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Stability and Accuracy of VAV Terminal Units at Low Flow

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1. Executive Summary

The main goal of this project was to determine how low VAV boxes can be stably and accurately controlled. The lower the minimum flow setpoint for a VAV box the greater then energy savings. The stability and accuracy of a VAV box depends on two main components: the flow probe (provided by the box manufacturer), and the zone controller/pressure sensor (typically provided by a separate controls manufacturer). These components were tested separately and as an assembly to determine the contribution of each component to any stability or accuracy issues.

8 inch VAV boxes from two manufacturers (Titus and Nailor) were tested under a range of inlet pressures and damper positions. The flow probes by themselves were found to be stable and accurate under all conditions with no loss of amplification or signal quality.

Controllers from four manufacturers (Siemens, Alerton, Johnson (JCI) and ALC) were first tested under a variety of conditions to determine how stably and accurately the controller alone could measure a known velocity pressure signal. Each of the four controllers were then tested on both of the VAV boxes to test how accurately and stably the controllers could maintain a given flow setpoint while the inlet pressure was fluctuating. Stability was not an issue for any of the controllers. All controllers were able to track fluctuating inlet pressure signals and all had good filters for smoothing "noisy" pressure signals. All controllers were also able to maintain very low flow setpoints without excessive damper adjustments, even when faced with fluctuating inlet pressures.

Accuracy at very low flows was an issue for the controllers. The two controllers with hot-wire type flow sensors (Alerton and ALC) were both very accurate at the calibration points but were found to under estimate actual flow at flow rates above the lowest calibration point. Thus the controller will always err on the side of supplying a little more than the desired minimum flow at very low setpoints so there is little risk of undersupplying at minimum flow.

The pressure-based sensors (Siemens and Johnson), were highly accurate immediately after calibration but drifted over time. The Siemens controller re-zero's the sensor twice a day (by shutting the damper) and thus is highly accurate immediately after re-zeroing but can drift quickly if ambient temperature drifts. Siemens offers an optional pressure shorting bypass valve to measure the zero more frequently without disturbing the flow. This auto-zero bypass should be used for minimum flow setpoints with Siemens controls below about 0.01" (inches water column), or about 20% of design flow. The JCI controller had a software bug that caused it to go out of calibration over time. We have pointed this out to JCI and once it is fixed, reasonable accuracy can be expected with the JCI controller.

While additional research is warranted, stability and reasonable accuracy can be achieved with VAV box minimum flow setpoints as low as 0.005". For a typical VAV box this is approximately 10% of the design flow rate. While most designers are using single maximum box control sequences with minimum flows in the range of 30%-50%, some engineers have had success employing a dual maximum strategy with minimums in the range of 10% to 20%. Simulation models have shown that switching from a 30% single maximum approach to a dual maximum approach with a 20% minimum can save $0.10/ft2-yr in reheat and fan energy (0.5 kWh/ft^2-yr and 0.08 therms/ft^2-yr). Multiplied across the billions of square feet of commercial space served by VAV boxes, the energy savings could be in the millions of dollars per year in California alone. This research will give design engineers the tools and the confidence to employ lower minimum setpoints and capture some of this untapped potential for energy savings.

2. Nomenclature

VAV Box/Terminal Unit: A device that modulates the volume of air delivered to or removed from a defined space in response to an external demand. A single duct VAV box includes a flow probe and a damper and may include a reheat coil. Sometimes "VAV Box" refers only to the components supplied by the VAV box manufacturer (damper and flow probe) and sometimes it refers to the complete system of the damper, probe and controller.

Flow Probe / Flow Grid / Flow Cross / Differential Pressure Probe: A set of bore tubes with orifices that is located in the inlet duct of a VAV box. It measures the differential velocity pressure in the duct and outputs an amplified pneumatic differential pressure signal.

Flow Sensor / Pressure Sensor / Pressure Transducer: A device that accepts a pneumatic differential pressure signal and produces an analog or digital electronic differential pressure signal (e.g. 0-10 volts). A flow sensor is part of most VAV zone controllers. There are at least two types of pressure sensors:

> **Hot-Wire Type Flow Sensors**: The pressure generated by the flow grid in the VAV box induces a small flow across a hot-wire type sensor (a.k.a. hot "thermistor") in the controller. This air speed is then appropriately scaled to determine the flow rate of the VAV box. The ALC and Alerton controllers tested use this type of sensor. (Note that a hot-wire sensor can also be placed directly in the box inlet and the flow grid eliminated. This type of sensor was not tested.)
>
> **Pressure-Based Sensors**: The pressure generated by the flow probe deflects a steel diaphragm in the controller. Small changes in the diaphragm are converted to electric analog signals. The Siemens and JCI controllers tested use this type of sensor.

Zone Controller / VAV Controller: A DDC controller for controlling a VAV box. It includes a pressure sensor, A/D converter, and damper actuator.

A/D Converter: A device for converting an analog electronic signal into a digital electronic signal.

Variable Air Volume (VAV): Ventilation equipment used to control air flow, heating and cooling by varying the amount of air flow into the space.

Amplification factor (F-Factor): Ratio of flow probe output to actual value of what the probe is intended to measure. For example, a flow probe with a reading of 1.0" of pressure at an actual velocity pressure of 0.43" would have an amplification factor of 1.0/0.43 = 2.3. F may be calculated from K with the following formula:

$$F = \left(\frac{4005 * A}{K} \right)^2$$, where A is the nominal duct area in ft^2.

Some installers use the term "amplification factor" to describe the factor that they need to multiply the factory default "K" value by to match the in-situ calibration. This value would normally be about 1.0.

Flow coefficient (K-Factor): Actual flow (in ft3/min) corresponding to a flow probe output of 1" w.g. K may be calculated from F with the following formula:

$$K = \frac{4005 * A}{\sqrt{F}}$$, where A is the nominal duct area in ft^2.

K-Factor is often used in terminal unit controls to calculate actual airflow using the following equation:

$CFM = K * \sqrt{\Delta P}$, where CFM is airflow in ft^3/min and ΔP is flow probe output in inches water gauge.

CFM: Air flow measured in ft^3/min.

FPM: Air velocity in ft/min.

Inches Water Gauge ("): Differential air pressure measured in inches of water gauge or water column.**Deadband**: An area of a signal range or band where no action occurs (the system is dead). One function of a deadband is to prevent oscillation or repeated activation-deactivation cycles (called 'hunting' in proportional control systems). Deadband can be achieved by adding hysteresis within the controller. In a zone temperature control sequence, deadband is when the space temperature is between the heating and cooling setpoints (or within the throttling range) and the zone airflow rate is at the minimum flow rate and there is no reheat or recooling taking place (e.g. the hot water valve is closed). Zone controllers also typically have a built-in deadband or hysteresis to prevent excessive damper movements when the measured airflow is close to the airflow setpoint.

3. Background

The reliable control of airflow rates in VAV systems is important for a number of reasons, most significantly: acoustics, ventilation, energy management and occupant comfort. At the low end of the control range, if the airflow setpoint is below the working range of the velocity controller, the unit may cycle between closed and partially open, resulting in excessive wear on the damper motor and causing varying sound levels leading to occupant complaints. Furthermore, minimum ventilation rates demand that low-end flows be as accurate as possible to ensure that the required minimum ventilation is supplied to the zone during periods of low thermal load (Int-Hout 2003).

On the other hand, VAV box minimum air flow setpoints are often set higher than necessary, at the expense of fan energy and reheat energy. One reason is because engineers do not have the tools to determine how low VAV boxes can stably control. Based on lack of information or misinformation they often end up applying rules of thumb across the board such as 30%-50% of design flow. However, VAV boxes have been shown to stably control well below this point without compromising comfort or ventilation requirements.

Most single duct reheat boxes are controlled using a single minimum control scheme: air flow is constant at some minimum flow setpoint in deadband and in heating mode. Relatively high minimum flow setpoints (e.g. 30%-50%) are often necessary to maintain supply air temperatures below some maximum temperature (e.g. 90°F) to prevent short-circuiting in heating mode. Minimum ventilation and control stability/accuracy should also be considered in this scheme, but the maximum temperature issue is usually the driver for setting the minimum flow.

Another control scheme is to use dual maximums: in heating mode the supply air temperature is reset from minimum (e.g. 55°F) to maximum (e.g. 90°F), then the air flow is reset from minimum (e.g. 15% of cooling maximum) to heating maximum (e.g. 30% of cooling maximum). In this scheme the minimum should be determined only by ventilation requirements and control stability/accuracy. Most likely, stability/accuracy will be the driver in this scheme (Taylor and Stein 2004).

The minimum controllable setpoint is not easily determined and is in fact the subject of considerable debate in the HVAC industry. It is a function of several factors:

- the basic measurement technology employed, the design of the flow probe (amplification and accuracy)
- the quality and features of the pressure to electrical (P/E) transducer, supplied separately or embedded in the controller and when necessary
- the analog-to-digital (A/D) conversion of the flow signal at the controller.

Both output resolution and measurement precision are critical performance parameters. One controversial issue is the linearity of the flow probe amplification factor at low flow rates. The zone controller software assumes that the amplification is constant across the entire range of possible flows. Some argue that amplification decreases at low flow rates (Troyer 2005). Others argue that it is constant throughout the output range (Int-Hout 2003).

Another controversial issue is the minimum velocity pressure setpoint (VPm) at which the controller can stably control. Several controls manufacturers have said VPm can be as low as 0.004" H2O (Taylor and Stein 2004). Others in the industry do not recommend setpoints below 0.04" (Santos 2004), an order of magnitude difference.

While most designers are using single maximum strategies with minimum flows in the range of 30%-50%, some designers have had success employing a dual maximum strategy with minimums in the range of 10% to 20% of the cooling maximum. Simulation models of typical office buildings have shown that switching from a 30% minimum single maximum approach to a dual maximum approach with a 20% minimum airflow setpoint can save $0.10/ft2-yr (see section below on Energy Analysis). Multiplied across the billions of square feet of commercial space served by VAV boxes, the potential economic and environmental benefits are significant. This research will give design engineers the tools and the confidence to employ lower minimum setpoints and capture some of this untapped potential for energy savings.

4. Project Objectives

- Develop a recommendation for minimum airflow setpoint at which typical VAV boxes can stably and accurately control.
- Determine the factors that contribute to instability or inaccuracy at low flow setpoints.
- Provide a basis which further research on this subject can build upon.
- Make recommendations for future research.
- Develop test methods which can be used to test VAV boxes and controllers and calculate the lowest airflow setpoint at which a particular VAV box and controller combination can accurately and stably control.

5. Test Facility

Testing of the performance of the VAV boxes and the controllers was carried out at the Pacific Energy Center of the Pacific Gas & Electric Company in San Francisco. Two new duct branches were added to an existing system in their HVAC Classroom, and platforms were suspended from the ceiling support grid. All controller hardware, reference flow meters, and sensors were located on the platforms.

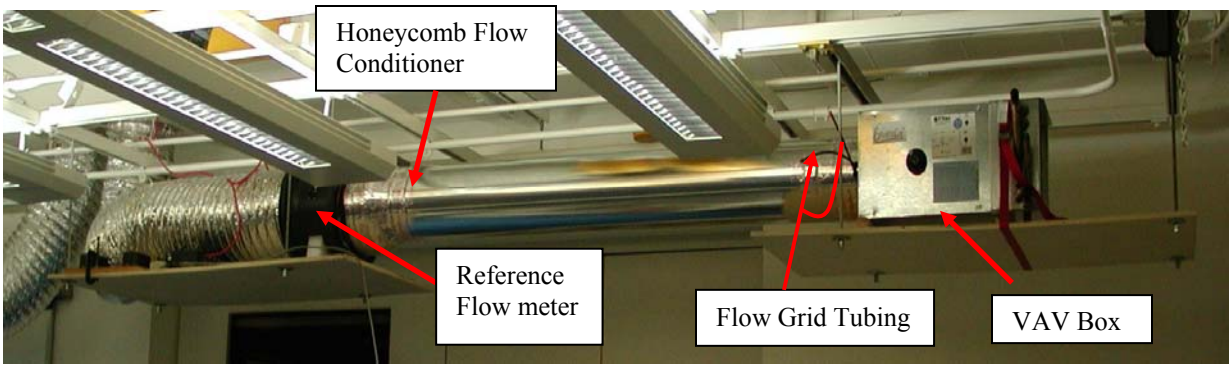


Figure 1: Photo of the Test Facility layout

The photo shows one of the new branches added to the existing HVAC system. From the left: flex duct comes down from the existing duct, across the platform and into the black "Duct Blaster" reference flow meter. From there it goes through a section of honeycomb and a reducer to a section of straight metal duct which enters the VAV unit. The static pressures at the entrance of the VAV box were controlled by means of manually adjusting the HVAC system dampers including those on the newly installed flex duct. High flows and inlet pressures required the operation of the fan built into the reference flow meter.

Information about the reference flow meter, pressure sensors, and data acquisition system and background "noise" can be found in Appendix A.

6. VAV Box Only Test Results

Eight inch VAV boxes from two manufacturers (Titus and Nailor) were tested under a variety of conditions in order to determine stability and accuracy of the amplified velocity pressure signal produced by the flow probe. Test conditions included the following:

Table 1: VAV box parameter test ranges

	Minimum	Maximum
Flow	20 CFM	700 CFM
Velocity	75 FPM	1800 FPM
Probe Signal	0.001 iwc	0.5 iwc
Inlet Duct Pressure	0.1 iwc	1.5 iwc
Damper position	Nearly closed	full open

Figure 2 shows the results from all the tests made on the Titus VAV box. Figure 3 shows a subset of that data with the Titus box damper 50% open. Other configurations and more detailed information about these tests can be found in Appendix B and C. The flow grid signal closely follows a line of constant amplification for velocities ranging from 75 to 1700 fpm. The results from all the tests made on the Nailor VAV box are shown in Figure 4.

