REDUCING AIRFLOW ENERGY USE IN MULTIPLE ZONE VAV SYSTEMS

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SYSTEMS

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ABSTRACT

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Variable Air Volume (VAV) systems are the most popular HVAC systems in commercial buildings. VAV systems are designed to deliver airflows at design conditions which only occur for a few hours in a year. Minimizing energy use in VAV systems requires reducing the amount of airflow delivered through the system at part load conditions. Air Handling Unit (AHU) fans are the major drivers of airflow in VAV systems and installing a Variable Frequency Drive (VFD) is the most common method of regulating airflow in VAV systems. A VFD drive does not necessarily save energy without use of an appropriate control strategy. Static pressure reset (SPR) is considered to be the most energy efficient control strategy for AHU fans with VFDs installed. The implementation of SPR however has many challenges; for example, rogue zones—zones which have faulty sensors or failed controls and actuators, system dynamics like hunting and system diversity.

By investigating the parameters associated with the implementation of SPR in VAV systems, a new, improved, more stable SPR algorithm was developed and validated. This

approach was further improved using Fault Detection and Diagnostics (FDD) to eliminate rogue zones.

Additionally, a CO₂-Demand Control Ventilation (DCV) based minimum airflow control was used to further reduce ventilation airflow and save more energy from SPR. Energy savings ranging from 25% to 51% were recorded in actual buildings with the new SPR algorithm.

Finally, a methodology that utilizes historical VAV data was developed to estimate the potential savings that could be realized using SPR. The approach employed first determines an effective system loss coefficient as a function of mean damper position using the historical duct static pressure, VAV damper positions and airflows. Additionally, the historical data is used to identify the maximum mean duct damper position realizable as a result of insuring a sufficient number of VAVs are fully open at any time. Savings are estimated by shifting the damper distribution mean at each time to this maximum value and reducing the static pressure to achieve the same overall system airflow rate. The methodology was tested on three different buildings with varying system characteristics. Savings estimates correlated well to the savings actually realized from SPR. This result has significant implications for energy service providers, who could use the predictions to guarantee savings from SPR.

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CHAPTER 1

INTRODUCTION

The Department of Energy estimates 41% of primary energy consumption in the US has been in the building sector, 46% of which is consumed in the commercial building sector. Of this, 49.2% of all building energy consumption is used for Heating Ventilation and Air Conditioning (HVAC) making it a significant portion of the national energy bill [1]. In 2011, the ventilation portion of commercial building energy consumption in the U.S. reached 1,580 trillion Btu accounting for 27.7% of HVAC energy consumption in commercial buildings. The ventilation energy use is projected to increase steadily by 0.4% annually through 2040 [2].

Currently, variable air volume (VAV) HVAC systems are the most popular choice among new commercial building constructions and major retrofits. VAV systems gained popularity in the 1970's and are gradually replacing constant air volume (CAV) systems. CAV systems regulate temperature in a zone by mixing cold and hot air and delivering a constant volume airflow rate of air into each zone. VAV systems are more energy efficient than CAV systems because fan power can be reduced at part loads; VAV systems also reduce mixing of hot and cold air streams and reheat energy use at part loads [3]. Energy efficiency in VAV systems is accomplished by taking full advantage of load diversity in a system: When cooling or heating loads are reduced, less air should be moved through the air distribution system; when cooling loads are reduced, less water should be moved through the chilled water coils; when ambient wet bulb temperature is low, less air should be moved through the condenser/cooling tower [4].

VAV systems have two major components, including: the applied air handling unit (AHU) which has the primary function of supplying conditioned air; and the terminal devices (e.g. VAV boxes, Zone dampers, etc.). The applied equipment must have a supply fan and can have either a return or exhaust fan. It has an outdoor air damper, a cooling coil and can have a heating coil. The terminal device which receives air from the AHU is usually a VAV box with a damper. The VAV box can have a fan and/or reheat coil.

In a typical building application, each thermal zone is served by a separate VAV terminal with independent controls to maintain occupant comfort. The VAV terminal controls specify important parameters such as minimum airflow rate, temperature set point and reheat set point. VAV terminal controls regulate the amount of air flowing to the zone and can be either pressure dependent or pressure independent. In pressure-dependent control, the damper position is controlled by a signal from the zone thermostat, and airflow through the terminal device is not tracked and multiple terminals in the same system affect one another. Pressure-independent control is more common in single-duct VAV systems, in which the zone thermostat sends a signal to a velocity reset controller, which in turn controls the damper position to maintain airflow required for the zone [5]. Pressure independent systems are more stable and save more energy than pressure dependent systems.

VAV systems in general have multiple control loops: a temperature control loop through the chiller; a pressure control loop through the air-handling unit (AHU); and a temperature and airflow control loop through VAV terminal units; [6] outdoor air and economizer control loop through the AHU. All control loops have to be designed and operated synergistically in order to deliver conditioned air and ventilation to a building at the minimum cost.

The AHU pressure control loop and airflow control loop work together to deliver the right amount of air to different zones in the system. As individual VAV boxes adjust their damper positions to let the required airflow into a zone, the pressure produced by the AHU supply fan is controlled to meet a duct static pressure setpoint. Traditionally dampers and inlet guide vanes (IGV) have been used to modulate the fan but variable frequency drives (VFD) have become very popular and are even required by energy codes (ASHRAE 90.1 and California Title 24) for fans above 5 hp [7,8].

The control of a fan is more important to the energy consumption of the fan than the type of fan selected and static pressure reset has the potential to reduce fan energy by up to 50%, reduce fan noise, vibration and bearing wear by reducing fan operation in surge. The long term energy consumption of a fan is determined by the application and control of the fan in a system, not by the peak efficiency of the fan, and the impact of the system on fan energy is more important than the selected fan [9]. Although energy efficiency has been playing a major role in the economies of developed nations for a long time, cost effective energy efficiency remains vastly untapped globally [10]. A recent article points to the fact that fan capacity control is one of the most widely appreciated yet underexploited phenomenon in the air movement industry [3]. Fan modulation should

also be able to produce adequate static pressure, eliminate excess static pressure, exploit load diversity, maximize fan energy savings and provide stable control to ensure comfort in all zones [4].

Fan control with a VFD drive can be classified as fan-outlet control, supply-duct control and critical zone control. Fan-outlet control requires the pressure sensor to be located at the outlet of the fan and sends a signal to the VFD to maintain a duct static pressure at design conditions. This method has the advantage of using a well calibrated factory mounted pressure sensor. In supply-duct control, the pressure sensor is typically located two-thirds of the way down the main supply air duct. The pressure sensor sends a signal to the VFD to maintain a set static pressure at this location to meet the design pressure requirement of the most remote terminals. This method requires field installation of sensors. Locating the sensor in the duct of a large system can be very difficult. In critical zone control, the sensor can be located anywhere in the supply air duct; the duct static pressure setpoint is dynamically reset to a point where the AHU fan generates only enough static pressure to meet the airflow requirement of the most critical zone at any moment [11].

Fan power to move air through a system with a system efficiency of η , a total system pressure drop of ΔP and a system airflow rate of \dot{Q} is calculated as shown in Equation (1.1).

$$Power_{fan} = \frac{\Delta P \times \dot{Q}}{\eta} \tag{1.1}$$

The intersection of system curves and fan curves for different types of fan control are shown in Figure 1 with the fan initially running at system design airflow rate of Q_0 and

total system pressure of P_0 . To visualize fan power requirements as a function of control types and airflow rate, consider a change in system airflow rate from Q_0 to Q_1 . In an AHU without a VFD, the closing of dampers at VAV terminals shifts the system curve, which raises the pressure from P_0 to P_1 on the fan curve and reduces system airflow from Q_0 to Q_1 . The fan power requirement is reduced, but not proportionally with the reduction in airflow rate since pressure drop increases. With a VFD installed in the AHU, fan-outlet control allows the fan to slow down to a new fan curve while maintaining the duct static pressure thereby resulting in a slightly decreased system pressure of P_2 . Savings can be increased by controlling VFD fan speed to maintain static pressure near the most distant VAV boxes. This is called supply-duct control. Supplyduct control allows the system pressure to be reduced to P_3 .



Figure 1. System curves for AHU fan control schemes: without VFD, with fan-outlet control, with supply-duct control and with critical-zone control.

In critical-zone control, the duct static pressure setpoint is constantly adjusted to meet the airflow requirement of the most critical zone; therefore, no minimum pressure needs to be maintained and the system curve can approach zero system pressure when system airflow rate approaches zero. Critical-zone control allows the system pressure to be greatly reduced to P₄.

The critical zone control method is the lowest cost and highest energy savings strategy because it allows factory installation and calibration of the pressure sensor and for most systems with direct digital control (DDC) and a Building Automation System (BAS), communications to the terminal devices required are already in place. The strategy or control sequence used to achieve critical zone control is commonly referred to as Static Pressure Reset (SPR). Even with the availability of advanced controls and BAS, control sequences are not programmed correctly [12]. Literature review and background research in the field were carried out for this dissertation; data was collected from several VAV systems with SPR implemented and major issues pertaining to programming of sequences were investigated.

CHAPTER 2

BACKGROUND

2.1 Static Pressure Reset

Static Pressure Reset is not a new concept in the HVAC industry. One of the earliest works to propose the strategy can be traced back to a proposal in 1989 where it was described as a terminal regulated air volume (TRAV). The proposal was to modulate the supply fan to meet VAV airflow requirements [13]. The TRAV system envisioned a fully integrated DDC system with a central control logic, taking information from the VAV terminal boxes and the AHU and making decisions. Such a system was futuristic in 1989, but DDC controls and BAS systems have become very common today.

