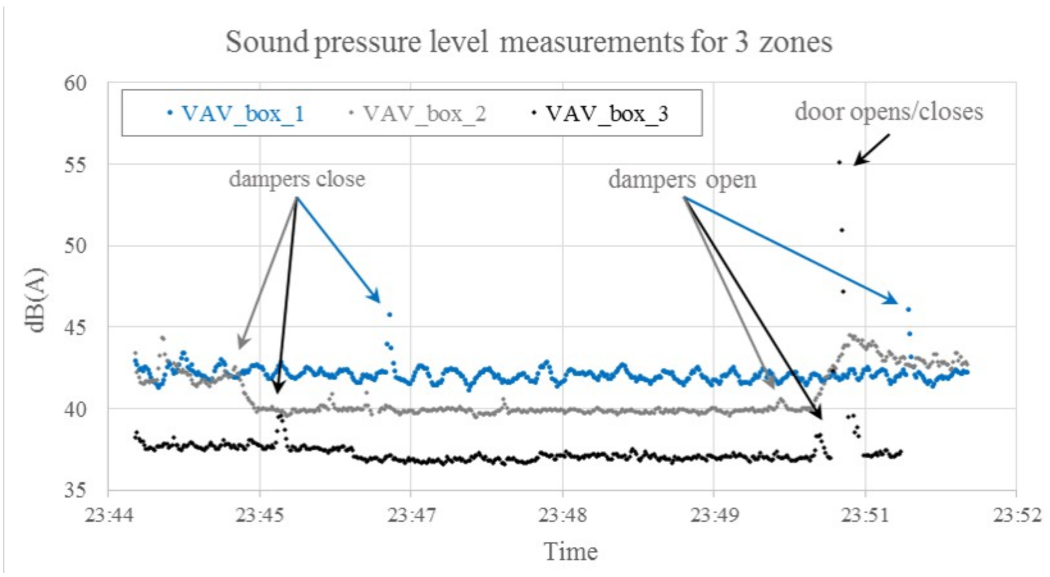

Figure 5.14: Sound pressure level measurements for 3 zones with clicking sound

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The VAV box 1 data from Figure 5 .14 is the noisy zone pointed out earlier in this section. The clicking sound that we noticed for this zone was barely perceptible. Although the peaks are slightly above 45 dB(A), the average sound level over the test period is 42.1 dB(A). We also noticed a clicking sound for VAV box 2 when the damper opens, and masked by the airflow sound when the damper closes. For this particular zone, we never received any complaints even though the noisy airflow causes the alternation between closed and open periods was more noticeable (+/- 3 dB(A) on Figure 5 .14). The data from VAV box 3 shows that the clicking sounds stay below 40dB(A) which is very small compared to sounds created by other typical office activities—i.e. a door that opens and closes (~55 dB(A)).

We understand that a multitude of parameters could have played a role in creating a situation where the sound of a particular box became an inconvenience for the building occupants. Even though we believe it is worth investigating noise-related issues associated with TAV, our tests were not able to replicate such situations. In all cases the noises were below the limit of perception, or just barely above, so much so that we had to overlay the damper position data on the acoustic data to identify when the dampers moved. We were only able to perceive a clicking sound for 4 boxes (out of 28); measurements for most of the boxes show a relatively flat line with no distinctive bump associated with damper movements.

5.2 Limitations of the study

We recognize several limitations in our study. Firstly, the data reported in the present study are from the swing season only. We expect to see greater fan power reduction during summer when the overall airflow requirements are higher and the fan operates closer to design conditions. However, it is likely that more zones would operate in cooling mode during this period, which would reduce the amount of time that TAV is active. Extending this study from one month to a year would answer these questions, but this was outside the feasible scope of this experiment given available time, and the needs of other research projects involving this building’s HVAC system.

Additionally, as we described in the results section, we did not have a heat meter to measure total reheat power, and instead we estimated the effect. The accuracy of this estimation method is unknown, but given the magnitude of the reheat energy consumption compared to fan and chilled water, it is unlikely to have a significant impact on the overall findings. Testing the accuracy of this estimation method would be an interesting research topic in itself, as it can potentially provide an energy cost estimate with the instrumentation common to most modern VAV systems (i.e. zone airflow and discharge air temperature sensors). A similar concern applies to the chilled water power estimates. However, as the same method was used to estimate energy use in both baseline and intervention periods, neither is likely to significantly affect the relative results of one approach versus the other.