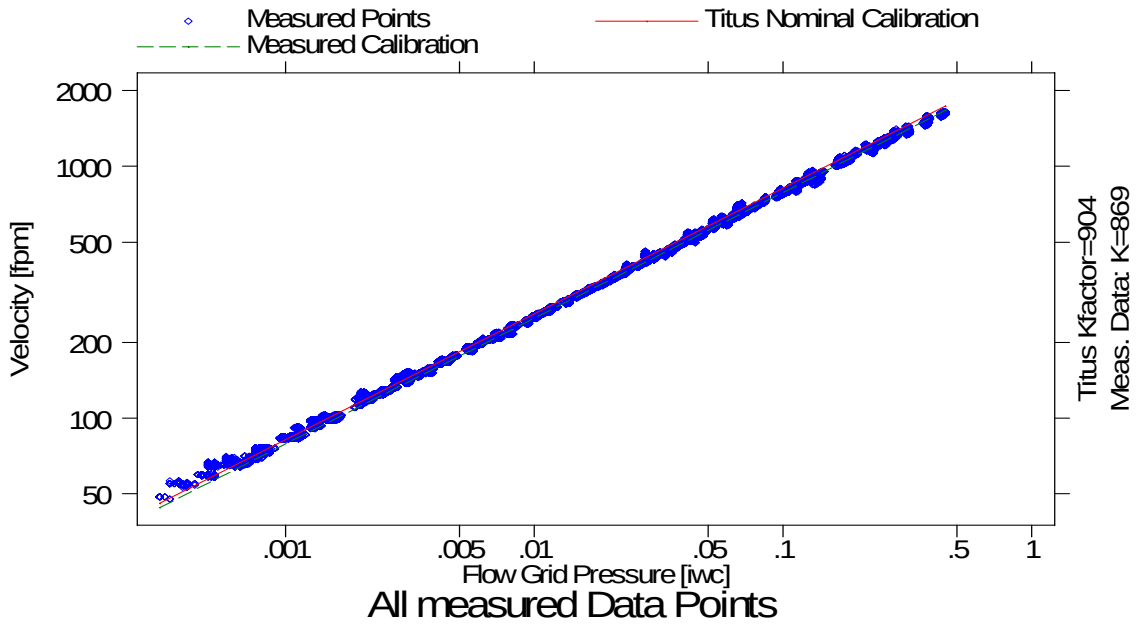


Figure 2. Calibration data for the Titus VAV box from all damper positions

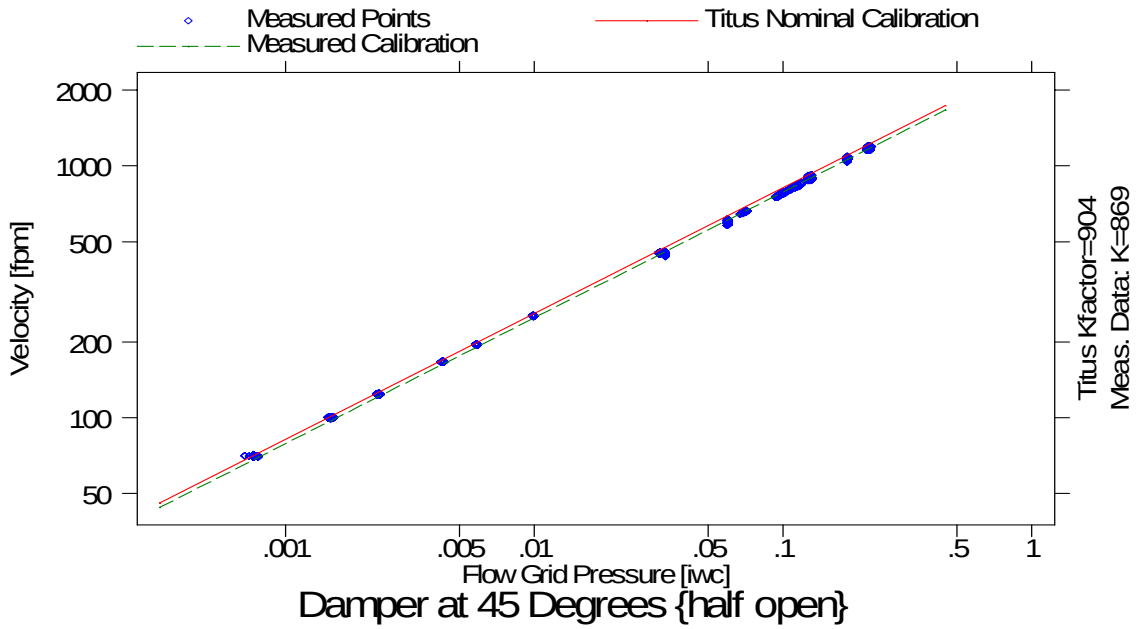


Figure 3. Sample Flow Probe Data: The Titus VAV box at 50% open

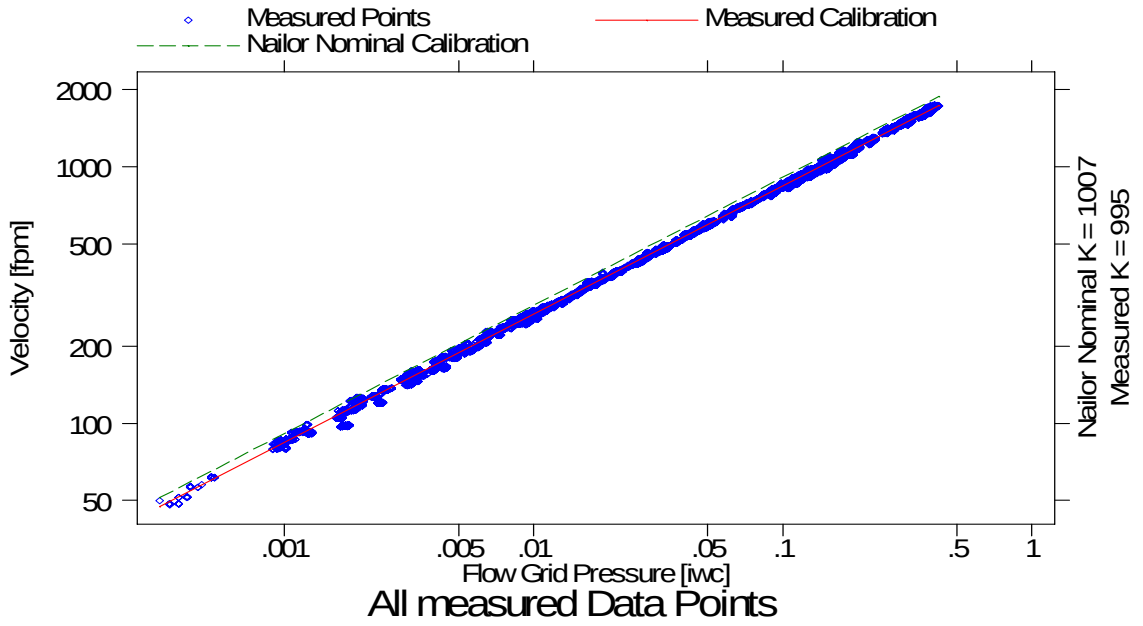


Figure 4: Calibration data for the Nailor VAV box from all damper positions

Additional figures at other damper positions for both VAV boxes can be found in Appendices B and C. These look much the same as Figure 3 with a slight deviation seen in Figure B6, for flows below 50 cfm, at a damper shaft position of 1.5 degrees for the Titus VAV box.

In summary, for the range of flows tested, about 75 to 1700 fpm, the velocity flow grid pressure amplification factor is constant and stable regardless of damper position or inlet pressure. Some controllers re-zero their pressure/velocity sensors by closing the dampers and assuming that this produces zero flow. Any leakage around the damper seals will allow some flow and thus an error in the sensor zero. The damper for the Titus box produced an excellent seal with no measurable leakage. The Nailor damper leakage was found to be 45 cfm at 1" wc or 0.002" wc on the flow grid. In normal operation, the size of this error will depend on the duct system static pressure, and thus may not be a constant. This error will cause the true flow to be higher than the flow which the controller reports.

7. Controller Only Test Results

7.1 Accuracy

Controllers from four manufacturers were tested under a variety of conditions to determine how stably and accurately the controller could measure a known velocity pressure signal. Details for each controller can be found in Appendices D, E, F, and G. Static and dynamic flow grid pressures were simulated by using pressure generating devices--either the Setra Micro-Cal pressure generator, similar manual means, or the reference flow meter fan. Reference data was collected at 2 Hz and the controller data was collected at 1 Hz. All controllers, except the ALC, used a two point in-situ calibration procedure at zero and maximum flow. The ALC controller used a 4 point calibration. The zero flow point was always determined by disconnecting the tubing to the controller and shorting it. The non-zero flow points were determined by the reference flow meter and then the controller calibration value was adjusted until the flows agreed.

The contributions to inaccuracy for each of the controllers are summarized in Table 2 and described in more detail below.

Table 2:. Contributions to the Inaccuracy of the controllers

Controller	Calibration Errors	Zero Drift	Deadband	Software Issues
Siemens (pressure-based)	Inaccurate below 0.001"; highly accurate above 0.001"	Significant temp-correlated drift (can be eliminated with optional bypass kit)	None--results in many damper movements	None
Johnson (pressure-based)	Inaccurate below 0.001"; highly accurate above 0.001"	Minimal drift	~±15 CFM (varies depending on signal noise)	Incorrectly re-zeroed the damper resulting in offset error
Alerton (hot-wire)	Highly accurate at calibration points (no flow and design flow), significantly underestimates actual flow at other values	No noticeable drift	3% of range (~15 CFM)	None
ALC (hot-wire)	Highly accurate at four calibration points with deviations between these points	No noticeable drift	±5 CFM	None

The Siemens and JCI auto-zero procedures will at times produce additional errors for dampers that do not fully seal, such as the Nailor VAV box in this study.

Figure 5 shows an example of how typical accuracy data was collected. In this case the Micro-Cal pressure generating device was used to make a series of stable pressures which were seen by the reference pressure sensor and the Siemens pressure sensor. The equivalent reference "flow"

was calculated from the "K" value at areas of stable pressure, seen in Figure 5 as green points. These are then averaged together to be compared to the flow reported by the Siemens controller.

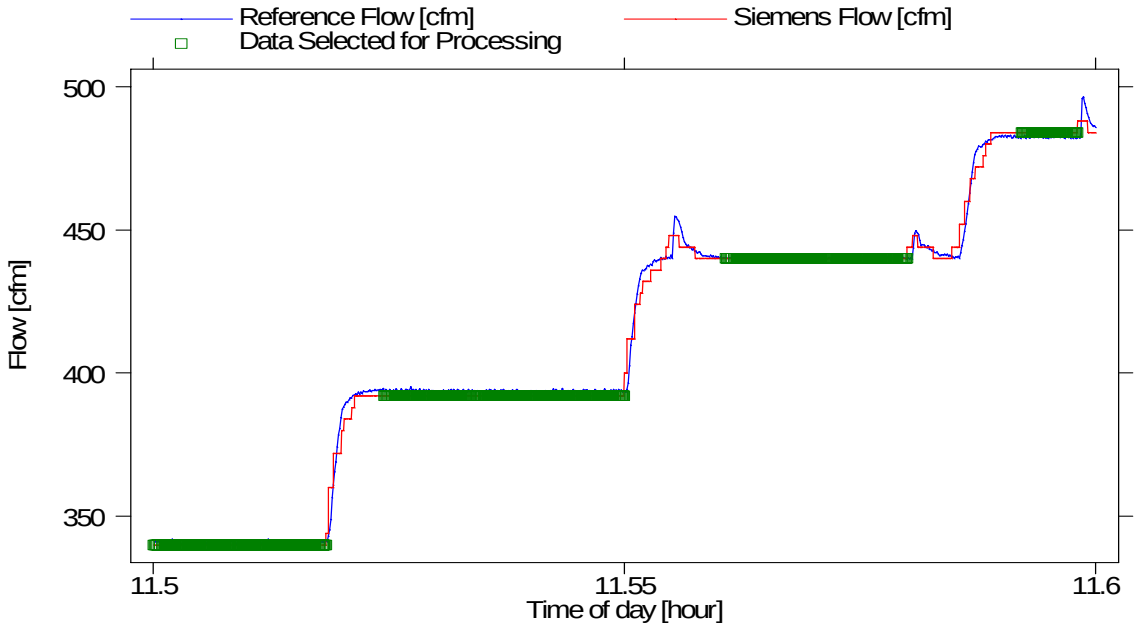


Figure 5: Typical data for determination of the accuracy of the calibration of the controller.

The controllers using a "hot-wire" type sensor use the pressure on the flow grid to produce a flow across the "hot-wire" located in the controller. The Micro-Cal is intended to be used to calibrate a pressure sensor and when attempting to make a constant pressure interprets the flow through the controller as a leak and reports an "error". For these sensors the accuracy data was generated using real flows from the reference flow meter. The pressures thus generated have more "noise" so longer averaging times were used to compensate.

The Siemens sensor (a pressure sensor) was found to be extremely accurate and stable under all conditions down to about 50 CFM (140 FPM, 0.003" signal, or about 10% of typical design flow) if the zero of the pressure sensor had been recently measured. Typically the accuracy was about ± 5 CFM. Below 50 CFM the sensor was sometimes inaccurate due to zero drift. If the sensor was recently re-zeroed then it was quite accurate, even below 50 CFM. Figure 6 shows the flow accuracy data for the Siemens controller on the Titus and NailorVAV boxes.

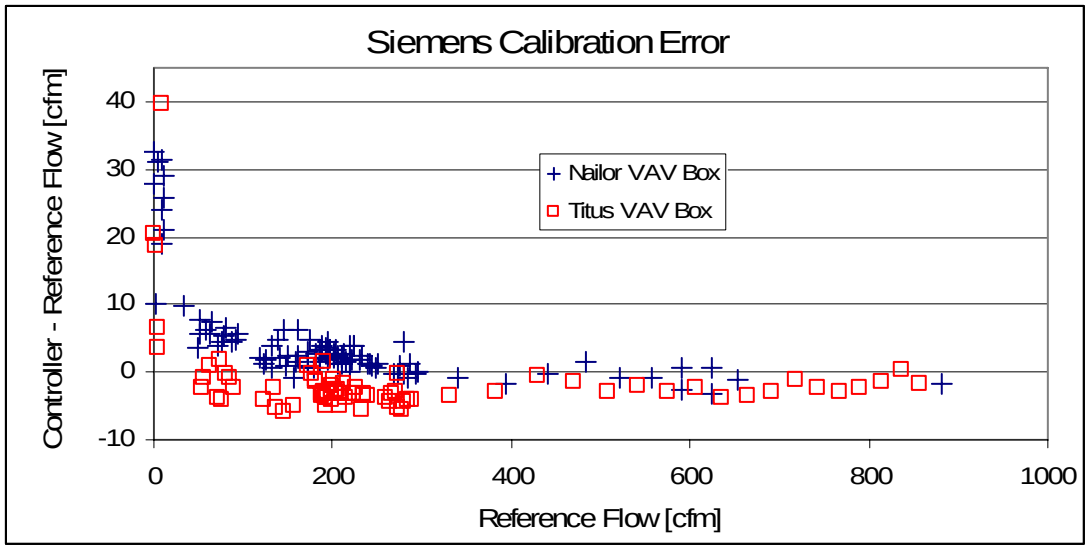


Figure 6: Accuracy of the flow determined by the Siemens controller on the Titus VAV box.

Within hours and sometimes minutes of re-zeroing the zero drift could cause the reading below 50 CFM to be off by as much as 100%. This behavior can be seen in Figure 7 where the zero is measured every 12 hours at which time the error in the pressure sensor zero resets to zero and then starts to drift again. See appendix D for the details of how this measurement was made.