An early study showing savings from resetting static pressure setpoint was by Englander at Princeton in 1990 [14]. He showed how this strategy is only effective for Variable Speed Drive (VSD) systems and how the strategy is only practical with a DDC control system. Trane engineers in 1991 described static pressure reset as the most energy saving control strategy for VAV systems and likely to be the default strategy for the future. They listed many challenges to the actual implementation, for example, rogue zones associated with faulty sensors or actuators, system dynamics like hunting (control instability forcing a VAV terminal device to be switching back and forth between cooling and heating), and system diversity (variation in the sizes and airflow demand for different VAV terminal units) [11].

Englander and Norford in a 1992 paper described a practical application of static pressure reset strategy using zone feedback controls in a small air system and went on to demonstrate savings in larger systems [15]. A later paper described the actual algorithm which requires the static pressure setpoint to be incremented by 5% when there is a single starved VAV box (when a VAV damper is 100% open and airflow is less than what is required in the zone) in the system [16]. Static pressure reset was implemented on a large system with 128 VAV boxes in which they simulated human occupancy using space heaters and the summation of airflow rates from VAV boxes as the control parameter to reset static pressure setpoint [17]. Most of the early work focused on using zone airflow feedback to reset the static pressure setpoint.

With an increase in the application of pressure-independent VAV terminal units in which the zone damper tracks the airflow accurately, the zone damper can be used directly for feedback to the system level controller by resetting the static pressure setpoint such that at least one of the VAV boxes remain open [6,18].

Using the most open VAV box as the critical zone became the most common method of resetting the static pressure setpoint. This method was criticized for implementation difficulty, being affected by VAV box malfunction, communication loss and the inability to be implemented on pneumatically controlled systems. Two new methods were proposed using AHU feedback: (1) Resetting the duct pressure setpoint by measuring the differential pressure across the supply fan to maintain a constant resistance in the duct; and (2) Resetting the duct static pressure based on airflow ratio [19].

Static pressure reset saves even more fan energy by interacting with other system components. Theoretical models developed by Liu showed the effect of static pressure reset on fan airflow, fan head, air leakage, fan power and thermal energy consumption. Liu's research showed that the most important savings come from reduction of duct leakage as a result of reduced static pressure in part load [20]. Static pressure reset interacts with other control strategies which can complement or negate the energy savings. The strategy with the most direct impact on static pressure reset is supply air temperature reset.

Theoretical savings from modulating fan capacity is cubic with the speed of the fan. The cubic savings are only approached in the case of a fan that is open (for example a cooling tower fan) [4]. Actual savings for VAV systems are much less and even lesser in cases where the electric motor driving the fan is oversized [21]. Savings have been reported from as low as 19% [6] to as high as 60% [22] for systems with static pressure reset compared to systems with a constant static pressure setpoint. The energy savings from static pressure reset depends on climate, building usage and the size of HVAC equipment, VAV system type (dependent vs independent) and other system control strategies.

2.2 Static Pressure Reset Control Methods

After identifying a critical zone and the control signal to be used for resetting the duct static pressure setpoint, it is important to select an appropriate control method. There are two types of control methods commonly used:

2.2.1 PID control

PID control refers to the proportional-integral-derivative controller commonly used in industrial control. In most cases, only the proportional-integral (PI) part of the control is used in a direct acting control loop. The controller works to maintain one of the VAV boxes at a fully open position. The fully open VAV box is considered to be the most critical VAV box requiring the most airflow at that particular instance. The static pressure shall be controlled from a minimum pressure to a maximum pressure determined during the air balance. The maximum pressure is the required pressure to provide design airflow in all VAV boxes downstream of the duct static pressure sensor. The supply fan speed is controlled to maintain duct static pressure setpoint when the fan is on [23].

PID control is difficult to implement and in general not recommended for resetting static pressure [23], as any instability in the control will cause the fan to go into surge [24]. Most of the difficulties with implementing PID control are in determining stable system parameters [17].

2.2.2 Trim and respond control

With trim and respond control, the static pressure setpoint is set to a minimum allowable duct static pressure value for a given system when the fan is off. When the fan is turned on, the BAS scans the system for dampers with high open position (e.g., above 90%) and then positively increments the static pressure setpoint by a small value (e.g., 0.05 in.W.C) and holds for a given amount of time (e.g., 5 minutes). After holding for a set time, the BAS scans the system again for dampers with high open position and if there is no open damper, the system decrements the pressure setpoint until it reaches the minimum allowed for the system.

If implemented in a system where the VAV zone damper position is unknown, a pressure request is made when the ratio of the zone's actual supply airflow to supply airflow setpoint is less than 90%. The control logic is designed to be slow-acting to avoid hunting which results in toggling between different airflow conditions at high frequency.

A variation of this sequence is having multiple dampers open to consider pressure increase and the response rate determined as a function of the number of dampers open [23,25].

2.3 Control Stability

To address the issue of instability in control, two separate PID loops were proposed by Wang; one for increasing the static pressure setpoint and one for decreasing the static pressure setpoint. Although this approach reduced overall system instability, it increased the dynamics of the control. Wang simulated and tested this approach on a big building and he showed that if the PID loops are not well tuned, they will cause wild oscillations in the systems and defeat the purpose of the static pressure reset [26]. Field work has shown the PID control to work well only with low proportional gain and high integral gain resulting in very slow response [23].

Although the Trim and Respond control strategy has been shown to have implementation difficulties, it is recommended as the best of the two control methods for static pressure reset [24]. This control strategy is more flexible and easier to tune than PID control and should be used for all demand-based reset sequences. It is also easier to deal with rogue zones (VAV with stuck damper or communication loss) [23].

2.4 Background: Data from Actual Systems

Although static pressure reset is an important energy saving strategy, implementation is a challenge and wrongly implemented algorithm results in more energy consumption or control instability than a system without static pressure reset. Figures 2- 5 shown below is a plot of duct static pressure and duct static pressure setpoint collected for different VAV systems over a three-day period showing implementation problems. These figures illustrate various controls issues associated with static pressure reset. The system in Figure 2 had no adequate response time; hence the static pressure setpoint changes dramatically in a short period of time.



Figure 2. Static Pressure Reset: No adequate system response time

In comparison, the system response shown in Figure 3 does not call for a gradual decrease in the pressure setpoint and the setpoint is reset to the minimum allowed as soon as the critical zone becomes satisfied. The rapid decrease in pressure drives the VAV dampers wide open, which calls for pressure increase and the system cycles continuously.



Figure 3. Static Pressure Reset: No gradual decrease in static pressure setpoint

Selection of incorrect system parameters for a trim and respond algorithm or incorrect tuning of a PID control loop can lead to control instability. Figure 4 shows a case where use of wrong parameters significantly increased system dynamics, leading to an unstable control system.



Figure 4. Static Pressure Reset: Unstable control

When a single zone is used as the critical zone to drive static pressure reset, the presence of rogue zones can lead to more energy consumption than a system without static pressure reset. A rogue zone will always drive the pressure setpoint to the maximum allowed, which is usually higher than the pressure setpoint for no-static pressure reset. Figure 5 shows a system where a rogue zone is driving the setpoint and even with the fan running at maximum capacity, it was unable to meet the maximum allowable setpoint of 1.5 in.w.g.



Figure 5. Static Pressure Reset: Rogue zone driving pressure setpoint to maximum allowed

Data was also collected from a system with a functioning static pressure reset which is plotted in Figure 6 below. The figure shows how the static pressure setpoint increases as the airflow requirements through the day increases and the duct static pressure closely tracks the setpoint.



Figure 6. Static Pressure Reset: Proposed logic based algorithm

In the first part of this thesis (Part I), a new static pressure reset algorithm is proposed with the goal of achieving more stable control. A logic based approach was used to implement a trim and respond method. Part II continues to explore the robustness of static pressure control by addressing two key issues: 1) High minimum airflow in commercial buildings and 2) The "Rogue" zone problem associated with static pressure reset. The final part (Part III) introduces the idea of using historical data to drive an informed static pressure reset control which allows energy savings from static pressure reset to be predicted using historical VAV damper position and airflow data.

CHAPTER 3

PART I: ENERGY EFFICIENT STATIC PRESSURE RESET IN VAV SYSTEMS

3.1 Introduction

The logic based algorithm proposed in this work is different from the conventional duct static reset algorithm that uses a single critical zone to drive the static pressure reset. Single zone critical zone will fail whenever a damper is driven open as a result of failure or control override. Some authors [24] have called for including a number of zones to form the critical zone. Including more zones reduces the effect of a rogue zone but has a practical drawback; it does not maintain a buffer zone which causes instability in the controls.

A practical static pressure reset control algorithm was designed using the trim and respond control method and implemented in an office building for validation. The proposed control strategy is computationally simple and utilizes control points that are readily available on most current and many older BAS. The proposed control strategy is widely applicable to VAV systems with VFDs installed at AHU fans, regardless of the types of chiller plants or terminal units. The work described below highlights ease of implementation and how to achieve control stability.

3.2 Methodology

To implement the algorithm, the following parameters must first be determined:

Terminal unit damper open threshold (*Pos_open*): The terminal damper position beyond which a terminal is considered open. This parameter should be sufficiently high to result in minimal pressure setpoints, yet sufficiently low to avoid starving terminal units.

Minimum number of VAV terminal units to be kept wide open (N_min) : This parameter is to be minimized to avoid starving boxes, but must be sufficiently high to avoid hunting.

Maximum number of VAV terminal units to be kept wide open (N_max) : This parameter is to be set sufficiently far from N_min to avoid hunting, but close enough to N_min to avoid starving terminal units.