5.3 Scope of the project relative to IAQ

It is critically important to note that TAV does not affect the outside airflow rate entering the building—this is controlled by the outside air damper at the air handling unit. SDH has an independently controlled outside damper that is separate from the economizer damper. By reducing the total building airflow requirement, TAV causes a higher fraction of outside air in the supply air leaving the air handling unit when it is operating at design outside air conditions (i.e. when the airside economizer is not active). Thus, though the distribution of how much outside air (or ventilation air) supplied to each zone changes, the total outside air provided in the overall building remains the same.

Extensive research has been conducted, investigating the relationship between ventilation rates, IAQ, occupants work performance and health [16] [17], especially on SBS syndrome prevalence [18]. Results have shown that higher ventilation rates are beneficial. However, a literature review from Carrer et al. [19] argues that the use of high ventilation rates to cover any risk of exposure is limited by the lack of scientific evidence to justify such rates, relative to the building energy cost they induce. Additionally, most of the reviewed studies do not account for outdoor air quality. As a Sundell et al. [20] review points out, further work needs to be pursued in order to understand the extent of the relationship between ventilation rates and health.

We are aware of the difficulty in identifying the ideal ventilation rate to provide to people in a given set of conditions. The aim of this study was not to identify optimum ventilation rates and does not take any stance on what these rates should be. It rather demonstrates that TAV allows us to precisely meet a particular zone flow rate, regardless of any perceived or real limitation on VAV terminal performance at low flow. The actual value for each zone of the building is introduced as an input to the control sequence which remains at the operator's discretion.

5.4 Future work

There are also a number of questions worth investigating from an occupant perspective. It would be interesting to evaluate to what extent the occupants perceive noise caused by damper movement, if the occupants are disturbed by changes in in airflow, and if TAV impacts their overall environmental satisfaction.

Additionally, Title 24 changed in 2013 to allow ventilation setback to 25% of the occupied rate when an occupancy sensor says a zone is unoccupied. Time averaged ventilation is specifically mentioned as an allowable approach. It is also possible to reduce outside air intake at the air handler based on actual occupancy.

Lastly, at the moment zone minimum airflow rates are constant values, regardless of air economizer status. However, when the AHU is in full economizer mode, there is far more outside air than when it is in minimum outside air mode. The ASHRAE Standard 62.1 multiple spaces equation allows a control strategy to dynamically reduce the zone minimum setpoints based on the total outside airflow rate at the AHU [21] [22]. There is a current ASHRAE Research Project (RP-1747) entitled "Implementation of RP-1547 CO2-based Demand Controlled Ventilation for Multiple Zone HVAC Systems in Direct Digital Control Systems" that is developing and testing control sequences for this purpose.

Dual-max VAV logic is a prescriptive code requirement for new buildings with minimum airflow setpoints generally limited to no more than 20% of the cooling airflow setpoint. The results of this study demonstrate that it is feasible to control to far lower average percentages, yielding further energy savings. This indicates that codes and standards should adopt VAV minimum airflows that match the zone minimum ventilation requirement instead of the 20% value, or VAV terminal controllable minimums.

6 Conclusion

A typical thermal zone in a building spends the majority of time in the deadband between heating and cooling, supplying a constant airflow defined by the minimum airflow setpoint. Due to a number of perceived and real issues regarding VAV terminal capabilities at low airflow, this setpoint is typically much higher than the minimum ventilation requirement defined by code. This has adverse energy consumption and occupant thermal comfort implications.

We tested a Time-Averaged Ventilation control strategy in the majority of zones of an institutional building on the UC Berkeley campus as a means of overcoming these issues. This study demonstrates that TAV is able to control the average minimum zone airflow to a particular ventilation rate, as defined by Title 24. Compared to the existing, base case scenario using single-max VAV logic, this strategy reduced the mean zone airflow fraction from 0.44 to 0.27 during the intervention period. The corresponding reductions in average heating, cooling, and fan power were 41%, 23%, and 15% respectively.

As TAV also adjusts the airflow supply when the zone is in reheat according to the zone heating demand, it has proven to be a functional approximation for dual-max control logic.

TAV was implemented here using sMAP software to override VAV terminal setpoints over the BACnet network, independently from the proprietary BMS software. Thus, using this approach, TAV can be applied as a low-cost retrofit strategy in any building, existing or new, that has a BACnet network and DDC controls to each VAV terminal.

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