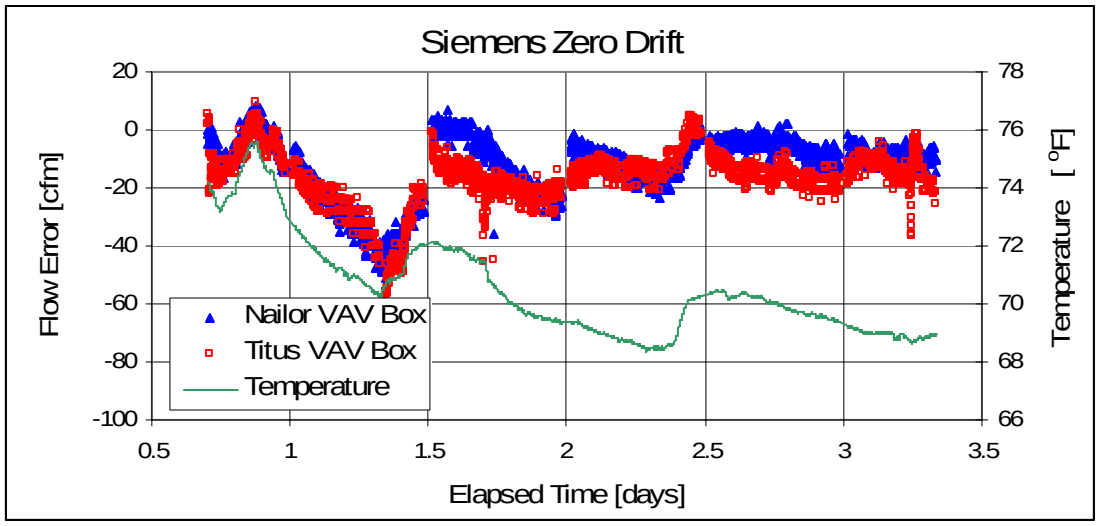


Figure 7: Zero drift of the Siemens pressure sensor.

This pressure sensor zero is seen to be closely dependent on changes in ambient temperature. Siemens offers a flow bypass kit which can rapidly re-zero the sensor several times per hour without interrupting the flow. Testing with this optional kit greatly reduced the error in the reported flow caused by zero drift.

The Alerton sensor (a hot "thermistor") was found to be stable at all flows but its calibration was not well described by a single "K" factor at all flow grid pressures. This is believed to be because of the complex non-linear nature of the response of the "hot-thermistor" sensor used in place of

the conventional pressure sensor. The calibration errors were as high as 50 cfm at about 100 and 300 cfm when the single point calibration was made at ~500 cfm, see Figure 8. Calibration errors were limited to 15 cfm at flows above 400 cfm.

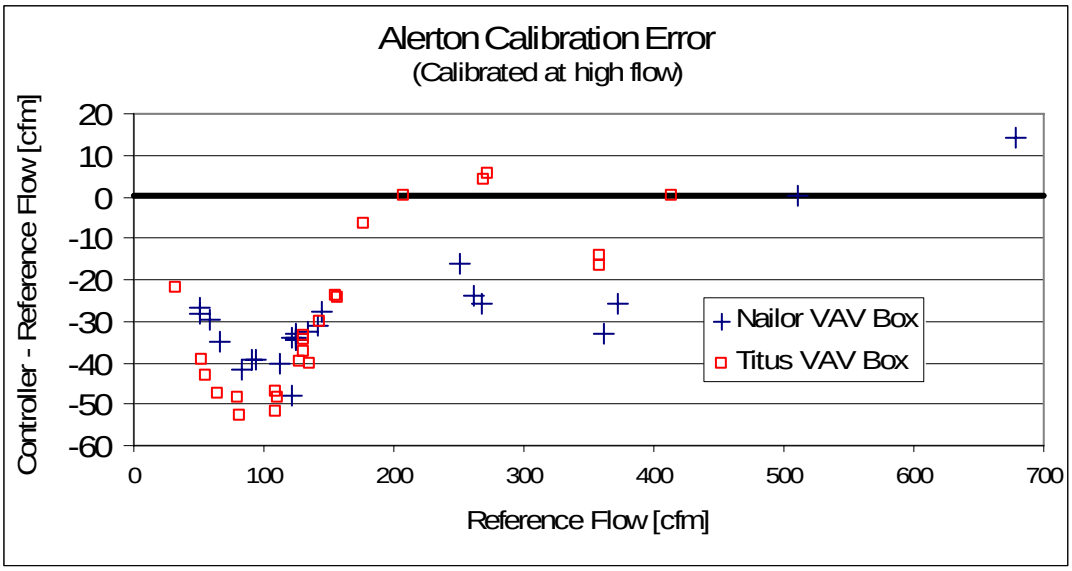


Figure 8 Calibration errors

A multiple point calibration is an option that was not studied in these measurements but should be investigated in any future research. As discussed in Appendix E, the choice of the flow used in the calibration can be an important factor in determining the accuracy of the calibration. Using a high flow results in an overall calibration that limits the error but may not be particularly good at low flows; using a single low flow for the calibration can result in large extrapolation errors due to the non-linear nature of the "hot-wire" type sensor.

Overnight testing of the Alerton controller, seen in Figure 9, showed that the error in the flow was relatively constant, indicating little drift in the zero value of the sensor.

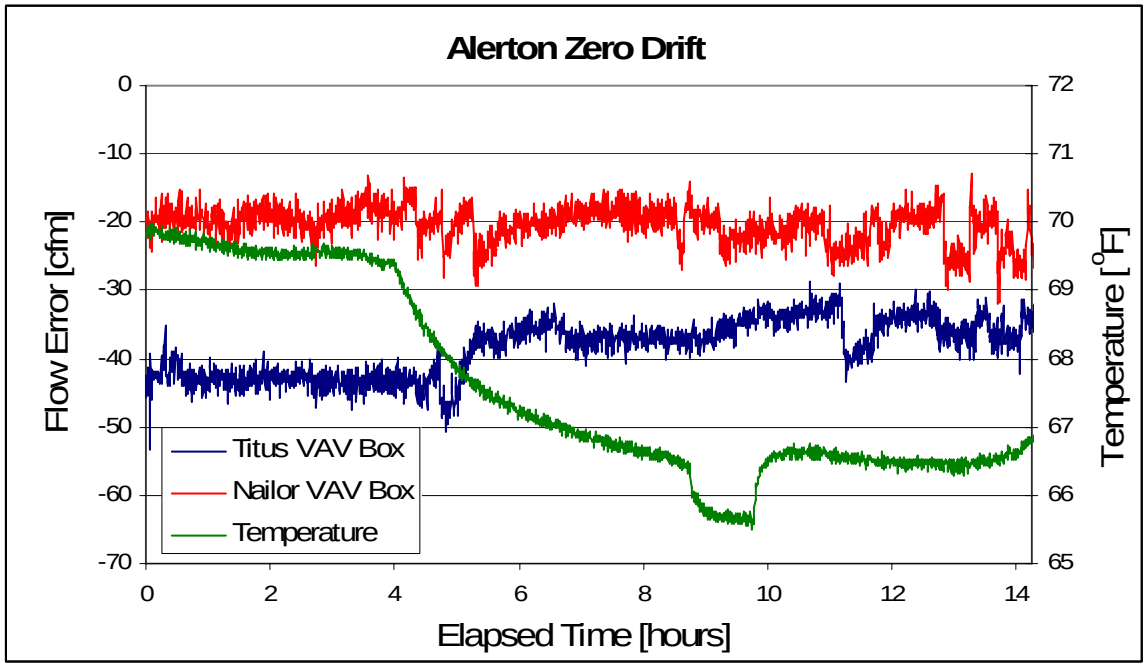


Figure 9: Flow error of the Alerton controller at 70 cfm

The Johnson Controls sensor, like the other pressure based sensor from Siemens, had a calibration error of less than 5 cfm between the maximum flow tested, about 650 cfm, and 50 cfm. The calibration errors below 50 cfm, seen in Figure 10, were larger, apparently due to a slight drift of the zero pressure.

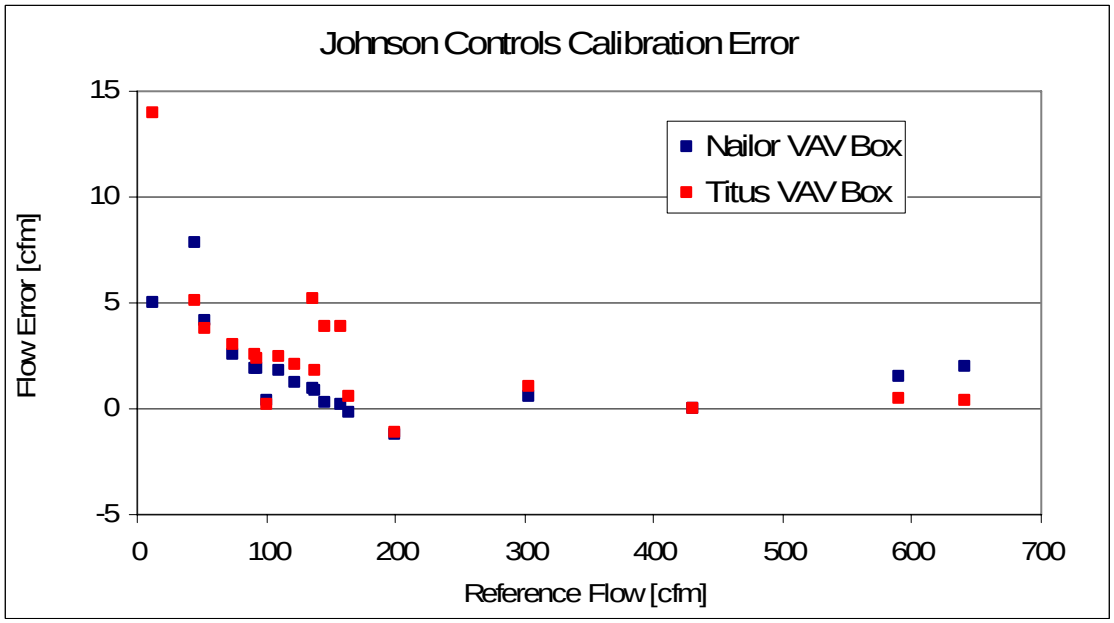


Figure 10: Calibration accuracy of the Johnson Controls controller.

Long term measurements of its zero drift indicated that it was quite stable, unlike the Siemens sensor. However the auto-zero procedure, which is automatically performed every two weeks, has an error, and the flows calculated after this procedure were off by 50% for one box, see Figure 11, and 100% on the other when the reference flow was about 50 cfm. This is seen as a serious problem but appears to be a software or firmware problem. Like Siemens, Johnson Controls offers a seldom used pressure bypass kit so that the zero may be measured frequently without interfering with the flow. It was not evaluated in this study.

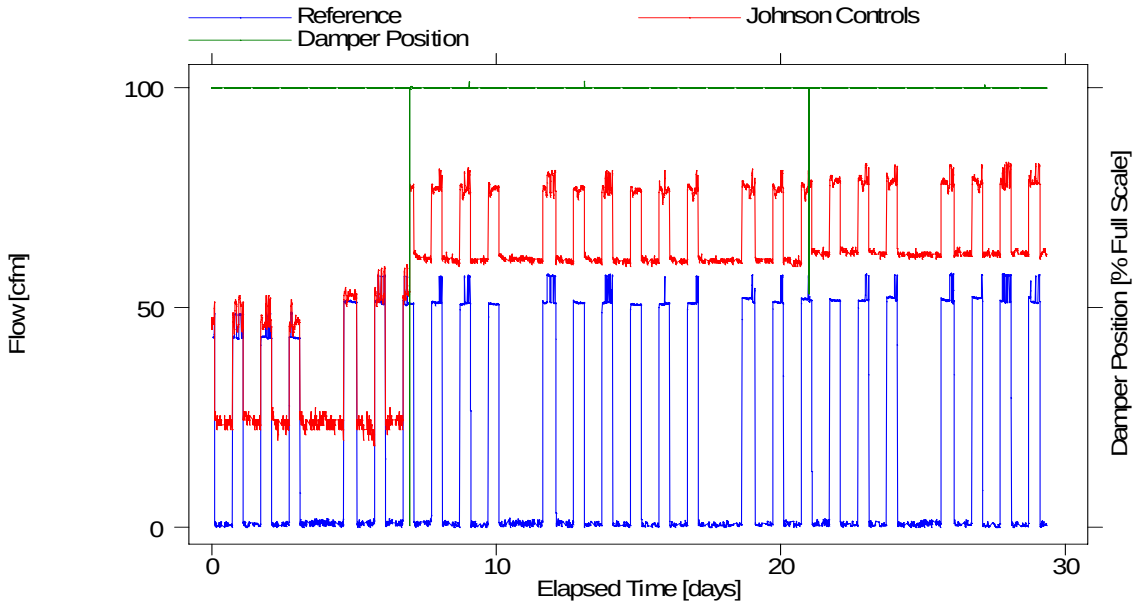


Figure 11: Reference and controller flows for a 29 day test. The spikes in the Damper Position at about 7 and 21 days indicate times when the zero of the pressure sensor is being checked.

The **Automated Logic Corporation (ALC)** sensor (a "hot-wire" type) was much like the Alerton "hot-thermistor" except that it uses a four point in-situ calibration procedure at the selected flows of 0, 75, 300 and 600 cfm. It had a stable response but using a single "K" factor for all flows resulted in calibration errors, shown in Figure 12, which were as high as 20 cfm at about 50 and 150 cfm (~midway between the calibration points). Errors of up to 30 cfm were seen at flows below 30 cfm and calibration errors were limited to 15 cfm at flows above 250 cfm.

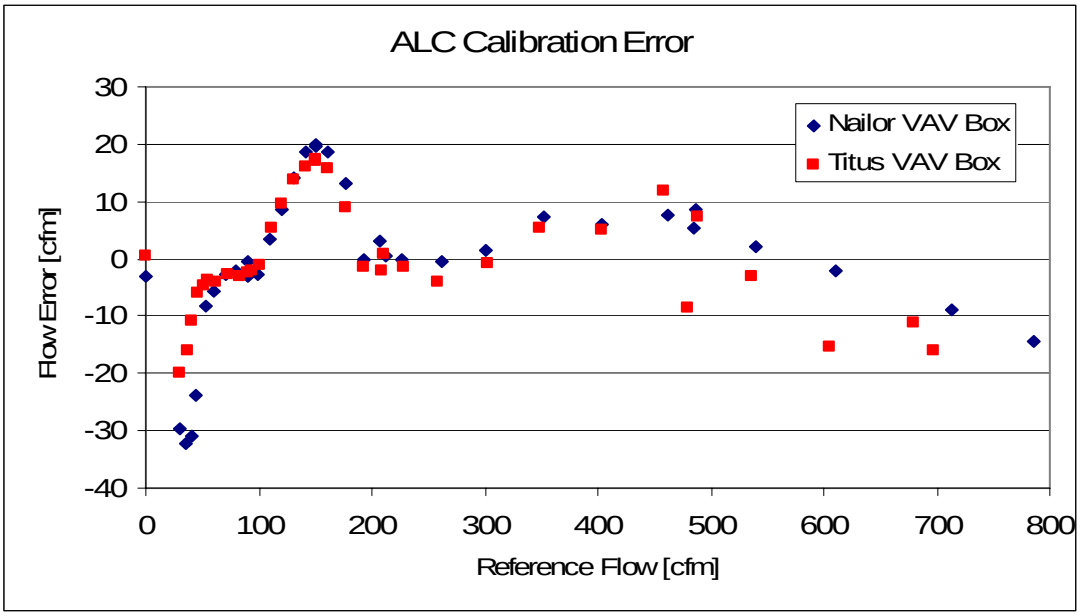


Figure 12: Calibration accuracy of the ALC controller

Overall the four point calibration procedure yielded a better calibration result than the two point procedure that was used for the Alerton controller, but was not as good as either controller that used a pressure sensor.