Minimum setpoint pressure (P_{min}): The minimum practical setpoint pressure the AHU fan could provide, constrained by the operation of the VFD and the duct pressure losses in the system.

Maximum setpoint pressure (P_{max}): The maximum setpoint pressure the AHU is designed to provide for the system.

Setpoint pressure change step (ΔP): The amount of change to the setpoint pressure that the controls can make in each loop. This variable must be sufficiently small to maintain control resolution but sufficiently large to allow the system to respond to transient demands reasonably quickly. Delay time (t_delay): Delay time after each cycle of control implementation. The 'delay time' should be long enough to account for the physical constraints of terminal damper units opening and closing, yet sufficiently short for the system to be responsive.

A new logic-based algorithm, shown schematically in Figure 7 is developed. This strategy executes the following steps in a loop:

- Poll through all terminal unit controllers and determine the number of terminals with damper position greater than *Pos_open*.
- If the number of open terminals is greater than N_max and the duct static pressure is less than P_max , increase the duct static pressure setpoint by ΔP .
- Else if the number of open terminals is less than N_{min} and the duct static pressure is greater than P_{min} , decrease the duct static pressure setpoint by ΔP .
- Else, maintain current duct static pressure setpoint.
- Delay by an amount of time defined in *t_delay*.



Figure 7. Algorithm for proposed static pressure reset control strategy

3.3 Normalized Fan Power Savings

Whenever a control strategy is applied to a system, it is very important to verify the actual savings that can be achieved by such a control strategy. Comparing fan power consumption in the period before implementing the control strategy (baseline period) and the period after implementing the control strategy (post-baseline period) will not accurately measure the savings because of differences in system load conditions such as weather, internal loads and occupancy for the two periods. Therefore, it is necessary to normalize the fan power consumption to minimize the effect of changing load dynamics for the baseline and post-baseline period.

VAV systems respond to load conditions by controlling the amount of airflow through the terminal devices which in turn regulate the amount of airflow produced by the AHU fan when a variable frequency drives (VFD) is installed. It is necessary to derive an adjusted baseline that represents the power the fan would have drawn using the baseline control strategy, but at the post-baseline airflow rates.

Fan power is the product of system pressure drop, ΔP , and a system airflow rate, \dot{Q} , divided by the total system efficiency, η .

$$Fan Power = \frac{\Delta P \times \dot{Q}}{\eta}$$
(3.1)

The total system efficiency is the product of the efficiencies of the fan, motor drives, motor and VFD. Thus, as shown in Equation 3.2, the total system efficiency can be calculated from power, pressure drop and system airflow rate:

$$\eta = \frac{\Delta P \times \dot{Q}}{Fan \ Power} \tag{3.2}$$

It is necessary to quantify the total system pressure drop for efficiency calculations. The total system pressure drop can be expressed as the sum of the pressure drop downstream of the duct static pressure sensor and that upstream of the sensor. Downstream of the sensor, all pressure drops add up to the measured duct static pressure (DSP) at the sensor. Upstream of the sensor, the pressure drop across the coils and filter is proportional to the square of the system airflow rate, where k is the proportionality constant. The total system pressure, excluding the outdoor air damper, is expressed in Equation 3.3:

$$\Delta P = \mathrm{DSP} + k\dot{Q}^2 \tag{3.3}$$

The constant of proportionality k can be calculated as shown in Equation (3.4):

$$k = \frac{\Delta P - DSP}{\dot{Q}^2} \tag{3.4}$$

The design parameters of the fan and system can be used to calculate for k in Equation 3.4 and applied for all values of airflow in the system. Combining Equations 3.2 - 3.4, the system efficiency can be computed as shown in Equation 3.5, where the system power, duct static pressure and system airflow rates are recorded inputs for the baseline case.

$$\eta = \frac{\left(\text{DSP} + \mathbf{k}\dot{Q}^2\right) \times \dot{Q}}{Power} \tag{3.5}$$

After evaluating Equation 3.5 with baseline data, the efficiency needs to be extended to all values of airflow. This is done by regressing the efficiency against airflow. An exponent curve of the form in Equation 3.6 can be used to quantify the system efficiency as a function of the system airflow rate in cfm.

$$\eta = a + b\dot{Q^n} \tag{3.6}$$

Where a and b are regression coefficients and n is the order of the curve.

Substituting Equation 3.3 and 3.6 into Equation 3.1, the fan power for the baseline control strategy can be calculated as a function of system airflow rate as shown in Equation 3.7, where DSP is the duct static pressure corresponding to the system airflow rate during the baseline period.

$$Fan Power = \frac{\left(\text{DSP} + k\dot{Q}^2\right) \times \dot{Q}}{a + b\dot{Q}^n}$$
(3.7)

Post-baseline fan power is recorded after the control strategy is implemented; this is referred to as the measured post-baseline power. Baseline fan power is calculated using Equation 3.7 with the duct static pressure (DSP) for the baseline period and airflow conditions (\dot{Q}) for the post-baseline period; this is referred to as the adjusted baseline power. The difference between the adjusted baseline power and measured post-baseline power is the energy savings from implementing the duct static pressure reset control strategy.

The energy savings calculation method presented here is also applicable to cases where fan power cannot be measured directly provided the airflow and static pressure can be measured and the efficiency can be calculated from the efficiencies of components that make up the AHU fan system.

3.4 Case Study

A case study was carried out on a 15,800 ft² office building located in Dayton, Ohio. The building is served by two AHUs and 25 VAV terminal boxes divided into two sections. Clear demarcation in the form of a 12-inch wall prevents air exchange between the two sections; however, some thermal energy may be exchanged. This study was carried out in the larger section of the building that includes one AHU and 21 VAV terminal boxes and serves 12,000 ft² of floor area. The AHU has a forward curved draw-through fan designed for low-pressure applications with a design total static pressure of 2.5 in. w.g.at commissioning. The fan has a design maximum airflow rate of 9,600 CFM at 8.87 BHP. The fan is driven by a 10-hp motor controlled by a VFD with 1,800 rpm at 60 Hz. A schematic of the AHU is shown in Figure 8 below.


Figure 8. Case study AHU schematic

The control strategy proposed in this study was tested during peak cooling periods in the summer of 2013. The values for the control parameters were selected based on the criteria described in the methodology section (Section 3.2) and are listed in Table 1.

Control Parameter	Value
Pos_open	90%
N_min	5
N_max	7
P_min	0.2 in. w.g. (50 Pa)
P_max	1.0 in. w.g. (249 Pa)
delP	0.05 in. w.g. (12 Pa)
t_delay	300 seconds

Table 1. Control parameters implemented in case study

Data was collected for 6 weeks at 5-minute intervals when the AHU ran at near peak conditions and required peak airflow rates. Testing during these conditions guarantees that airflow requirements using the proposed control method, that reduces duct static pressure, can be met during the rest of the year when airflow rates are lower. This logic based approach developed here was applied to the system under similar weather and load conditions and the resulting energy savings were compared to the baseline case.

3.5 Results

At the design system airflow rate of 9,600 cfm, the fan curve indicates a total pressure rise across the fan of 2.5 in. w.g. The observed duct static pressure at the design airflow rate is 1.0 in. w.g. Therefore, the proportionality constant for this system can be calculated as:

$$k = \frac{\Delta P - DSP}{\dot{Q}^2} = \frac{2.5 \text{ in.w.g.} - 1.0 \text{ in.w.g.}}{(9,600 \text{ cfm})^2} = 1.63 \times 10^{-8} \frac{\text{in.w.g.}}{\text{cfm}^2}$$

The system efficiency during the baseline period was computed using Equation 3.5 and the plot of efficiency vs airflow of the system is shown Figure 9 below.



Figure 9. AHU system efficiency against airflow rate during the baseline period

The slight downward trend is the result of additional economizing, and hence lower pressure drop across the outside air damper, as well as reduced air leakage from the ducts, at low airflow conditions. A linear regression is observed to best quantify the system efficiency as a function of the system airflow rate in cfm.

$$\eta = a + b\dot{Q}^n = 0.3341 - 1.047 \times 10^{-5} \times \dot{Q}$$

The adjusted baseline fan power was calculated using Equation (3.7) and the results are compared to post-baseline measured fan power in Figure 10. The proposed control strategy resulted in AHU fan power savings in all operating conditions throughout the data collection period. The fan power savings are on average 51% of the baseline period fan power consumption.



Figure 10. Measured post-baseline power and adjusted baseline power versus system airflow rate for four full days in the post-baseline period. Fan power reduction occurred at all airflow rates

3.6 Summary

In summary, Part I presented a practical control strategy that dynamically resets the duct static pressure setpoint in multiple zone VAV systems. A methodology to calculate fan power normalized for airflow rate was also developed. The selection and tuning of control parameters allow the proposed control strategy to be applicable to buildings with a variety of building automation systems and hardware limitations. The proposed control strategy was implemented in an office building in Dayton, Ohio. Fan power savings were realized during the post-baseline period, without affecting the performance of the building HVAC system. The average observed fan power savings in this case study after implementation of the proposed control strategy was 51%, compared to a constant duct

static pressure setpoint. Higher savings are expected in buildings with over-sized AHUs. Higher savings are also expected in buildings with continuous HVAC operations, since fan power savings are greatest at the low airflow rates that occur during unoccupied hours.

The logic based approach described in this part has two major limitations:

- 1.) The determination of system parameters to ensure stable and energy efficient control is largely based on trial and error.
- 2.) Energy savings cannot be estimated before implementing the control strategy due to lack of knowledge of the system's final response.

In other to overcome these limitations, a data driven statistics based approach to static pressure reset is investigated in Part III of this dissertation.