Unlike the other controllers, the ALC controller will report a negative flow rather than forcing a zero value. This could potentially yield a more accurate determination of the flow in situations of extremely noisy signals at very low flows where forcing "negative" flows to be zero results in a positive bias. These situations are probably rare, but these negative flows allow a direct measurement of the drift of the sensor at zero flow. Figure 13 shows that the zero was quite stable, at about negative 3 cfm, for several days. It also does not appear to have any correlation to ambient temperature as was seen with the Siemens sensor.

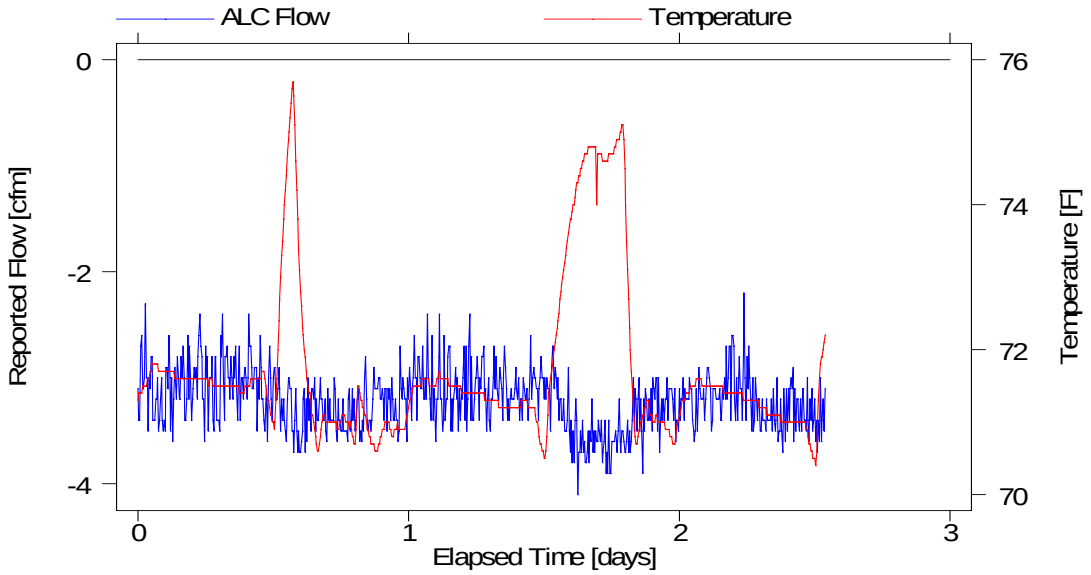


Figure 13: Zero drift of the ALC controller.

7.2 Stability

Dynamic flow grid pressures changes were made to assess the ability of the controllers to track changes in the flow. Dynamic behavior was investigated for large and small changes in the flow grid pressure at fast and slow rates of change to the flow grid pressure. All of these controllers were able to track these changes to the flow pressure signal when the dynamics used were within the normal operating ranges, consistent with the errors previously seen in their calibrations. Tests where the flow pressure cycled rapidly show a smoothed or filtered value for the controller's reported flow that is consistent with the filtered value of the reference flow meter. The filter time constant was different for different controllers, but in all cases the flow value was stabilized within 30 seconds.

Figure 14 shows an example of the kind of data generated using the Setra Micro-Cal pressure generating device. Starting from zero, it jumps to a given pressure then ramps up in steps to a higher pressure, and then back down, at which point the operator, after a small delay, can initiate another test. It could not be used for the "hot-wire' type sensors because they determine the VAV flow based on a small flow through the controller which appeared to the Micro-Cal as a leak. The maximum rate at which the pressure can be adjusted is limited by the Micro-Cal to allow for an accurate determination of the pressure with its internal sensors.

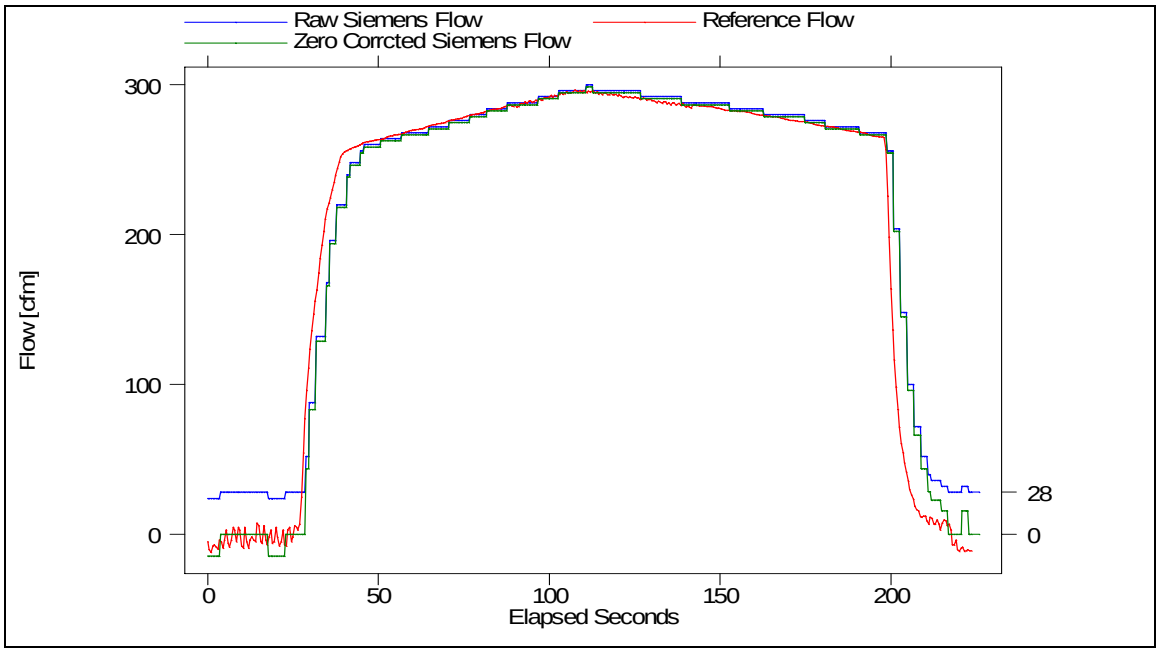


Figure 14: Controller stability data taken using the Micro-Cal pressure generator

Figure 15 shows an example of the dynamic changes generated by "manual" means. This consisted of connecting a closed end tube to the pressure sensors and pinching the tube to increase the pressure inside it. The tubing and pressure sensors in this arrangement form a closed volume of air which is very sensitive to changes in temperature, thus the drift in the "unpinched" value of the flow (pressure) seen in Figure 15.

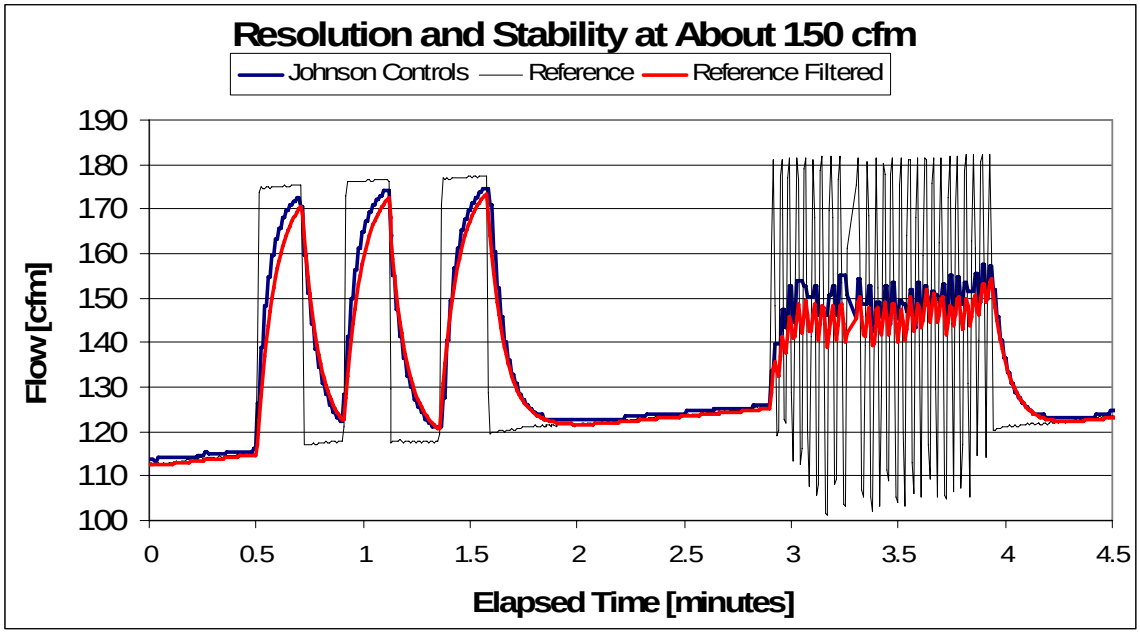


Figure 15: Stability data taken using "manual" adjustments to a system of closed tubing

Figure 16 is an example of data generated by operating the reference flow meter fan. It has the advantage of being able to make repeated cycles about a given flow (pressure) without returning to zero and is not sensitive to temperature. Both the "manual" and "reference fan" methods generate flow changes at higher frequencies than the Micro-Cal was able to produce.

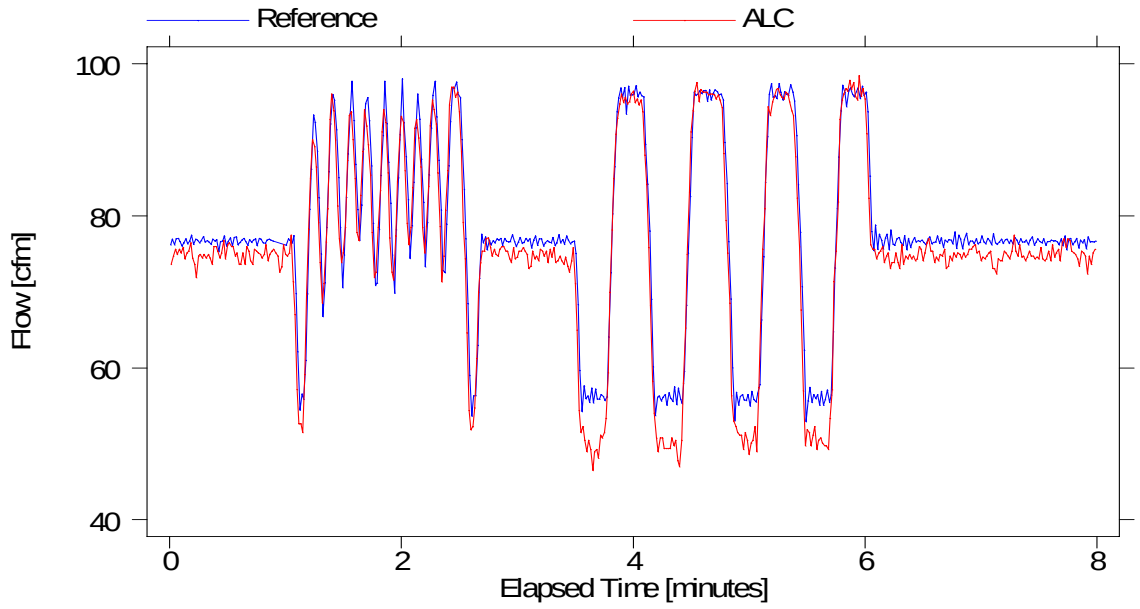


Figure 16: Stability data taken using the reference fan and associated software

The method by which the dynamic pressures were produced does not seem to affect the analysis of the performance of the controllers.

8. Controller + Box Test Results

Controllers from the four manufacturers were tested on both boxes (eight complete systems tested) under a variety of conditions to determine how stably and accurately each system could maintain a given flow setpoint and how often the damper needed adjustment. Flow setpoints from 50 to 400 CFM were tested at inlet pressures from 0.1" to 1.5". Table 3 summarizes the results for a representative sample of these tests. The results for all tests are in the Appendices.

Most controllers seek to maximize the lifetime of the damper actuator motor by minimizing its use. This is often accomplished by using some sort of dead band around the flow setpoint. If the dead band is large the flow may be significantly lower than the requested flow. On the other hand a dead band that is too small can result in oscillating flow and continual adjustment of the damper position.

All these controllers could quickly, in less than four minutes, adjust the dampers to achieve any flow setpoint of 50 cfm or greater, when used with their default dead bands. This was true for any combination of flow set point and inlet static pressure where the static pressure was high enough to produce the requested flow. The Siemens controller is the only controller that does not appear to have a deadband built into the damper control. Thus it had the most damper adjustments but was still stable (without hunting).

The **Siemens controller** produced a very stable flow with both the **Titus and the Nailor VAV boxes** under all conditions. It was able to quickly reach the flow setpoint, usually within 2 cfm, when subjected to inlet pressure changes. Figure 17 shows the Siemens controller response to changes in the inlet static pressure at a flow setpoint of 400 cfm for the Titus VAV box.

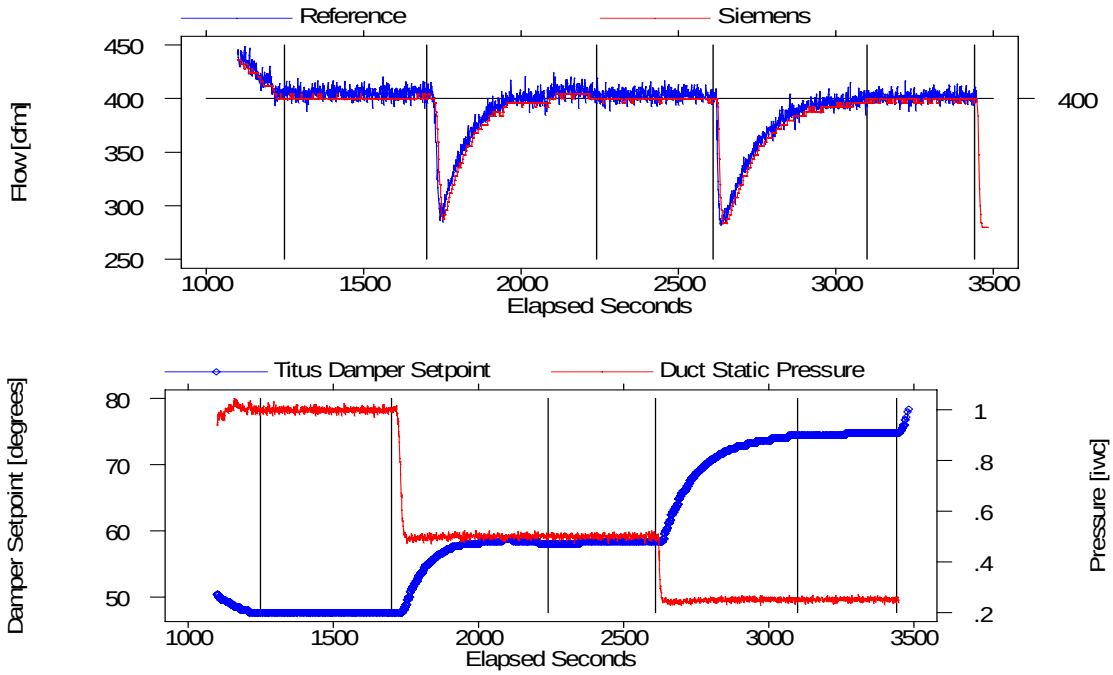


Figure 17: Siemens controller response to inlet static pressure changes at 400 cfm setpoint

Areas of stability between adjustments to the inlet static pressure were determined as seen in Figure 17 at times of approximately 1250 to 1700, 2250 to 2600, and 3100 to 3450 elapsed seconds. These periods were investigated to determine how close the flow was to the setpoint and how often the damper was adjusted.

Because it has a very narrow flow dead band, possibly none, the damper is occasionally adjusted without any apparent change in the inlet static pressure. Figure 18 shows the controller response to changes in the inlet pressure at a flow setpoint of 50 cfm for the Siemens controller on the Nailor VAV box. It took about twice as long to reach setpoint and twice as many damper adjustments with the Nailor box compared to the Titus box, Figure 19. This can be explained by the fact that the Nailor box has an opposed blade damper with 45 degrees of travel, while the Titus box has a round damper with 90 degrees of travel. The same controller has finer control when used with a damper with more travel.

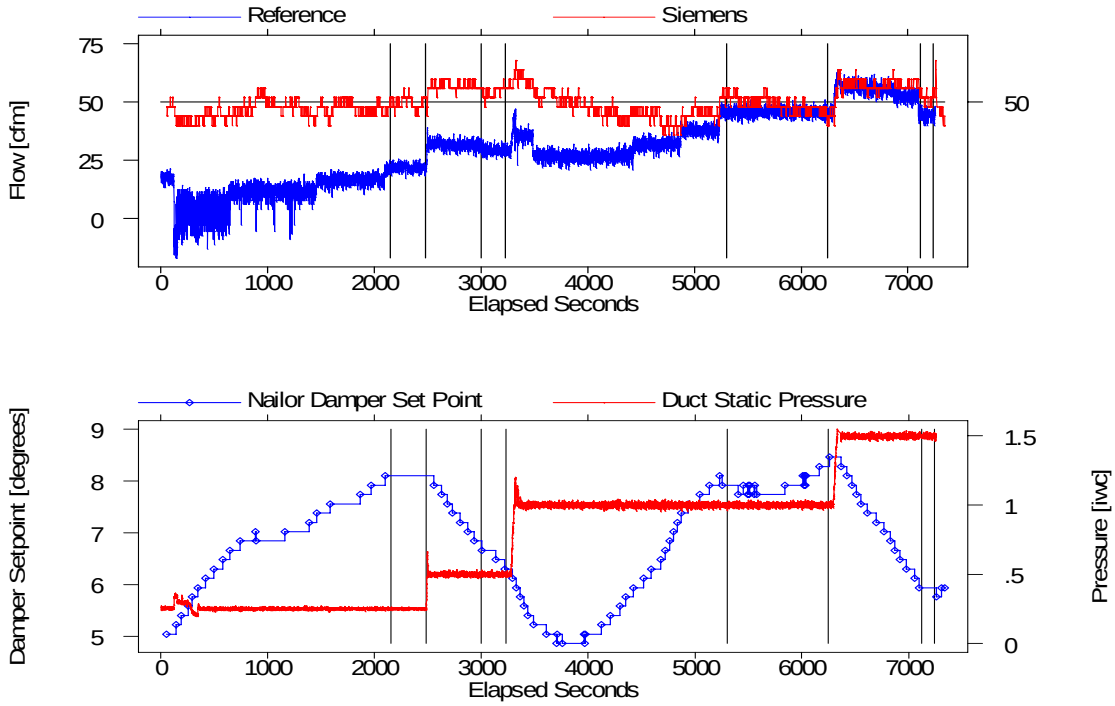
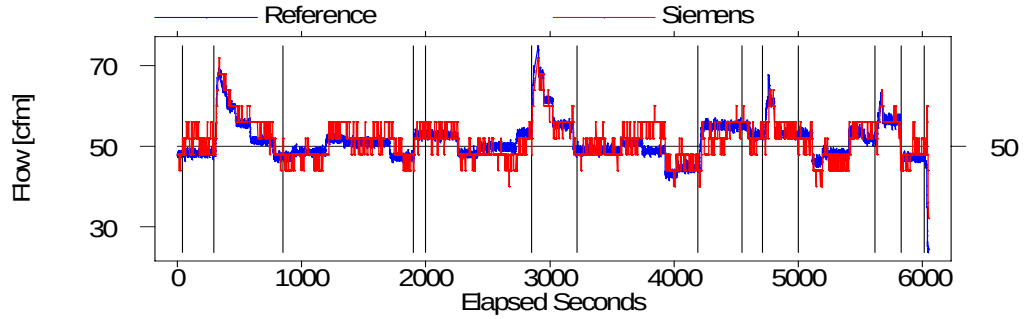


Figure 18. Siemens controller response to changes in the duct inlet static pressure at a flow set point of 50 cfm on the Nailor VAV box.

The reference flow starts out much lower than the controller determined flow because the zero value of the controller's pressure sensor had not been recently measured.



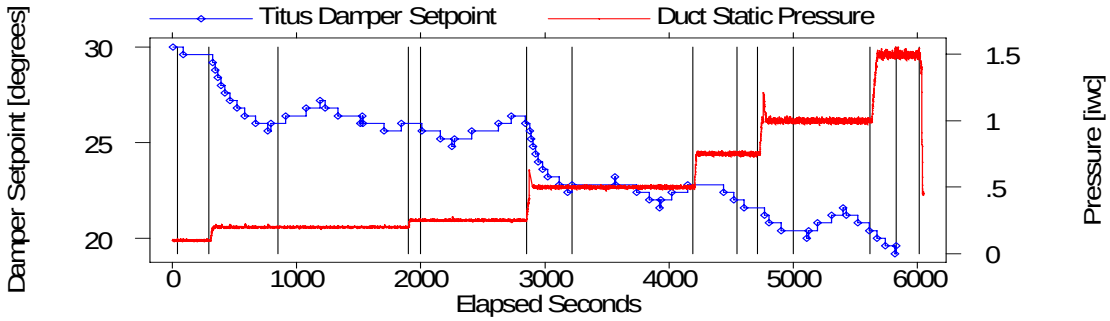


Figure 19 Siemens controller response to changes in the duct inlet static pressure at a set point of 50 cfm on the Titus VAV box.

Figure 20 shows similar data for the **Alerton controller** with the Titus VAV Box at a flow set point of 70 cfm. The initial part of the test, till about 10:40, had a dead band of ±3 cfm. The size of the dead band in the Alerton controller is equal to 3% of the maximum – minimum flow range. Thus to get a ±3 cfm dead band the maximum flow was set to 140 cfm and the minimum was 40 cfm. 140 cfm is not a typical flow rate for an 8" box and was selected just to see the impact of a smaller deadband. A flow outside of the "maximum" and "minimum" will be measured but cannot be used for the setpoint. After 10:40 the maximum was increased to 1000 cfm and the minimum lowered to 0 CFM resulting in a dead band of ±30 cfmFlows and damper position in the initial part of the test are continually being adjusted and do not settle at the flow setpoint. With the larger dead band, the damper settles to a new, constant position, and the flow is closer to the flow setpoint.

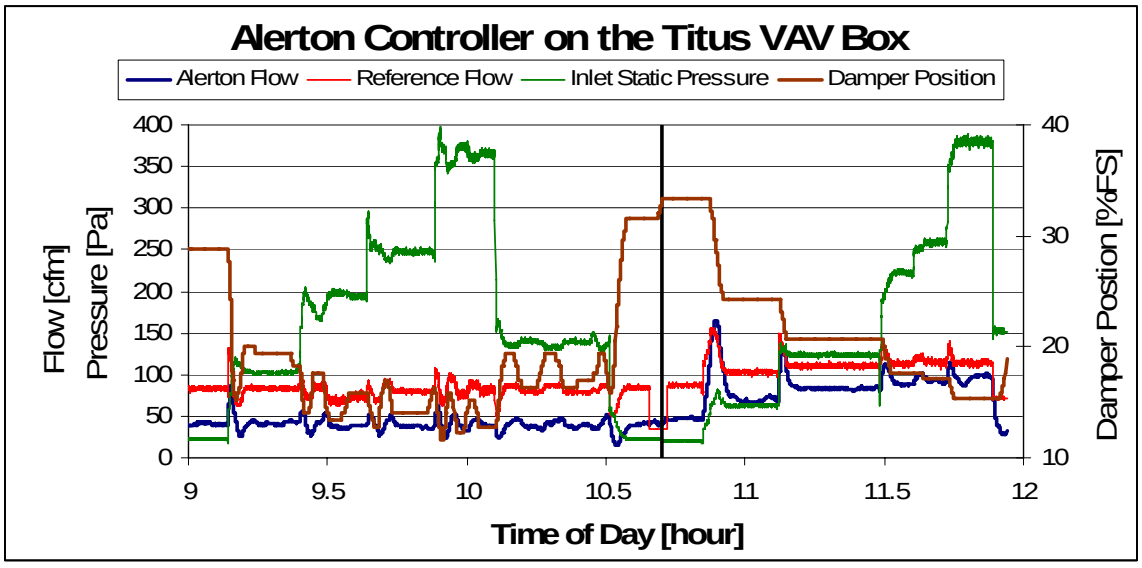


Figure 20: . Alerton controller response to changes in the duct inlet static pressure at a set point of 70 cfm on the Titus VAV box.

The dead band of the **Johnson Controls controller** is adjusted by the controller based on the amount of signal "noise". In these tests the dead band was about 15 cfm. Figure 21 shows typical

data for the Johnson Controls controller at various inlet static pressures, in this case at a setpoint of 50 cfm. Because of the dead band, the damper position was not changed even when the inlet static pressure was adjusted from 125 Pa (0.5 iwc) to 375 Pa (1.5 iwc)

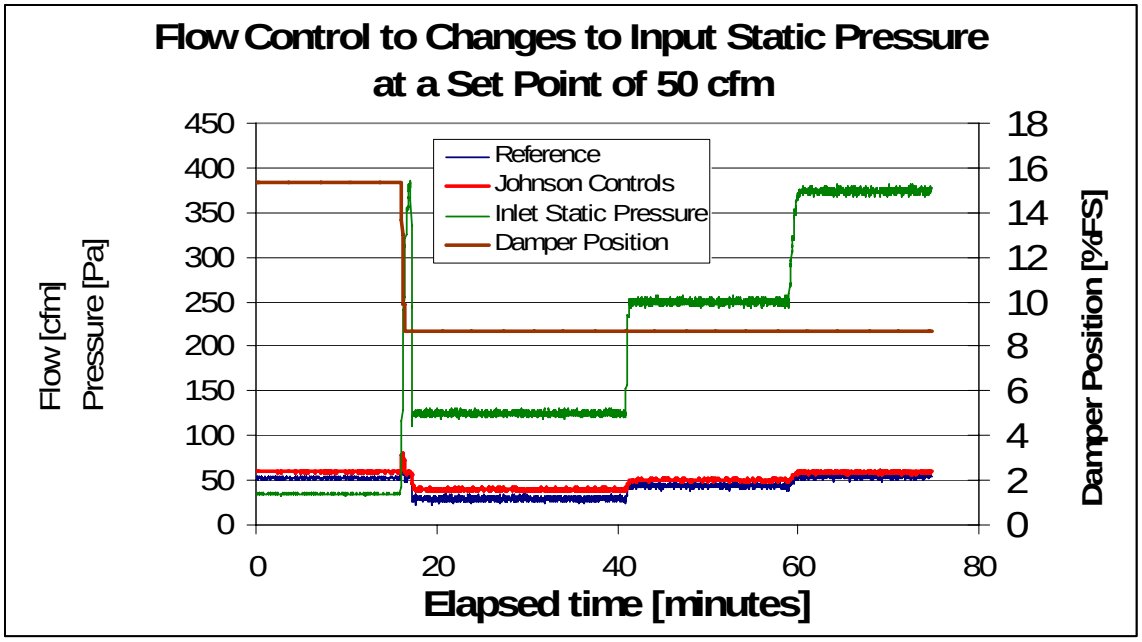


Figure 21: Johnson Controls controller response to changes in input static pressure at a flow set point of 50 cfm.

The ALC controller has a flow dead band that corresponds to one second of damper movement. Thus it is dependent on damper position and the inlet static pressure. The dead band is evaluated after every damper movement. In these tests the dead band appears to be about 5 cfm for flows under 200 cfm, it was not determined for higher flows. Figure 22 shows the response of the controller to changes in the flow setpoint. The desired inlet static pressure was 250 Pa (1.0 iwc) which the reference flow meter fan was unable to produce until the flow was reduced to 200 cfm. The data for the ALC controller was recorded every second, and in the accuracy and stability tests the recorded values changed every second. But when put into the normal operating mode, where the controller operates the damper as in these measurements, the recorded flow data remained constant for ten second blocks of time.