Part II of this dissertation will extend the logic based static pressure reset algorithm by using Fault Detection and Diagnostics (FDD) to eliminate rogue zones. In Part II, airflow reduction using CO_2 – Demand Controlled Ventilation (DCV) is also investigated as a means to reduce part-load fan energy consumption.

CHAPTER 4

PART II: ENERGY SAVINGS FROM ROBUST CONTROL OF STATIC PRESSURE BASED UPON ZONAL OCCUPANCY FOR MULTIPLE ZONE VAV SYSTEMS

4.1 Introduction

The major advantage of a VAV system is in part-load operation. In CAV systems, supply air fans operate at full or near full design capacity all the time; part-load conditions are accommodated by mixing hot and cold air streams or by reheating cold air. In VAV systems, the cooling/heating airflow rate in each zone is determined by the deviation of the zone temperature from its setpoint and is usually driven by a PID logic in the VAV controller. VAV boxes that do not require heating or cooling at a given time will close to a minimum position required for ventilation. Supply fan speed and airflow is varied to meet these demands, which results in fan energy savings.

VAV systems also address part-load conditions for ventilation air. Ventilation air in multiple zone recirculating VAV systems is typically made up of fresh outdoor air and recirculated air. Traditionally, the ratio of outdoor and recirculated air was fixed based on design requirements, and in most cases remained the same over the lifetime of the

system. Today, two prominent strategies are employed to reduce the ventilation airflow rate of VAV systems. These are now included in ASHRAE Standard 62.1 - Ventilation for Acceptable Indoor Air Quality, and include: (i) Ventilation Rate Procedure that uses a prescribed zone ventilation rate depending on the type of activities carried out in the zone; and (ii) Indoor Air Quality Procedure. The indoor air quality procedure allows for the use of CO₂ – Demand Controlled Ventilation (DCV) [27]. CO₂-DCV is shown to be very effective in airflow reduction especially in areas that are seldom occupied, e.g., conference rooms, school gymnasiums, etc. [28,29]. Another method to achieve ventilation airflow reduction is to set occupancy and standby statuses for each individual VAV box in a system and use occupancy sensors to drive control of the VAV box. A study reported by the PNNL showed on average that 17.8% can be saved nationally by using occupancy sensors to control VAV boxes in office buildings [30].

Reduction of airflow via VAV damper manipulation is one way to achieve fan energy savings in a VAV system. Another method is via reduction in total system pressure since the required fan power to move air through a VAV system is

$$Fan Power = \frac{\Delta P \times \dot{Q}}{\eta}$$
(4.1)

where ΔP is total system pressure drop, \dot{Q} is system ventilation airflow rate, and η is the system ventilation efficiency. Numerous studies have been carried out in the area of VAV system pressure control [11,23,24,31]. The approach leading to the highest energy savings is the critical zone based duct static pressure reset. Critical zone based duct static pressure reset is when the duct static pressure setpoint is changed continuously to meet the airflow requirement of the most critical VAV box(es). Several implementation

strategies exist but they all point to minimization of system pressure to achieve desired airflow. Static pressure reset , however, suffers from a challenge that is referred to as the rogue zone problem [23,24]. Rogue zones are zones that constantly demand high airflow and drive the pressure reset request as a result of failure of a component (VAV Dampers or Thermostat). Taylor (2007) indicated, rogue zones must be addressed if static pressure control is to be successful and suggested the use of periodical trend reviews to exclude rogue zones [24]. Another researcher suggested that another way to eliminate the rogue zone problem is to oversize the VAV boxes in questionable zones [23].

A real-time method to address the rogue zone problem as part of the overall duct static pressure reset strategy is to use Fault Detection and Diagnostics (FDD). FDD methods are well established in other fields like Aerospace, Automotive, Manufacturing and Process Control Engineering, but it is still relatively new in HVAC. California's Title 24 [8] requires FDD in some HVAC applications. A good review for FDD methods, classification, ease of implementation and applications can be found in [32,33]. FDD methods can be classified broadly into Quantitative (simple and complex physics and mathematical based models) and Qualitative methods (Expert rules, threshold limit and first principles). Rule-based methods are one type of a Qualitative FDD that uses system knowledge and process history data to derive a set of rules to isolate faulty operation from proper operation [34].

For this study, a coupled CO₂-DCV strategy and a reset algorithm based upon a rulebased FDD method to diagnose faulty thermostats and VAV dampers is used to vary ventilation rates and save energy. The uniqueness of this approach is its exploitation of

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available BAS data and utilization of real-time feedback from the zones to ensure the proper operation of the duct static pressure reset algorithm.

4.2 Methodology

The overall objective of this study is to minimize ventilation airflow and implement a duct static pressure reset algorithm with fault detection capabilities to guarantee minimum ventilation power and energy savings. To implement the proposed algorithm, a Building Automation System (BAS) with continuous and automated real-time data collection is necessary. Sensor and control point data used in this study as organized by zone/system include:

Zonal data

- Volumetric airflow rate (cfm) for each VAV terminal unit
- Damper position on each VAV terminal unit
- Occupancy Status on each VAV terminal unit
- Minimum airflow rate setpoint for each VAV terminal unit
- Zone temperature and zone temperature setpoint for each zone

Overall system data

- Duct static pressure sensor for the VAV system
- Power sensor on the VFD
- Duct static pressure setpoint

The implementation requires three steps: (i) resetting the minimum zone airflow based on the CO₂ value in the zone; (ii) detecting rogue zones in the system by performing FDD; and (iii) resetting duct static pressure based on the damper positions of the critical zones. These steps are detailed below.

4.2.1 Resetting the minimum zone airflow based on CO₂ value

The first step in this process is to determine the maximum allowable CO_2 value in the zone. This can be calculated from ASHRAE Standard 62.1 using Equation 4.2 below which was derived by Taylor (2006) for steady-state CO_2 production in a zone.

$$C_{z} = C_{OA} + \frac{8400E_{z}m}{R_{p} + \frac{R_{a}A_{z}}{P_{z}}}$$
(4.2)

where C_z is the maximum allowable CO₂ concentration of the zone, C_{OA} is CO₂ concentration of the outdoor air, E_z is the zone air distribution effectiveness determined from ASHRAE Standard 62.1-2013 Table 6.2.2.2, m is the zone activity level and is determined from Figure C-2 of the Standard 62.1-2013, R_p is the occupant ventilation rate component, R_a is the building ventilation rate component and both can be determined from Standard 62.1-2013 Table 6.2.2.1 A_z is the floor area of the zone that is occupied, and P_z is the design number of occupants in the zone [27]. The maximum allowable zone CO₂ is then used to drive a simple reset algorithm that will linearly change the value for the zone minimum airflow from 0 ft³/min to the maximum design value.

The desired minimum airflow, F_z , in the zone at any point in time can then be calculated with the linear reset Equation 4.3 as shown:

$$F_z = (C_c - C_{OA}) \times \frac{F_{Min}}{C_z - C_{OA}}$$
 (4.3)

In this equation, C_c is the current CO_2 concentration of the zone and F_{min} is the design minimum airflow. The latter is calculated in the design phase of the VAV system or can be calculated using the prescriptive ventilation rate procedure of standard 62.1 [27]. The

reset algorithm runs continuously in a loop with a time delay to avoid excessive changes and slowing down of the BAS communication lines. The logic in setting the minimum airflow rate is shown in the flow chart in Figure (11) below.



Figure 11. Logic used to reset the zone minimum airflow based on the zonal CO₂ concentration

4.3 Rogue Zone Fault Detection and Diagnostics (FDD) Rules

In duct static pressure reset algorithms, the static pressure is reduced until it identifies one or more zones with open terminals. VAV boxes in which the damper is constantly fully open reduce or eliminate the energy saving potential of duct static pressure reset algorithms, thus, it is important to identify these "rogue" zones. A rogue zone may be the result of an undersized VAV box or a failure of one of two sub-systems; namely the zone thermostat or VAV Damper. The zone thermostat can fail to communicate its value to the

BAS or it can send a stale value which does not change after a considerable amount of time. An incorrect space temperature value that is not close to the zone setpoint will keep the VAV damper open trying to satisfy the zonal heating and cooling requirements. A VAV box controller can also fail to communicate its damper position to the BAS or a VAV box with a stuck damper will fail to modulate.

An algorithm for fault detection and diagnosis (FDD) was developed to identify each of these failure modes. Any VAV box identified to be in fault is excluded from the static pressure reset algorithm. A number of methods to implement a rule-based FDD in a live site were considered. The main constraint for this approach is that the FDD not affect the normal operation of the HVAC equipment, thereby jeopardizing human comfort. Additionally, implementation of this approach is limited to the sensors already deployed on a site.

4.3.1 Thermostat FDD rule and VAV box FDD rule

The thermostat FDD algorithm uses three rules to detect the three identified rogue zone failures.

Rule 1 – Thermostat communication error: If a communication failure between the thermostat and BAS is detected, the duct static pressure reset algorithm will need to be aware in order to exclude that VAV box. This failure mode is only useful for installations where a thermostat communicates directly to a BAS. Many installations will have the thermostat communicating via the VAV box controller.

Rule 2 - Thermostat reporting a continuous zero value: Some battery powered thermostats produce a zero signal when the battery is dead; the BAS receives a

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continuous value of 0°F/null value. This rule checks for 0°F/null values for a time period greater than the data collection interval *data_int* of the BAS. This rules checks the occupancy status of the zone (as inferred from the specified occupancy schedule for the zone) to ensure it is occupied in order to prevent wrong diagnosis of thermostats that are actually measuring a 0°F value at a given time.