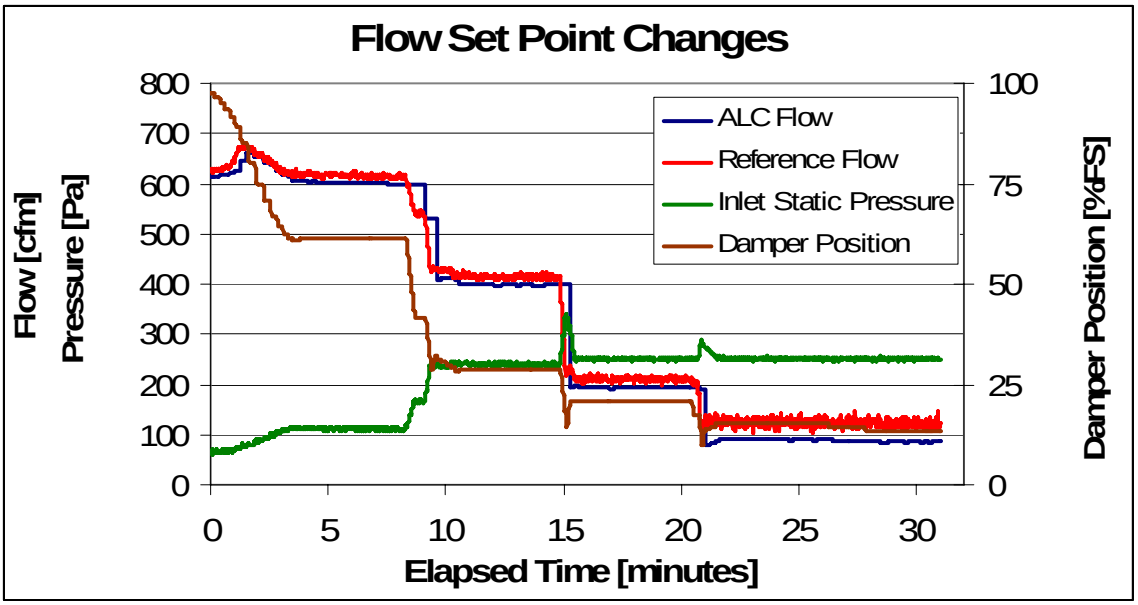


Figure 22: Controller response to a change in flow set point.

Table 3: Complete System Stability Test Results of Selected Examples					
		~ Inlet Static Pressure 0.5 iwc (125 Pa)		~ Inlet Static Pressure 1 iwc (250 Pa)	
Controller	Flow Set Point [cfm]	Controller Reported Flow [cfm]	Damper Adjustments per Hour	Controller Reported Flow [cfm]	Damper Adjustments per Hour
Siemens	50	50	31	50	47
	100	99	27	99	35
	400	400	11	400	1
Johnson Controls	50	57	0	59	0
	100	95	0	106	0
	300	294	0	na	na
Alerton	70	83	0	93	0
	150	144	0	147	0
	300	305	0	315	0
ALC	50	48	0	50	0
	75	76	0	77	0
	200	193	0	192	0

Table 3 has example stability data for the controllers. Data for many other configurations are given in the Appendices.

The accuracy of the "Damper Adjustments per Hour" is limited by the relatively short times for each of these tests, often about 5 minutes. These should be viewed as representative only.

Only the Siemens controller continued to make small adjustments to the damper position after the flow setpoint had been reached. The Siemens controller flow was also the closest to the flow setpoint. Forcing the flow dead band to be small, as might happen if the VAV box was greatly oversized, could result in unstable control.

9. Energy Analysis

VAV box minimum flow setpoints are often set at 30%-50% of design flow. There are a number of factors that determine what the "right" minimum is including:

1. Control sequence – If a single maximum sequence is used then the minimum must be high enough to prevent stratification in heating (typically 30%-50%). With a dual maximum sequence, stratification is not an issue and the minimum is determined by the other factors.
2. Ventilation requirements – Requirements can range from 5% to 50%, depending on the design cooling load and occupant density. 10% is common for perimeter zones.
3. Stability and accuracy of VAV box controls – The conclusion from this research is that under typical conditions, boxes will be stable and accurate down to about 10% flow.
4. Comfort (including "dumping") and air change effectiveness -- Conventional wisdom says that comfort cannot be maintained below about 30%. However, preliminary research shows this is not true. Additional research is required (see Recommendations for Future Work below).

It is hoped that as a result of this research design engineers will be encouraged to use dual maximum zone controls with low minimum flow setpoints resulting in significant energy savings. In order to determine the potential energy savings of dual maximum zone controls a detailed energy analysis was performed.

DOE-2.2 was used to compare the energy performance of three zone control sequences: Single Maximum, Dual Maximum with VAV Heating and Dual Maximum with Constant Volume Heating. These three sequences are depicted schematically below and described in detail in Appendix I. Basically, the minimum flow setpoint in the Single Maximum sequence is limited by the maximum discharge air temperature at which the design heating load can be satisfied. To maintain good mixing and prevent stratification the supply air temperature cannot exceed about 95°F. Thus the minimum flow setpoint in this sequence is typical limited to about 30-50% by the maximum temperature and is not limited by the minimum flow at which the box can stably and accurately control. With a dual maximum sequence the minimum flow setpoint is not limited by the discharge temperature in heating but is limited by the ventilation requirement or the controllable minimum. The Dual Maximum with VAV Heating sequence is widely used and is recommended by the authors. The Dual Maximum with Constant Volume Heating is not recommended but is included in this analysis because some engineers use this sequence and therefore it is instructive to see the energy implications.

The basecase model is a typical office building in Sacramento with a packaged VAV and hot water reheat system. This model was also run in San Francisco, Los Angeles, Chicago and Atlanta. Numerous parametric analyses were also run to determine the impact of supply air

temperature reset, of single maximum sequences with 40% and 50% minimums, of oversized zones, of systems that are left running 24/7 and of very lightly loaded buildings.

Note that a 20% minimum was used in the Dual Max simulations even though this research has shown than under typical conditions boxes should be stable and accurate down to 10%. 20% was used because this energy analysis is also being used to support a new code requirement in Title 24 and ASHRAE 90.1 requiring that minimums be no greater than 20%. Since it is a proposed code requirement, it must be sufficiently conservative so that it does not require people to do something that might not work effectively. If a designer were designing a highly noise sensitive space they might oversize the VAV box (e.g. use an 8" box for a design flow of 300 CFM.) 10% flow for an oversized box is likely to be below the setpoints recommended herein for accurate control. One option is to rephrase the code requirements in terms of inlet velocity (e.g. minimum shall be less than 200 FPM) or even probe signal (e.g. minimum shall be less than 0.005") but such a paradigm shift would require a major education campaign to make sure engineers understood it. Thus 20% was used because it is familiar to engineers and is sufficiently conservative to cover the vast majority of realistic scenarios.

In the basecase model the Dual Max-VAV saved 5 cents/ft^2-yr compared to the single maximum but the Dual Max-Constant Volume actually used 2 cents/ft^2-yr more energy than the Single Maximum case even though it has a lower flow in deadband (20% versus 30%). As shown in Figure 23, the Dual Max-VAV savings go down if supply air temperature reset is employed and go up if the zones are oversized, if the fan runs 24/7 or if the minimum flow for the Single Maximum sequence is higher than 30%. It is estimated that the average savings of the Dual Max-VAV sequence for a typical office building would be approximately 10 cents/ft^2-yr.

According to the California Energy Commission (http://www.energy.ca.gov/reports/2000-07-14_200-00-002.PDF) there are approximately 6 billion square feet of existing commercial buildings in California. Of this area, about 2 billion square feet is office and university/college. The office + univ/college sector is expected to add about 50 million square feet every year through the end of the decade. If we assume that half of existing and new buildings in these sectors are VAV systems and that 0.5% of existing VAV systems will be retrofit annually with new lower minimum setpoints and that 20% of new VAV systems will be installed with lower minimum setpoints then the penetration will be about 10 million square feet per year. At an estimated savings of \$0.10/ft^2 this comes out to \$1 million in energy savings the first year, \$2 million the second year, and \$10 million per year in year 10.

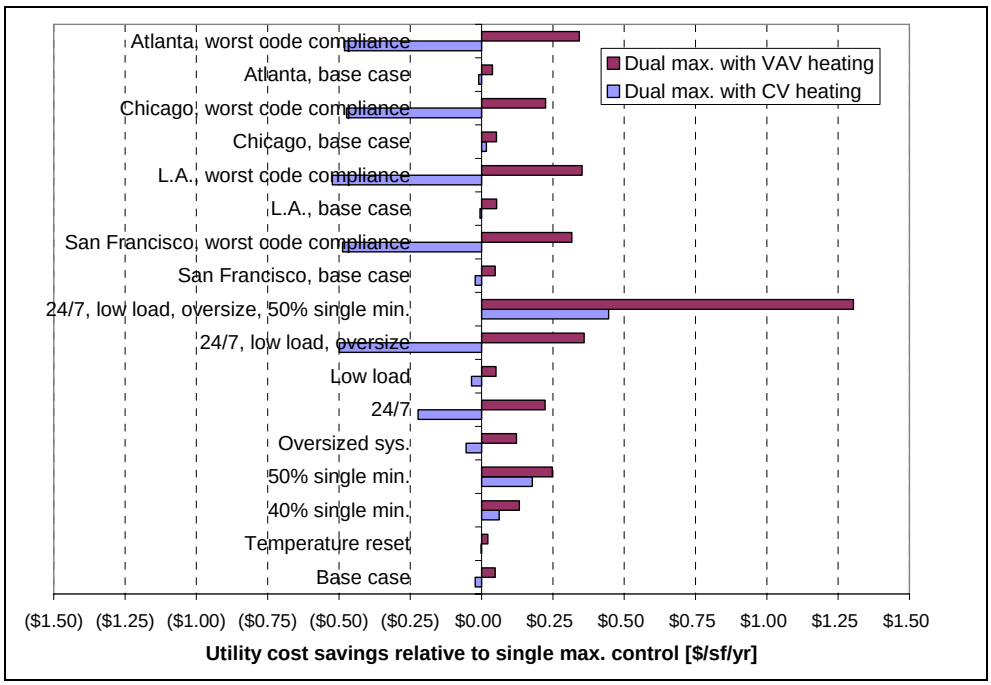


Figure 23 Annual utility cost savings

Schematics of Modeled Zone Control Sequences

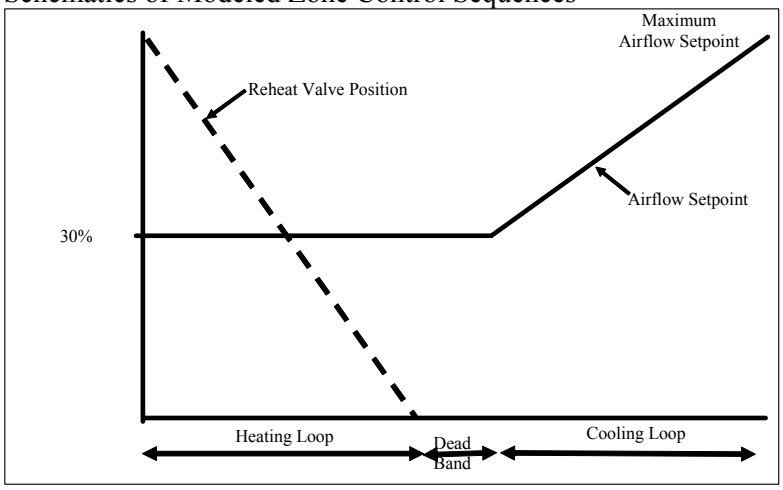


Figure 24. Single Maximum Zone Control Sequence

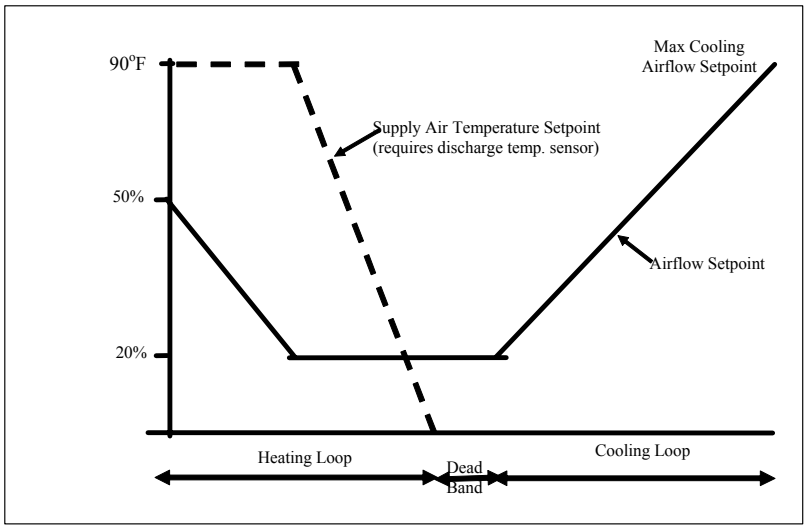


Figure 25. Dual Maximum with VAV Heating – Temperature First

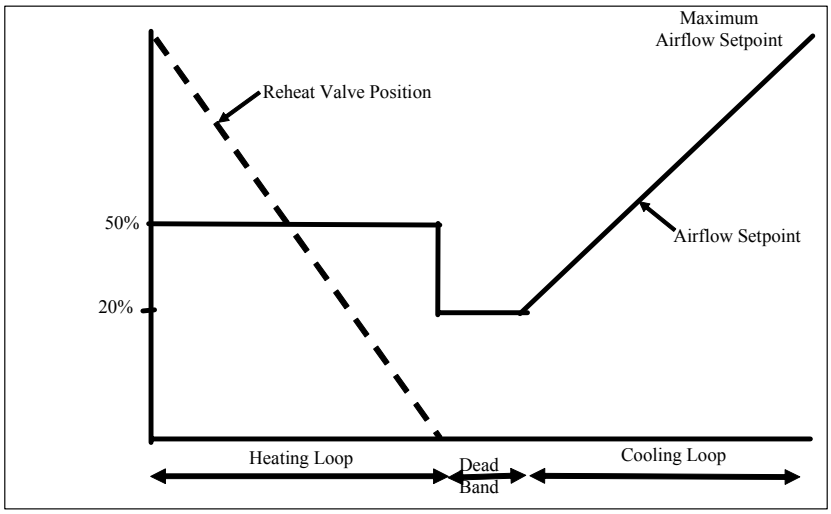


Figure 26. Dual Maximum with CV Heating

10. Conclusions

A summary of the sources of inaccuracy of the flow reported by the controllers is found below in Table 2. The following conclusions are made:

10.1.1 Flow probes in the VAV boxes are accurate and stable under all conditions, including flows down to 0.001" (85 FPM) at any damper position.

10.1.2 All controllers tested were stable at flow setpoints as low as 0.003" (140 FPM).

10.1.3 Steel diaphragm pressure sensors, such as the one in the Siemens controller, can have a zero drift within hours of re-zeroing of 0.003" or more. However the Johnson Controls controller, as the Alerton and ALC controllers, had a very stable zero.

10.1.4 The reason for the error in the zeroing procedure of the Johnson Controls controller needs to be determined. It is possible that it was an installation error, but it did not become apparent for over two weeks.

10.1.5 Tight temperature control (within 2 degrees) will reduce the zero drift seen in the Siemens controller. Conversely if temperature drifts significantly at night when the fan is off and when the sensor is re-zeroed then zero drift during the day could be greater.

10.1.6 High accuracy at low flow can be achieved with a controller such as the Siemens controller at setpoints down to:

- 0.003" (140 FPM) if an auto-zero bypass feature is installed
- 0.01" (300 FPM) without an auto-zero bypass (accurate to about 15% of reading at 300 FPM)

10.1.7 Controllers with hot-wire anemometer sensors, such as the Alerton and ALC controller, do not have significant zero drift but have significant accuracy problems at flows away from the flow at which the sensor was calibrated.