Rule 3 – **Thermostat reporting a stale value:** When the value reported from the thermostat does not change after a considerable amount of time, *stale_int*, the value is said to be stale. Stale values can be due to many causes. The scope of this work is to identify stale values and exclude the thermostat and linked VAV box from the duct static pressure reset algorithm.

The thermostat FDD rule is summarized in a flow chart in Figure (12a) below.

The VAV box FDD algorithm uses two rules to detect two important failures.

Rule 1 - VAV box communication error: Most BAS are able to detect a communication error between a controller and the BAS. If a communication failure is detected, the SPR algorithm will need to be aware in order to exclude that VAV box.

Rule 2 - Stuck damper position: To identify a stuck damper, the damper position of the VAV box will be compared to the reported airflow value at the two extreme values of the damper (fully closed damper position and fully open damper position).

Fully Closed: VAV damper is reporting a fully closed position and a significant amount of airflow.

Fully Opened: VAV damper is reporting fully open and no significant amount of airflow can be detected.

The VAV box FDD rule is summarized in a flow chart in Figure (12b) below.



Figure 12. (a) Thermostat failure detection and diagnostic (FDD) (b) VAV damper FDD

4.4 Static Pressure Reset Algorithm

The duct static pressure reset algorithm used in this study is based on the algorithm developed in Part I.

4.5 Case Study

This study was carried out on a multiple zone recirculating VAV system that includes one AHU and 20 VAV terminal zones and serves 12,000 ft² of floor area. The AHU has a forward curved draw-through fan designed for low-pressure applications with a design total static pressure of 2.5 in. w.g. at commissioning. The fan has a design maximum airflow rate of 9,600 ft³/min at 8.87 BHP. The building is occupied from 8 AM to 5 PM weekdays. The AHU controls, VAV terminal controls and sensors are all interconnected using the ASHRAE BACnet protocol. The BAS provides continuous and automated realtime data collection and data was collected for 4 weeks at 5-minute intervals. A high density area with intermittent occupancy is the best zone for CO₂-DCV strategy as reported in [35] therefore a CO₂ sensor was installed in the largest zone in the building, which is a training/conference room. The AHU supply fan airflow used in this study is the summation of the VAV airflows measured at each VAV terminal due to a lack of airflow station on the AHU.

Tables 2-4 document respectively the assumptions and parameters used for the case study, for fault detection and diagnostics (FDD), and for static pressure reset. The parameters were obtained from Tables 6.2.2.1 and 6.2.2.2 of Standard 62.1-2013 using the conditions of the case study building. C_z was calculated from Equation (4.2) using the values in Table (2) to get 1,546 ppm. However to avoid occupant discomfort, a C_z value of 1,100 ppm was used, which is about 700 ppm above the C_{OA} as recommended in [35].

Parameter	Value	Parameter	Value
R _p	5 cfm/person	m	1.0 activity met
R _a	0.06 cfm/ft^2	Az	780 ft ²
Pz	50 people	COA	400 ppm
Ez	0.8	F _{min}	960 cfm

Table 2. Parameters to calculate maximum allowable CO₂ in zone

Table 3. Parameters for fault detection and diagnostic (FDD)

Parameter	Value	Parameter	Value
data_int	15 minutes	open_frac	10%
Stale_int	120 minutes	close_frac	50%

Table 4. Parameters for duct static pressure reset algorithm

Parameter	Value	Parameter	Value
Pos_open	90%	P _{max}	1.5 in. w.g.
N _{min}	3	ΔP	0.05 in. w.g.
N _{max}	5	t_delay	10 minutes
P _{min}	0.5 in. w.g.		

4.6 **Results**

The CO₂ based DCV developed here was applied to the main conference room of the building described in the case study section. The zone CO₂, zone minimum airflow setpoint and zone airflow are plotted against time for a typical day in Figure 13. Figure 13(a) shows the case before the application of the CO₂-DCV strategy; the minimum airflow setpoint is constant at 680 ft³/min, which was the airflow maintained in the zone for most of the day though the zone CO₂ varied from 400 ppm to 1000 ppm for that particular day. Figure 13(b) shows the case after the strategy was applied and the minimum airflow setpoint varied from 20 ft³/min to 350 ft³/min. The zone airflow was high during system startup to get the zone comfortable, but the airflow dropped to minimum airflow at about 10:30am and remained so for the rest of the day. The results

clearly show a reduction in the required minimum airflow which will result in a reduction of the ventilation air portion of the VAV system.



Figure 13. Zone minimum airflow and setpoint (a) before CO₂-DCV, (b) after CO₂-DCV

For validation of the duct static pressure algorithm, Figure (14) below shows a time plot of duct static pressure and duct static pressure setpoint for a typical day; Figure (14a) without a reset and Figure (14b) with a reset. Without duct static pressure reset, the setpoint is constant (1.5 in. w.g.) and with a reset, the setpoint changes throughout the day (0.5 in. w.g. to 0.8 in. w.g.) depending on the number of open VAV dampers in the system. In both cases, the duct static pressure is tracking the setpoint except when it is unable to keep up e.g. between 4:30am and 8:30am in Figure 14(a).



Figure 14. Duct static pressure vs setpoint (a) before DSP reset and (b) after DSP reset

In order to validate the FDD rules, an experiment was conducted to simulate a failure to see if the failure drives the static pressure reset. The static pressure setpoint was reset between 0.5 to 1.0 in. w.g. Figures (15) and (16) show time plots of the results of the test. In each case, the test was conducted over a period of three days:

- Day 1 shows the system operation without the failure and no FDD
- Day 2 shows the system operation with failure introduced but no FDD
- Day 3 shows the system operation with failure and FDD

Figure (15) shows the case of a zone thermostat failure. The first day shows a functioning thermostat correctly tracking the zone temperature and the static pressure being reset. The second day, a thermostat failure was introduced by taking the battery out of the thermostat; the zone temperature falls to 0°F creating a rogue zone which keeps the static pressure constant. During the final day, the thermostat failure continued, but the thermostat FDD algorithm was introduced which excludes the rogue zone and allows the static pressure reset to function correctly.

A similar scenario with the case of a stuck damper failure is shown in Figure (16). The stuck damper failure was simulated by overriding the damper open to 100% in the second day. The FDD was introduced in third day and the pressure reset functioned correctly.



Figure 15. Time plot of duct static pressure vs setpoint and zone temperature vs setpoint depicting thermostat failure rule



Figure 16. Time plot of duct static pressure vs setpoint and VAV damper position depicting VAV damper failure rule

AHU fan power consumption and system data were collected for 2 weeks before implementing the static pressure reset algorithm and for 2 weeks after implementing the algorithm. It was observed that the average airflow demand during the period after reset was higher than that during the period before reset; this was driven by difference in weather conditions and building load. To compare the energy savings from the reset, it is important to normalize the fan power with airflow. Normalizing the fan power with airflow will allow for the calculation of the fan energy consumed before the reset using the airflow conditions after the reset as described in Section 3.3.

After implementing the reset, the average duct static pressure was reduced from 1.30 in w.g. to 0.77 in w.g. The average power draw from the AHU fan decreased by 20% while the average required airflow to the zones increased by 3%. For the 2 weeks' period in this study, the fan energy savings were calculated to be 25% as shown in Table 5 below.

Parameter	Before Reset (02/02 to 02/13)	After Reset (03/02 to 03/13)
Average Static Pressure (in. w.g.)	1.30	0.77
Average Airflow (ft ³ /min)	6,084	6,291
Average Power Draw (kW)	6.38	5.10
Total Energy Consumption (kWh)	841	672
Adjusted Energy Consumption	899	-
(kWh)		
Energy Savings Percent (%)	-	25%

Table 5. Results from duct static pressure reset algorithm

4.7 Summary

VAV systems vary airflow rates to match zone thermal loads, and, as a consequence,

reduce fan, cooling and heating energy use below that required for CAV systems.

However, the airflow rates must also be sufficient to meet zone ventilation requirements,

and the duct static pressure must be sufficient to force air into the zone with the highest

pressure requirement.

To meet zone ventilation requirements, a minimum airflow rate is generally designated for each zone and corresponding VAV box. Traditionally, this minimum airflow rate was determined based on design conditions. However, to improve energy efficiency, the minimum airflow rate can be reset based on scheduling for zones with regular hours and using CO₂-DCV for zones with irregular occupancy like conference rooms. This study describes the implementation of a CO₂-DCV based minimum airflow control algorithm, and demonstrates its effectiveness at varying the minimum airflow to meet occupancy requirements.

One way to maintain sufficient duct static pressure is to determine the required duct static pressure based on design conditions. To reduce fan energy use, the duct static pressure can be dynamically reset based on zone damper position. However, the effectiveness of duct static pressure reset control is often compromised by rogue zones in which the damper position gets stuck at 100% open. This study describes the implementation of a rule-based FDD that is well within the capabilities of current BAS systems to increase the robustness of duct static pressure reset control. Two FDD algorithms were introduced (i) thermostat failure, (ii) VAV damper failure, and it was demonstrated that simulated failures did not compromise the static pressure reset algorithm.

Finally, a control scheme with 1) CO₂-DCV based minimum airflow, 2) a rule-based FDD to compensate for rogue zones, and 3) an advanced duct static-pressure reset algorithm was introduced into an office building. Fan energy savings of 25% were recorded compared to a system with constant static pressure.

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Part III of this dissertation will extend the logic based static pressure reset algorithm developed in Part I. A methodology that utilizes historical BAS data to drive static pressure reset is investigated.

CHAPTER 5

PART III: STATISTICALLY INFORMED STATIC PRESSURE CONTROL IN MULTIPLE-ZONE VAV SYSTEMS

5.1 Introduction

Static pressure reset was first proposed by Hartman in 1989 [13]. This approach is associated with adjustment of the supply air pressure to be just sufficient to supply the needed air to the most critical zones having fully opened VAVs. A California governmental resource offers a claim that a static pressure reset strategy, has the potential to reduce fan energy by up to 50%, reduce fan noise, vibration and bearing wear [23]. Other studies have reported savings for static pressure reset from as low as 19% [6] to as high as 60% [22]. Actual energy savings from implementing static pressure reset have been discussed in previous studies using the following approaches: comparing short term fan energy consumption before and after strategy implementation from 2 days to a few weeks [18], [31]; and comparing annual fan power consumption before and after strategy implementation [36]. Englander provided a more robust method where annual fan power consumption for pre and post strategy implementation is recorded and normalized by airflow. Englander admits that this method cannot be used for prediction [14]. Ultimately, the fan energy savings depends on climate, building usage and the size of HVAC equipment, the type of VAV boxes (dependent vs independent) and other system control strategies.

In addition to commenting upon savings, Englander and others suggested three main challenges to static pressure reset, namely:(i) rogue zones; (ii) system dynamics where the zonal system dynamics could more readily establish a hunting instability where a VAV terminal device switches back and forth between heating and cooling modes; and (iii) system diversity (huge variation in the types and sizes of VAV boxes) [11], [14], [15], [16], [17].

While static pressure reset has been established as a very useful control strategy, it is very difficult to predict the amount of fan energy savings that can be achieved when static pressure reset is implemented in a system. Theoretical savings from modulating fan capacity is cubic with the speed of the fan. The cubic savings are only approached in the case of a fan that is open (for example a cooling tower fan) [4] and it is not applicable to VAV systems. In VAV systems , the affinity laws over-predict savings by a wide margin [37]. Actual savings for VAV systems are much less in cases where the electric motor driving the fan is oversized [21]. Alternatively, ASHRAE Guideline 14-2002 later updated to ASHRAE Guideline 14-2014 which suggests use of whole building energy model simulation to predict savings from any measure. Theoretically, this approach could be used to estimate savings from static pressure reset. However, whole building simulations are expensive, not very accurate and are not able to capture the dynamics of control strategies [38].

There are no known approaches which permit Energy Services Companies (ESCOs) to estimate savings prior to implementation. This inability represents a huge barrier to

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adoption, as the building owner or user would like to understand the cost benefit to them prior to implementation. It is therefore important to establish a method to predict the amount of savings that can be achieved from static pressure reset. With such an ability:

- 1. ESCOs would be positioned to guarantee savings and payback to their customers for implementing static pressure reset; and
- 2. Certain savings could help to justify upgrade of HVAC controls from older pneumatic systems to newer DDC systems.

In this study, a statistical model is proposed for predicting energy savings before implementing static pressure reset algorithm. The method posed seeks to leverage historical data available in many commercial buildings in the Building Automation System (BAS). The model posed, through reliance on statistical analysis of the BAS data, seeks to overcome particularly the diversity challenge of static pressure reset (huge variation in the types and sizes of VAV boxes) [11]. The model is based upon establishment of a physical correlation between duct static pressure, mean VAV damper position, and system-wide volumetric airflow rate. The model is statistically based, relying upon a statistical representation of VAV damper positions at any time, and using this representation plus understanding of the pressure, airflow and VAV damper position relationship to predict the VAV position shifts. Energy savings can be then estimated for each time during the historical data period studied. Savings estimates can be extrapolated to a year if the period studied is less than a year.

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5.2 Methodology

The proposed methodology for implementing static pressure reset leverages historical BAS data. The assumption is that historical damper position during the data collection period can be used to estimate the required static pressure reduction.

The following data points are necessary:

- AHU fan power consumption
- Damper position from all VAV terminal devices
- Airflows from all VAV terminal devices
- Duct static pressure, and
- Duct static pressure setpoint

Ideally the sampling period should be 30 minutes or less and the time period studied should include heating and cooling seasons to permit estimation of future annual savings from implementing static pressure reset. BAS systems generally retain data for 1 year or more. So, generally this type of data will be available.

In considering how savings could be estimated from static pressure reset, it is first necessary to describe how this strategy may be implemented in practice. In the simplest case, a brute force approach is used, whereby the static pressure set point is arbitrarily reduced from a nominal value, say 2 in. w.g., to a lower value, say 1 in. w.g. This method is usually seasonal. The second means to yield savings from static pressure reduction is through a controllable static pressure reset methodology. This approach requires dynamically changing the static pressure setpoint in the system based on some criteria (most open damper, highest airflow request, high mean damper position, etc.). The criteria used could include any of the following algorithms:

- Adjustment of the static pressure based upon the position of the most open damper; e.g., if the most open damper position is less than 100%, then the static pressure could be reduced and the most open damper opened further.
- Adjustment of the static pressure based upon the highest airflow request from an individual damper; in this case the damper position from this zone would be set to fully open and the static pressure would be adjusted based upon airflow rate measured feedback;

None of these approaches have considered how the other VAV positions might change as the static pressure changes, and thus it has not been possible to estimate a priori the final required static pressure by the system. Actual VAV systems do not operate on a fixed system curve. The system curve which is the total system pressure as a function of airflow behaves differently depending on the location of the boxes that are modulating, the location of the static pressure sensor(s), and the static pressure control algorithm [39]. However, if the desired static pressure that could insure that one or more of the criteria above are satisfied could be estimated at each time, then it would be possible to estimate savings.

In the following sections, a method using historical data to show how changes in damper position affect pressure loss in the VAV system is described.

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5.3 System Loss Coefficient (K) and Weighted Mean DamperPosition (D̄)

An overall system loss coefficient (K) using the duct static pressure (DSP) and airflow Q through the VAV boxes is first defined, according to:

$$K = \frac{DSP}{\dot{Q}^2}$$
(5.1)

This is analogous to pressure loss coefficient across dampers but on a system level. In effect this overall system loss coefficient, accounts for the unique duct and damper characteristics for a facility, and is determinable from historic BAS data. Figure 17 below shows a plot of the K value versus the weighted mean damper position, i.e., the average damper position, \overline{D} , weighted by the size (nominal airflow) of the individual VAV boxes, *VAV_j*, at a given point in time. Equation 5.2 shows how the mean damper position is calculated.

$$\overline{D} = \frac{\sum_{i=1}^{n} (VAV_j \times VAV_{j,damper\ pos})_i}{\sum_{j=1}^{k} VAV_j}$$
(5.2)

Weighting of the VAV damper position by sizes eliminates the effect of different VAV box sizes in a system.



Figure 17. Plot of system loss coefficient vs weighted mean damper position (K- \overline{D} plot) using annual data from an actual system

The general form of the K- \overline{D} plot is a piecewise continuous function consisting of a quadratic portion where the pressure loss in the system is dominated by VAV dampers and a linear portion where the pressure loss is dominated by duct loses. The equation takes the form below:

$$f(\overline{D}) = \begin{cases} a(\overline{D}_c - \overline{D})^2 + c, & 0 < \overline{D} < \overline{D}_c \\ b(\overline{D} - \overline{D}_c) + c, & \overline{D}_c < \overline{D} < 100 \\ c & \overline{D} = \overline{D}_c \end{cases}$$
(5.3)

where a, b and c are coefficients of the slope to be determined for a particular system, and \overline{D}_{c} is the change point between the quadratic and linear portions of the plot which is also specific to a given system. Using the Heaviside function, Equation 5.3 above can be rewritten as:

$$f(\overline{D}) = a \times H(\overline{D}_c - \overline{D}) \times (\overline{D}_c - \overline{D})^2 + b \times H(\overline{D} - \overline{D}_c) \times (\overline{D} - \overline{D}_c) + c$$
(5.4)

Differentiating the Equation 5.3, an equation for the change in $K-\overline{D}$ is

$$\frac{\partial}{\partial \overline{D}} f(\overline{D}) = \frac{\partial}{\partial \overline{D}} (a \times H(\overline{D}_c - \overline{D}) \times (\overline{D}_c - \overline{D})^2 + b \times H(\overline{D} - \overline{D}_c) \times (\overline{D} - \overline{D}_c) + c)$$
(5.5)
$$\frac{\partial}{\partial \overline{D}} f(\overline{D}) = -a(\overline{D}_c - \overline{D})^2 \times \delta(\overline{D}_c - \overline{D}) + 2a(\overline{D} - \overline{D}_c) \times H(\overline{D}_c - \overline{D}) + b \times (\overline{D} - \overline{D}_c) \times \delta(\overline{D} - \overline{D}_c) + b \times H(\overline{D} - \overline{D}_c)$$
(5.6)

For every system, a small value ε can be defined such that below this value there will be no significant reduction in system pressure loss K with increased \overline{D} . Thus, ε corresponds to a small value of $\frac{\partial}{\partial \overline{D}} f(\overline{D})$. Given this condition, equation 5.6 can be rewritten as:

$$-a(\overline{D}_{c}-\overline{D})^{2} \times \delta(\overline{D}_{c}-\overline{D}) + 2a(\overline{D}-\overline{D}_{c}) \times H(\overline{D}_{c}-\overline{D}) + b \times (\overline{D}-\overline{D}_{c}) \times \delta(\overline{D}-\overline{D}_{c}) + b \times H(\overline{D}-\overline{D}_{c}) < \varepsilon$$
(5.7)

For $(\overline{D}_c = \overline{D}), \frac{\partial}{\partial \overline{D}} f(\overline{D}) = b$, corresponding to the linear portion of equation 5.4. The point $(\overline{D}_c = \overline{D})$ creates a practical limit to the reduction in pressure that can be achieved and still maintain control of the system therefore we are only interested in the solution of

$$\frac{\partial}{\partial \overline{D}} f(\overline{D}) \text{ when } (\overline{D} < \overline{D}_c)$$

The solution of Equation 5.6 when $(\overline{D} < \overline{D}_c)$, reduces to $2a(\overline{D} - \overline{D}_c)$

$$2a(\overline{D} - \overline{D}_c) < \varepsilon \quad or \quad 2a(\overline{D}_c - \overline{D}) > \varepsilon$$
(5.8)

For every system, the value of ε will be different. It can be determined by the number of dampers that can be kept open for the particular system without compromising comfort.