- Calibration at more than one point may improve accuracy.
- Calibration at the "minimum" set point instead of zero flow may improve accuracy, though proper in-situ measurement of this flow is unlikely

10.1.8 Large dead bands used by some controllers to reduce that amount of damper actuator usage may be too big in low flow applications. Conversely dead bands that are too small could lead to either unstable control or frequent damper adjustments.

10.1.9 Accuracy specifications are generally not available. Accuracy specifications should include the minimum and maximum velocity to be measured and maximum drift due to all sources but especially should include a temperature coefficient. Accuracy specifications should include information about long term stability with recommended recalibration intervals.

10.1.10 Based on typical VAV box selections, VAV box minimum flow setpoints of 10% of design flow will be stable and accurate.

10.1.11 Using a dual maximum zone control sequence with a 20% minimum will save about $\$0.1/ft^2$-yr compared to a single maximum sequence with a 30% minimum.

10.1.12 If dual maximum control sequences are used in only a small fraction of the VAV boxes installed every year in new construction and HVAC retrofits, millions of dollars of annual energy savings could be achieved statewide.

11. Discussion

One of the objectives of this research is to recommend test methods which can be used to test VAV boxes and controllers and calculate the lowest airflow setpoint at which a particular VAV box and controller combination can accurately and stably control. Based on this research a test method for VAV box flow probes does not appear to be necessary since the probes tested were stable and accurate under all conditions tested. One potentially significant condition that was not evaluated in this research is inlet condition. Additional research should be conducted to determine the effect of non-straight inlet conditions on probe performance.

Arriving at a good test method for VAV controllers is challenging because of all the factors that appear to affect controller stability and accuracy. These factors include:

- Ambient temperature drift (e.g. Siemens)
- Auto zero software issues (e.g. JCI)
- Auto zero frequency (e.g. 12 hrs for Siemens, 2 weeks for JCI)
- Auto zero bypass valve option (e.g. Siemens and JCI)
- Choice of calibration points (e.g. ALC, Alerton)
- Other factors not covered by this research (e.g. long term drift issues such as accumulation of dust on hot-wire sensors over months or years)

Based on this research a very preliminary controller test is described below:

1. Connect the controller to a standard commercial VAV box
2. Record the ambient temperature
3. Calibrate the controller using standard calibration procedures at max flow setpoint of 0.6" pressure signal and minimum flow setpoint of 0.01"
4. Wait for at least the longer of:
    a. 1 week
    b. 2 auto-zero cycles

    If the controller does not have an auto-zero, then the wait period will be at leas one week.
5. Perform Stability Test and Accuracy Test. The controller "passes" the test at a given minimum pressure signal if it passes both the Stability Test and the Accuracy Test at that signal.
6. Stability Test
    a. Inlet duct pressure stable at 1.5"
    b. Set the controller to the desired flow setpoint (VPsignal).
    c. Begin recording damper movements (time T)
    d. After 15 minutes (T+15) the duct pressure shall slowly fall to 0.5" over at least 10 minutes and no more than 30 minutes.
    e. Stop recording damper movements at T+60 minutes
    f. If the total number of damper movements is less than ?? then the test is passed.
7. Accuracy Test
    a. Inlet duct pressure stable at 1.0" (Perhaps min/max of 0.95 to 1.05 with a std of 0.02)
    b. Set the controller to the desired flow setpoint (VPsignal).
    c. Record actual flow rate with reference flow meter every minute (or less) for at least 12 hours.
    d. During the test period the ambient temperature must fluctuate by at least 5°F with a change of no more than 2°F per hour (to prevent someone from quickly changing the temperature and then quickly changing it back)
    e. The test is passed if all of the following are met:

i. the average actual flow is within 20% of the desired flow
ii. the average actual flow during any 60 minute period is within 40% of the desired flow.

12. Recommendations for Future Work

In order to achieve the energy savings that this research has shown are possible, there are two important areas where additional research is needed: human comfort and additional box/controller tests.

12.1 Human Comfort

Stability and accuracy are not the only concerns that engineers have when selecting the minimum flow setpoint. Another concern is comfort (including "dumping") and air change effectiveness. This is where more research is needed.

Researchers at UC Berkeley (Fred Bauman, Charlie Huizenga, Tengfang Xu, and Takashi Akimoto) did some very important research on this topic in 1995. They used a test chamber and basically they found that acceptable comfort conditions could be maintained at 25% flow. This is in sharp contrast to ADPI information in the ASHRAE Handbook and in diffuser manufacturers' literature which suggest that comfort cannot be maintain below about 30%-50% flow. Unfortunately, their research was never published.

Basically, more research is needed on this subject in order to convince engineers and diffuser manufacturers that acceptable comfort can be achieved with standard overhead VAV diffusers at 10-20% flows.

The research should include lower flow rates than the 1995 UC Berkeley work. It should also evaluate the impact of other variables such as different supply air temperatures and zone loads. In addition to lab tests, this research should also include field measurements and occupant surveys at real buildings with low minimums.

This research will be extremely valuable to engineers in terms of diffuser selection. It may also encourage diffuser manufacturers to develop new products that perform better at very low flows.

Hopefully this research will show that acceptable comfort can be maintained at 10%-20% flow, at least under certain conditions (i.e. diffuser type, supply air temperature, etc.). If so, the research could have far reaching implications in terms of getting changes made to the ASHRAE Handbook, to manufacturers' literature and to the way engineers calculate minimum flow rates. It could also lead to changes to Title 24 and ASHRAE 90.1 that would require lower minimum flow rates.

12.2 More Box/Controller Experiments

ASHRAE plans on sponsoring a continuation and expansion of this work to other inlet conditions and other equipment manufactures. Here are some recommendations for that work based on the results reported here:

- The calibration accuracy of the Alerton controller should be retested using their multi point calibration procedure. This might be done in several combinations of selected flows in order to determine the optimal set.

- The problem with the zeroing procedure of the Johnson Controls controller should be determined. If this was due to incorrect installation or setup configuration problems then a check for these during installation should become standard practice.
- Measurements over longer times, i.e. at least a month, should be planned to assess the issue of long term drift and to uncover unknown issues as was found while testing the Johnson Control controller. These should include a wider range of ambient temperatures than the current study. Introduction of pollutants (ASHRAE "dust") into the air stream should be considered to simulate long term aging.
- A definition of "excessive damper movement" and "inlet pressure stability" should be made and used to define how long the "complete system tests" need to be to assess damper movement.
- It takes lots of points to assess the accuracy of the calibration of these controllers. It is likely that at least 10 points must be evaluated from the lowest proposed setpoint and twice its value, and at least 5 points from the proposed setpoint to half its value. An additional ten to twenty roughly evenly spaced points between the maximum and twice the lowest setpoint values should also be evaluated.
- A better understanding of what pressure fluctuations are in real buildings will help in assessing the previous concern. These "pressure fluctuations" can be from real changes in the bulk flow, turbulence that is transmitted back to the flow sensor or vibrations from various sources.
- Some limits to ranges of over which these tests should encompass should be agreed on. These should include:
    - The minimum and maximum inlet static pressure to be used in the "complete system tests"; (the values of 1.5", 1", 0.5" and 0.25" WC are suggested)
    - The size and frequency of the pressure fluctuations to be used in the "stability" tests; (the "stability" test suggested in section 6 of the discussion and a second test to evaluate the controller response to "noise" that might be the largest "noise" expected as installed in buildings are suggested)
- A metric needs to be defined to asses the results of the "stability" tests. Specifically what rate of change in the flow should a controller be able to track? Is getting the correct average good enough? At what frequency? A slow response is OK and often desirable in most applications.
- It might be interesting to expand on some of the software issues seen: trend limits, both trend rate and resolution; and ability to control dead bands and other control variables. The controller software used was not designed for research yet, for the most part, worked remarkably well.
- What low flows might be used with CO2 based ventilation controls? Should these tests be conducted at these flows, or does accuracy matter at all in this case?
- The Micro-Cal was a great tool to calibrate pressure sensors, which is what it was designed for. But the Duct Blaster (reference flow meter) was used for most of these test results as the Micro-Cal could not be use with "hot-wire" type sensors. It is probably easier to use one method than two.

13. Acknowledgements

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Automated Logic Corporation: Steve Tom
Johnson Controls: John Burgess, Andrew Walton
Siemens: Dennis Thompson, Fahad Rizqi, David Scarborough
Syserco: Eddie Olivares, Robin James, Brad Leonard
Alerton: Dave Smith
Tempco Equipment (Nailor): Chuch Shane
Kruger: Dan Int-Hout
Energy Logics (Andover): Jeff Ginn
ACE-Corporation: Shad Buhlig

And especially the staff at the Pacific Energy Center and the Energy Training Center (PG&E): Ryan Stroupe, Christine Condon, Maria Arcelona, Myra Fong, Gary Fagilde, and Steve Blanc.

14. References

ASHRAE Research Project 1137, Field Performance Assessment of VAV Control Systems Before and After Commissioning, June 2004.

ASHRAE Research Project 1157, Flow Resistance and Modulating Characteristics of Control Dampers

California Energy Commission (CEC), Advanced Variable Air Volume System Design Guide, 2003

Griggs, E. I., Swim, W. B., Yoon, H. G. "Duct Velocity Profiles and the Placement of Air Control Sensors", ASHRAE Transactions 1990.

Dan Int-Hout, "VAV Box Airflow Measurements", white paper by Dan Int-Hout, Chief Engineer, Krueger , 4/17/2003, http://www.krueger-hvac.com/lit/pdf/airflow_measure.pdf

J. Jay Santos, "Common Control Problems with Pressure-Independent VAV Boxes", HPAC Engineering, October 2004.

Steve Taylor, Jeff Stein, "Sizing VAV Boxes", ASHRAE Journal, March 2004.

Appendix A: Test Facility Layout and Data Acquisition System

Test Facility Layout

Testing of the performance of the VAV boxes and the controllers was carried out at the Pacific Energy Center of the Pacific Gas & Electric company in San Francisco. Two new duct branches were added to an existing system in their HVAC Classroom, and platforms were suspended from the ceiling support grid. All controller hardware, reference flow meters, and sensors were located on the platforms.

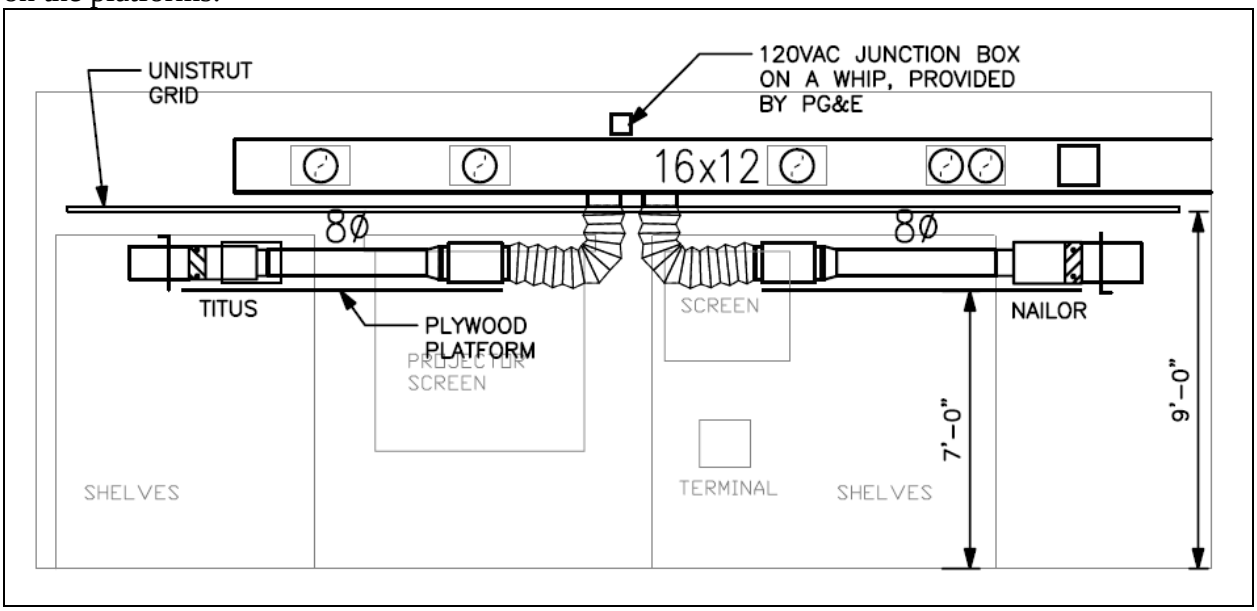

Figure A1: Drawing of the Test Facility layout

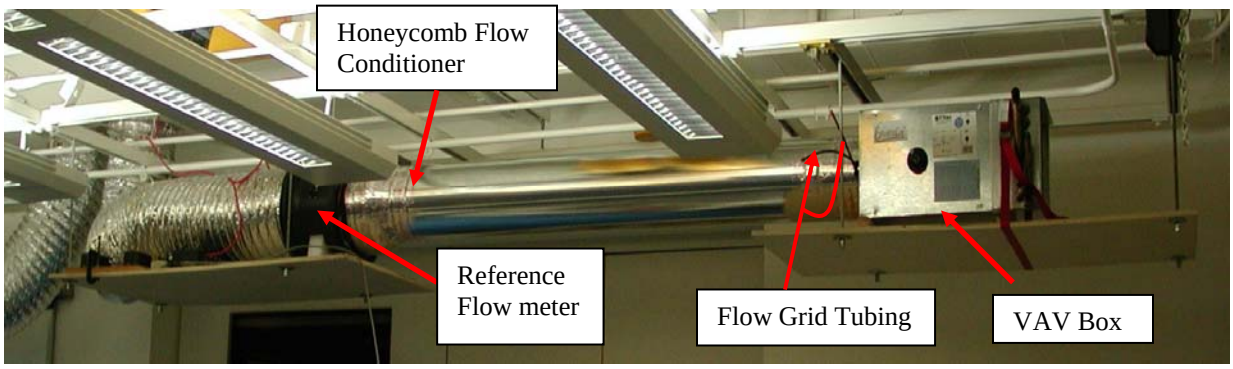

Figure A2: Photo of the test setup.

The photo shows one of the new branches added to the existing HVAC system. From the left: flex duct comes down from the existing duct, across the platform and into the black "Duct Blaster" flow meter. From there it goes through a section of honeycomb and a reducer to a section of straight metal duct which enters the VAV unit. The static pressures at the entrance of the VAV box were controlled by means of manually adjusting the HVAC system dampers including those on the newly installed flex duct. High flows and inlet pressures required the operation of the fan built into the reference flow meter.

Data collection system

The data was collected using an Automated Performance Testing (APT) unit manufactured by the Energy Conservatory. This unit has two important features that distinguish it from other data loggers. It includes eight pressure sensors that are auto-zeroed on a user selectable interval (every 10 minutes used) and it can control an associated calibrated fan, the "Duct Blaster" to maintain a constant, selectable pressure or flow. The APT also has 8 voltage input channels. Its data collection rate is a function of the number of channels and the communications to the controlling PC. Data was typically recorded at 2 Hz.