A study was conducted on the efficacy of the K- \overline{D} plot on different multiple zone VAV systems. Ten datasets ranging from 2 weeks to 24 months were collected from 7 different buildings and Equation 5.4 were fitted to the data. The resulting plots are presented in Appendix A and tabulated results are presented in Table 7 of Appendix A, and all the datasets show a good fit.

5.4 Distribution of Mean Damper Positions for a Given Number of Open Dampers on the K-D Plot

For every system, the value of ε will be different. It can be determined by the number of dampers that can be kept open for the particular system without compromising comfort. The K- \overline{D} plot can also be segregated into different regions matching different number of open dampers in a system as in Figure 18 below. Here it is obvious that for this system, an increasing weighted mean damper position is associated with an increasing number of dampers open. Thus, the ε condition effectively represents a limit to benefits from static pressure reduction. As the number of dampers that are nearly fully open (higher mean damper position) increases, there is a diminishing effect on the overall system loss coefficient, almost certainly due to duct losses beginning to control the system losses (rather than VAV losses). As a result, there is also a diminishing effect on the duct static pressure reduction that could be achieved with additional increase in the mean damper position. Thus, in the end, a target mean damper position at any time can be associated with this ε position.



Figure 18. K- \overline{D} plot showing different damper openings

For every time sample, a number of dampers can be considered open; these open dampers can be associated with a weighted mean for the entire system at that particular time. With a given number of open dampers, there is a clear range on where the mean of the damper can lie on the K- \overline{D} plot. For every system, there is a data distribution associated with an open damper. Although this distribution is not necessarily normal, it can be assumed to be normal for very large data set due to the independent nature of damper positions. This is especially true for pressure independent VAV systems since damper positions are driven by zone thermostat and not duct pressure or fan airflow.

5.5 At What Damper Position is a VAV Box Considered Open?

In the previous section, damper openness was discussed, however that raises the question; how open should a VAV damper be open to be considered open? An ASHRAE sponsored research study on dampers and airflow have shown that beyond 70%, at the relatively low pressures a HVAC system operates, no substantial airflow increase can be recorded. In fact, it has been shown that this ratio becomes worst for oversized VAV boxes and 25% damper opening may correlate to 99% airflow through the damper for oversized VAV boxes [40,41].

In this study, an effort was made to determine damper open position. A different approach that counted the number of open dampers based on different opening threshold values was used. Four thresholds were considered: 60% open; 70% open; 80% open and 90% open. Six months of VAV damper position data was collected at 15-minute intervals, and a count was conducted for the number of open dampers based on the thresholds. The results are shown in Figure 19 below. The frequency of the distribution for the threshold at 60% open, the frequencies were very similar. Thus, indicating choosing any open position above 70% will result in no significant loss of accuracy. In this study, the 90% open position was considered in all cases.

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Figure 19. Number of open dampers by open damper position

5.6 Energy Savings from Pressure Reset

To save energy from static pressure reset, the goal is to be able to increase the average damper position which will decrease the pressure loss in the system and ultimately reduce the fan energy consumption. Most pressure reset algorithms are designed to work by continually changing the pressure setpoint in order to keep one or more VAV dampers open. The number of VAV dampers to be kept open for a particular system will depend on a number of factors including climate zone, building type (Critical vs Non- critical e.g Hospital vs Mall), system design (VAV sizes and Duct layout), pressure sensor location (Upstream vs Downstream of the Duct) and control response (Quick vs Slow).

In this study, we want to establish fan energy savings based on the number of dampers that can be open in a system. A suitable value of ε can be selected and \overline{D} can be solved in Equation (5.8), which can then be used in conjunction with the distribution shown in Figure 18 as a guide as to how many dampers can be kept open for that particular system. To predict energy savings, the overall damper loss coefficient K and subsequently the system pressure loss associated with a particular damper opening will need to be predicted.

5.6.1 Predicting K values for new weighted mean damper position

The goal of the pressure reset algorithm at any particular time is to keep the number of dampers open such that the mean damper position will shift to a new mean damper position as shown in Figure 20 below. This new mean damper position is associated with a new improved K value which is lower than the baseline K value.



Figure 20. K- \overline{D} plot showing mean damper shift before and after duct static pressure reset for a single time sample
5.6.2 Predicting duct static pressure after reset

After calculating the values of K_{improved}, we can predict a new improved duct static pressure from Equation (5.1)

$$DSP_{improved} = K_{improved} \times \dot{Q}^2$$
(5.9)

Airflow values will be required to predict the duct static pressure using the equation above. In practice, the airflow through the VAV boxes cannot be predicted because airflow is driven by different processes including weather, occupancy, internal loads and occupant behavior. However, because we wish to use a statistical approach, it can be safe to assume the distribution of airflows in a building does not change dramatically from year to year except if the building design or usage is changed which usually requires a new air balance. If sufficient data is collected, due to the randomness of airflow requirement the airflow should theoretically approach normal distribution.

5.6.3 Predicting energy savings

To predict energy savings, the total system efficiency must be calculated by dividing the fluid power by the recorded fan power in the historical data which we will refer to as the baseline data (Power_{fan,baseline})

$$\eta_{system} = \frac{\Delta P_{baseline} \times \dot{Q}_{baseline}}{Power_{fan,baseline}}$$
(5.10)

where ΔP is total system pressure drop, \dot{Q} is system airflow rate, and η is the system efficiency.

The predicted fan power can be calculated using this system efficiency.

$$Power_{fan, predicted} = \frac{\Delta P_{predicted} \times \hat{Q}_{baseline}}{\eta_{system}}$$
(5.11)

The predicted fan energy consumption can then be calculated using the number of samples in the baseline period.

$$Energy_{fan,Predicted}(kWh) = \sum_{k=0}^{n} Power_{fan,predicted,i}(kW) \times \frac{Sampling \ rate \ (mins)}{60 \ \binom{mins}{hr}}$$
(5.12)

where n is the total number of samples.

The predicted fan energy savings can be calculated by subtracting the baseline fan energy consumption to the predicted fan energy consumption for the post-baseline case.

$$Energy Savings_{fan, predicted} = Energy_{fan, baseline} - Energy_{fan, predicted}$$
(5.13)

5.6.4 Extending short term energy savings prediction to annual energy savings prediction

The fan energy savings predicted in Equation (5.13) will only apply for the period of the collected baseline data, e.g. If only a few weeks' data was available, then the predicted savings will be for the few weeks. If annual baseline data is available, then Equation (5.13) will give predicted annual fan energy savings. However, if only a few weeks' data is available for the baseline, it can be extended to annual energy savings by multiplying by the time fraction of available data as in Equation (5.16) below.

No of samples per hour =
$$\frac{\text{Sampling Time (mins)}}{60 (mins/_{hr})}$$
 (5.14)

$$Operation Hours_{fan,baseline} = \frac{Total Number of Samples}{No of samples per hour}$$
(5.15)

(5.16)

5.7 Case Study

The data used to validate the models developed in this study was collected from 3 different buildings with both VAV and BAS systems. All the buildings are located in Southwest Ohio. All of the VAV systems were multiple zone recirculating VAV systems with a single AHU and multiple VAV terminal units. Table 6 below lists the characteristics of the buildings and data used for this study. This table identifies the type of building, its location, the floor area, the AHU schedule, total number of VAVs, the duration of historical data available from the BAS, and the data sampling period.

	Туре	Location	AHU	# of		Data Sampling
Building			Sched.	VAVs	Duration	Period
	Office	Vandalia, OH	06:00 to			
Building 1			19:00	20	1 Month	5 Minute
	High	Cedarville, OH	06:00 to			
Building 2	School		16:00	31	3 Months	15 Minute
	Elementary	Xenia, OH	Varies ¹			
Building 3	School			21	1 Month	15 Minute

Table 6. Properties of selected VAV systems for case study

The baseline condition in each of these buildings is associated with no static pressure reset. Additionally, for each of these buildings, a static pressure reset strategy based upon the trim and respond algorithm to keep a number of dampers open above 90% as

 $^{^1}$ The AHU scheduled run hours varies from 08:30 to 12:00 during summer and 05:30 to 15:30 during winter

described in Part I of this dissertation was implemented [31]. Post-implementation data was collected. The post-implementation data was analyzed to measure actual savings.

Two studies were conducted on Building 1 at 2 different time periods; each time period consisted of 2 weeks of baseline data collection and 2 weeks of post-baseline data collection. The first study was carried out during very low temperatures and the second study during warmer temperatures.

Building 2 baseline data collection was done during 2 summer months (July and August) and post-baseline data collection was done for the period of 1 month (October) for validation. The occupancy hours for both periods were the same due to reduced summer occupancy of the school. After the study, post-baseline data was collected for a one-year period to compare annual fan energy consumption with predictions.

Building 3 baseline data collection was for one month and post-baseline for one month. During the initial baseline data collection in building 3, it was observed that the system pressure was too high and no data was available at the lower end of the K- \overline{D} plot therefore the system pressure was lowered to collect data at the lower end.

5.8 Results

For validation of the methodology developed here, a pressure reset requiring 5 VAV dampers to be kept open was implemented in building 1. This number of open dampers coincided with the epsilon condition described previously. The baseline and post-baseline data were plotted on the same K- \overline{D} plot in Figure 21. The post-baseline data clearly shows a shift in mean damper position (x-axis) as the system pressure is lowered and this shift occurs along the fitted line for the baseline data. Figure 21 (a) shows results from a

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study conducted during very cold weather and Figure 21 (b) shows results from a warmer period. The cold weather study had more baseline data covering a wider range of the K- \overline{D} plot therefore there was a better data fit.



Figure 21. K-D plot showing baseline and post-baseline data for Building 1 (a) First study during cold weather (b) Second study during warm weather

For building 2 the number of open dampers coinciding with the epsilon condition was 6, as shown in Figure 22. It is important to note that the historical data for this building has

relatively low mean damper positions. Thus, there isn't good data for constructing $K-\overline{D}$ fit at high mean damper positions.



Figure 22. K-D plot showing baseline and post-baseline data for Building 2

Building 3 has an oversized VAV system and when the study began, after the first week, as apparent from Figure 23, it was observed that the overall mean damper position for the system was below 25%. Thus, the constant static pressure maintained was far too high, and there was clear opportunity for savings from static pressure reset. In order to utilize this data for this study, the constant static pressure was lowered from 1.0 in w.g. to 0.5 in w.g. However, even at 0.5 in w.g, only 1 VAV damper could be kept open without going into the linear region of the K- \overline{D} plot and losing control thereby compromising comfort.



Figure 23. K-D plot showing baseline and post-baseline data for Building 3

Using the average K value associated with a specified number of open dampers associated with the epsilon condition, the predicted static pressure needed at each time based upon the methodology described in Sections 0 was determined. The predicted improved static pressure was then compared to the actual static pressure using a static pressure reset.

The case of the first study on building 1 with 5 VAV dampers open is plotted in Figure 24 below. It is observed that the predicted ducted static pressure at the mid-point of the weighted mean damper position reflects well the actual static pressure needed to achieve comfort.



Figure 24. Building 1 Study 1: Comparison of predicted duct static pressure vs actual recorded Duct Static Pressure

The predicted static pressure was used to predict the fan energy savings associated with opening of the dampers and the results are shown in Figure 25 below. An upper and lower bound for predicted savings are also shown. The upper and lower bounds are calculated using the mean value of \overline{D} associated with the number of open dampers, and a standard deviation (*sigma*). For this study, we assume $\overline{D} - (2 \times sigma)$ to be the lower bound of the distribution and $\overline{D} + (2 \times sigma)$ to be the upper bound of the distribution. Most interesting is that the predicted energy savings in all cases mirror the actual savings, although the data from building 2 showed a significant underestimation of savings. This is due to the lack of baseline data at the lower end of the K- \overline{D} plot as shown earlier in Figure 22.

The static pressure reset implemented in building 2 was extended to one year. The recorded energy savings for the 1-year period cannot be calculated due to lack of sufficient baseline data which is only 2 months, however the predicted fan energy post-baseline was calculated to be 11,180.13kWh and the recorded post-baseline fan energy 9,650.20kWh. This result over estimates post-baseline fan energy consumption by about 16% which will under estimate savings by about 16%.



Figure 25. Predicted vs. actual recorded fan energy savings with lower and upper bound of prediction for the post-baseline period of study ranging from 2 weeks (Building 1) to 3 months (Building 3)

5.9 Summary

This part investigated a method of predicting savings from pressure reset in a VAV system using historical data. The role of VAV damper positions in relation to AHU fan power consumption was investigated which led to the development of the K- \overline{D} plot. The

plot can be easily generated with historical data and important information about the operation of a VAV system can be extracted from the plot. This methodology has been used to accurately predict DSP savings, or at least a range of savings. An ESCO could easily rely upon the lower end of the range to estimate savings.

The biggest challenge to the use of this methodology include lack of recorded fan power and the lack of enough data that falls into the lower end of the K- \overline{D} plot. If the airflow profile for the data collected in the baseline period is very different from the long time airflow profile of the system, the method will lose accuracy. Thus, application of this approach likely would coincide with some short-term testing where DSP could be changed. The new data could then be used to aid savings predictions.

CHAPTER 6

CONCLUSION

VAV systems are the most popular systems for ventilation in commercial buildings and research have shown that a VAV system when operated correctly is currently one of the most energy efficient methods of commercial building ventilation [42]. VAV systems vary airflow rates to match zone thermal loads, and, as a consequence, reduce fan, cooling and heating energy use below that required for CAV systems. However, the airflow rates must also be sufficient to meet zone ventilation requirements, and the duct static pressure must be sufficient to force air into the zone with the highest pressure requirement. Correct operation of a VAV system requires minimizing the duct static pressure required to deliver airflow to zones. Minimization of duct static pressure is achieved through static pressure reset.

The first part of this dissertation presented a practical static pressure reset control strategy that dynamically resets the duct static pressure setpoint in multiple zone VAV systems. The control strategy was based on a trim and respond algorithm that included multiple zones in the reset strategy. The new control strategy was an improvement over the widely used critical zone reset and demonstrated more stability and it is applicable to buildings with a variety of building automation systems and hardware limitations. A methodology

to calculate fan power normalized for airflow rate was also developed as a means of accurately accounting for fan energy savings. The proposed control strategy was implemented in an office building and the average observed fan power savings after implementation of the proposed control strategy was 51%, compared to a constant duct static pressure setpoint.

The second part of this dissertation extended the static pressure reset control strategy developed in the first part by including fault detection and diagnostics (FDD) to eliminate rogue zones. Rule-based FDD that is well within the capabilities of current BAS systems was developed to increase the robustness of duct static pressure reset control. Two FDD algorithms were introduced (i) thermostat failure, (ii) VAV damper failure, and it was demonstrated in an actual building via simulated failures that the FDD can prevent the failure of the static pressure reset algorithm. A CO₂-DCV based minimum airflow control algorithm was also developed, and its effectiveness at varying the minimum airflow to meet occupancy requirements was demonstrated. The improved algorithm was implemented in an office building and fan energy savings of 25% were recorded compared to a system with constant static pressure.

The final part of this dissertation investigated a method of predicting fan energy savings from static pressure reset in VAV systems using historical data. The role of VAV damper positions in relation to AHU fan power consumption was investigated which led to the development of the K- \overline{D} plot. The plot can be easily generated with historical data and important information about the operation of a VAV system can be extracted from the plot. For a system with available data at the low end of duct static pressure, this method accurately predicts energy savings. The biggest challenge to the use of this methodology

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include lack of recorded fan power and the lack of enough data that falls into the lower end of the K- \overline{D} plot. Difference in the airflow profile for the data collected in the baseline period from the long time airflow profile of the system, can also decrease the accuracy of the method.

Finally, this work demonstrates how smart algorithms can be programmed into standard BASs to enhance the energy efficiency of VAV systems while meeting ventilation requirements and handling real-world equipment failures that otherwise compromise the overall performance of VAV systems. The role of historical data in driving control strategies and how they can allow for the prediction of energy savings before implementing smart algorithms is also demonstrated.

6.1 Future Work

As more BAS systems are installed and more data becomes available, statistical methods will be the faster and easier in determining system parameters which can be easily used to drive control algorithms. Other control strategies such as Supply Air Temperature (SAT) reset, Chilled Water (CHW) reset and Hot Water (HW) reset can benefit from similar statistical analysis as demonstrated in this work. What is lost in accuracy will be more than gained in speed in trying to apply these statistical models.

Due to this similarity of fan systems to pumping systems, this work can easily be extended to cover pumping systems. The major difference is in the closed loop nature of pumping systems vs the open loop of fan systems. Better accuracy is expected from pumping systems due to the closed loop.

Pressure is an intermediate variable that is traditionally used to achieve good temperature

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control in a building. However, the advancement of controls and BASs have allowed temperature based resets to be proposed. Statistical methods should be even more relevant due to the variation in zone temperatures.

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APPENDIX A

K-D PLOT FOR DATA DISTRIBUTION

Building and Data	Number of VAVs	Duration	Data Collection rate	Fit Coefficients			R ² of fit	
				a	b	c	xc	
Bldg 1 D1	21	2 Weeks	5 Minute	0.00049	-0.001	0.135	71.44	0.92
Bldg 1 D2	21	4 Weeks	5 Minute	0.00079	-0.009	0.185	61.45	0.95
Bldg 1 D3	21	4 Weeks	5 Minute	0.00039	-0.071	0.125	80.05	0.80
Bldg 2 D1	31	6 Months	15 Minute	0.00087	-0.0087	0.177	64.54	0.92
Bldg 2 D2	31	15 Months	15 Minute	0.00114	-0.0064	0.231	63.34	0.93
Bldg 3	20	1 Month	15 Minute	0.0119	-0.0057	0.188	26.59	0.93
Bldg 4	21	1 Month	10 Minute	0.00402	-0.0069	0.242	50.21	0.62
Bldg 5	20	3 Months	15 Minute	0.00087	-0.0012	0.044	37.41	0.78
Bldg 6	6	12 Months	15 Minute	0.00064	-0.0011	0.067	51.68	0.80
Bldg 7	22	24 Months	15 Minute	0.00076	-0.0005	0.125	81.44	0.68

Table 7. K- \overline{D} plot and goodness of fit for different datasets





Figure 26. K-D plot showing distribution fit for (a)Building 1- Dataset 1 (b)Building 1-Dataset 2 (c)Building 1- Dataset 3 (d)Building 2 - Dataset 1 (e)Building 2 - Dataset 2 (f)Building 3 (g)Building 4 (h)Building 5 (i)Building 6 (j)Building 7