Air Flow Meter

A "Duct Blaster" was used as the reference flow meter. It has 4 flow ranges selected by inserting three different flow restrictor "rings". Together these span a range of 20 to about 1500 cfm. The four ranges are: 20 to 125, 87 to 300, 225 to 800, and 594 to 1500 cfm. The highest range can only be used if the "Duct Blaster" entrance is in free air and is pushing against a pressure of no more than 0.2 iwc. (50 Pa) this limited its usefulness. Each range has a specified accuracy of 3% of reading. A reading of 20.0 CFM, for example, indicates that the actual flow is between 19.4 and 20.6 CFM. The range in use was recorded by means of a rotary switch, which has a series of resisters as part of a voltage divider circuit. The resulting output voltage is calibrated to indicate the flow meter range 0, 1,2, or 3. Other values can be used to indicate special conditions, like #4 indicates that this data should not be used. When used in an "in-line" mode, as was done for most of these tests, the highest flow range may not be used, limiting the upper flow to about 600 cfm.

Pressure Sensor Calibrations

Two types of pressure sensors were used in these measurements, a Setra 264 sensor rated from 0 to 0.1 iwc (25 Pa) with an accuracy of 0.25% of full scale, 0.00025 iwc (0.0625 Pa), and an 8 channel APT, made by The Energy Conservatory, used in the –1.6 to 1.6 iwc range, with a rated accuracy of 1% of reading or 0.2 Pa, whichever is greater.

All pressure sensors were calibrated using the Micro-Cal provided by Setra Systems. The Micro-Cal has two reference sensors, one at +-1 iwc and another at +-0.1 iwc. Each range is accurate to 0.04% of full scale. The Micro-Cal was used in the 1 iwc range, where the accuracy is 0.0004 iwc or 0.1 Pa for the APTs and in 0.1 iwc range for the Setra 264 sensors.

Calibration with the Micro-Cal is accomplished by creating a table of requested pressures and defining a minimum length of time to maintain each requested pressure. The Micro-Cal adjusts a piston in a system of sensors and connecting tubing, to control the pressure in the system to the requested set point. In a system with leakage the piston must be continually adjusted and a warning, or failure, notice is given. Two of the APTs eight pressure channels were found to be leaking, and were not used in these measurements.

Using the Micro-Cal, each of the working APT pressure channels was calibrated. The resulting accuracy is better than the factory specifications, with only a few reading being more than 0.1 Pa different from the Micro-Cal. The resolution of the APT is 0.0004 iwc (0.1 Pa). It should be noted that the calibration adjustments were about 1%.

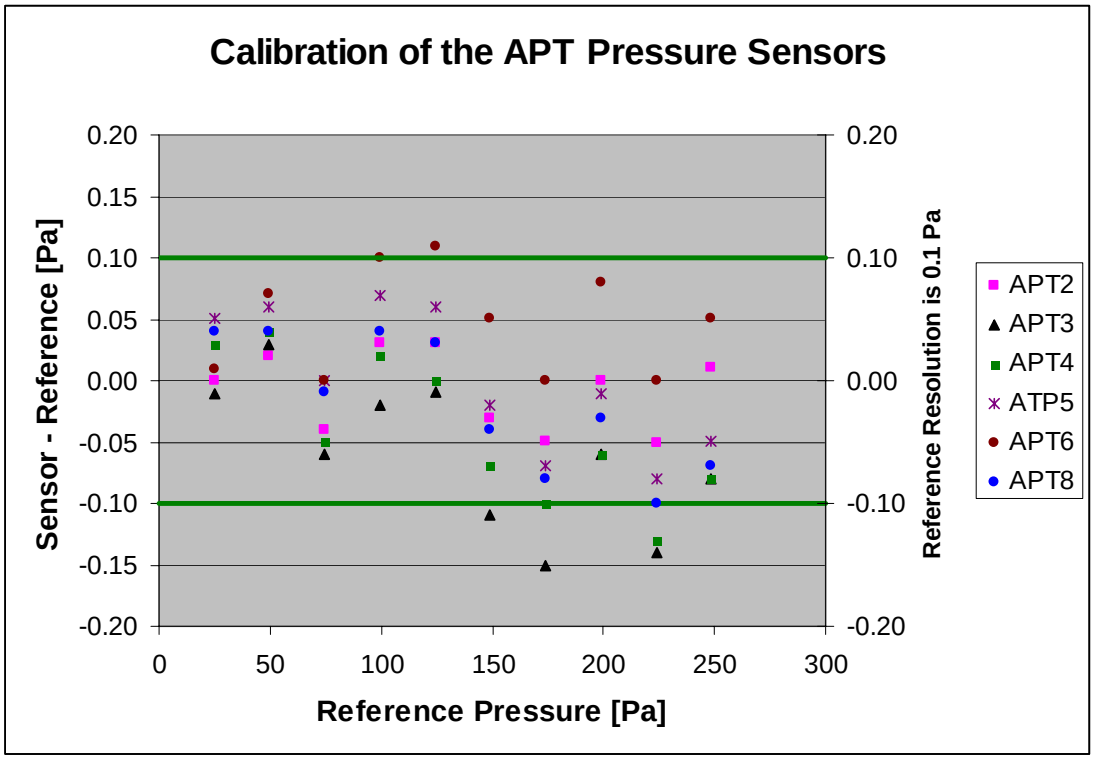

Figure A3: The residual error in the APT pressure sensor calibration is almost entirely within the resolution bounds of the Reference meter.

The Setra 264 sensors have a voltage output of just over 5 VDC. The APT data logger also has 8 DC voltage input channels. These voltage input channels are limited to 4 VDC max so the Setra sensors were used with a voltage divider circuit to reduce the full-scale voltage. The two Setra 264 sensors were then calibrated with the Micro-Cal. These, as their specifications imply, show an accuracy of better than 0.1 Pa through out their range. The resolution of the Setra sensors is 0.00002 iwc (0.005 Pa).

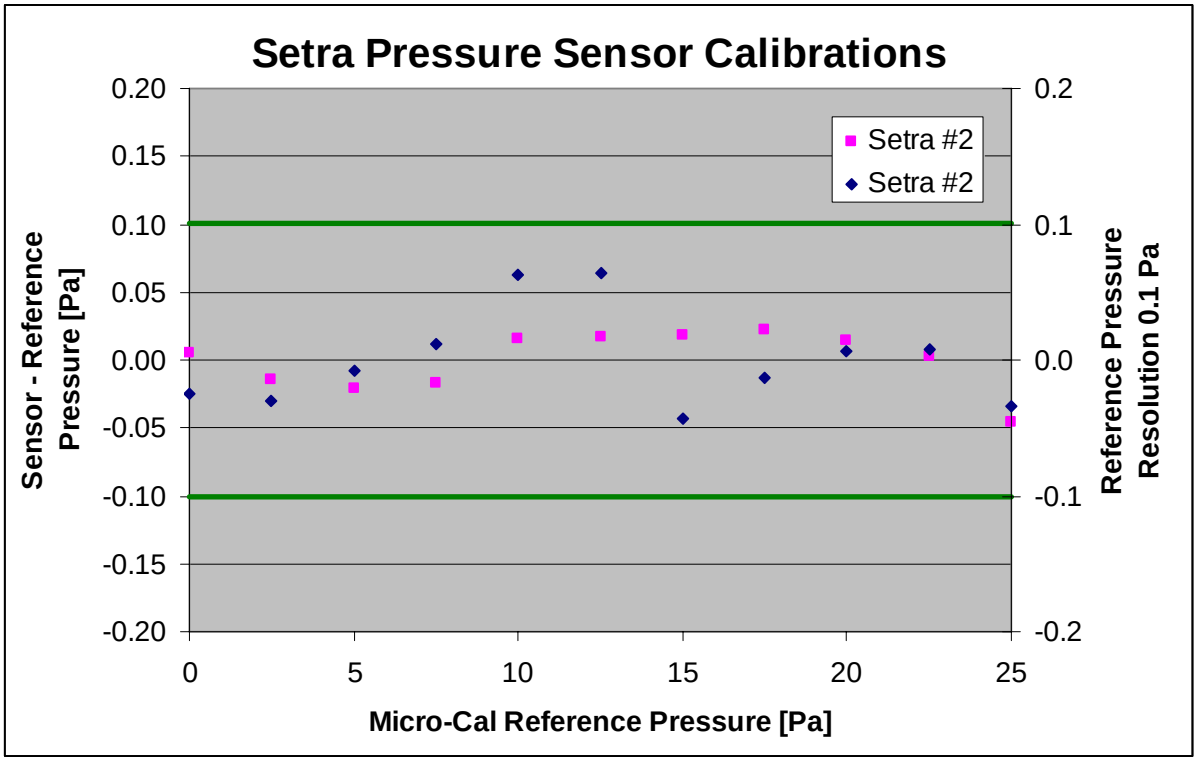

Figure A4: The residual error in the Setra 264 pressure sensor calibration.

Pressure Sensor Assignments

One Setra 264 and one APT channel were connected to the flow grid of each VAV. For all analysis the Setra was used when it was in range. An APT channel was used to measure the static pressure just upstream of each VAV and to measure the pressure on the "Duct Blaster" flow meter. A "leaky" APT channel was used to monitor the duct system static pressure at the point where the two duct takeoffs for the VAVs join with the main duct system. This pressure is not used in any calculation and it is believed that the "leak" in this sensor is not likely to significantly impact the pressure that it measures.

Damper Position Sensors

The VAV damper positions were measured by causing a potentiometer to rotate as the damper shaft rotated. The resulting change in the resistance of the potentiometer was used in a voltage divider circuit to create a changing voltage which was then logged by the APT. Calibration and repeatability measurements were made by repeated cycling of the dampers through their operating range. These measurements indicate that the position could be reproduced within 1 degree. A check of the "zero" position was made before each measurement. It should be noted that the Nailor damper shaft operates in a 45 degree range, while the Titus operates over a 90 degree range.

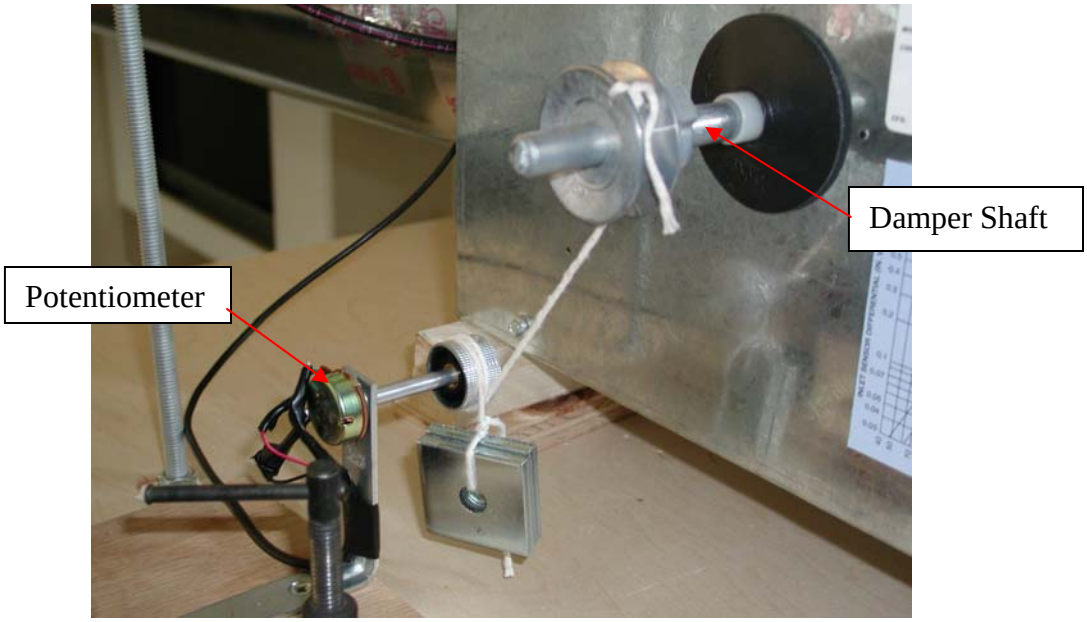

Figure A5: Photo of the Damper Position sensor.

The photo shows this setup on the Titus VAV box. A pulley wheel is mounted on the damper shaft and is connected to a knob on the potentiometer by a string. Tension is maintained on the assembly at all times by a suspended weight. It is believed that this setup could be improved by replacing the string with something less elastic, such as a small cable.

The controllers, when used, also monitored the damper position. All controllers tested, except the Johnson Controls controller, used a floating point actuator where the position was calculated by the controller software by summing the time the motor moved the damper shaft clock-wise and counter clock-wise. The controller would rezero its position whenever it received a command to fully close, which it forced once or twice a day. This calculated position had more resolution than the purpose built sensors and appeared more accurate as well. The accuracy is expected to decline if the rezeroing period is much greater than a day or the damper is frequently repositioned. The Johnson Controls controller has an internal sensor to monitor the position of the damper.

Temperature Sensors

The air temperature in the duct was measured with a sensor provided by The Energy Conservatory as part of the APT and is rated at +-0.5 C. A second sensor was used to check that this sensor was reasonable, but no attempt was made to individually calibrate this sensor.

The controllers came with a temperature sensor. No attempt was made to calibrate these sensors.

Analysis Notes

The data collected with the APT is exported to an ascii format file which can be easily be read with programs like Excel. Much of the analysis was done with a statistical software package called STATA. STATA has a command/programming language which can easily perform common analysis tasks like data smoothing and binning.

Background "Noise"

The measurement of any pressure is affected by several sources of "noise". One source is the vibration transmitted to the pressure sensor via its mounting and another comes from the actual fluctuation in the pressure signal.

All buildings vibrate as traffic rolls past outside, the wind varies, and people walk down the hallway. Another significant source of vibration is from the HVAC fans that move air. These fans are typically mounted on shock absorbers and/or partially isolated from the duct system by flexible couplings to reduce transmitted sound and vibration.

The pressures measured in a duct system will vary, even if the building could be made vibration free, because of factors like: the turbulent nature of the flow, pulsing flow off the fan blades, fan imbalance, and duct design.

A few of these factors have been examined in these tests at the PEC. One indicator of the level of the "background noise" is the fluctuation of the system static pressure. The system fan normally operates at full speed and the normal operating system static pressure is about 0.2" wc (50 Pa) but can rise to as high as 0.7" wc in some modes.

When the system static pressure averages about 0.4 iwc (100 Pa), 50% of the readings fall more than 0.012 (3 Pa) away from the average value. Wider ranges were seen at lower system static pressures.

The two sensors that were used, the APT and Setra, have different response time characteristics. The APT signal has a much shorter response time. Thus it reports a much "noisier" looking signal than the Setra. The Setra sensor may be inherently slower (by design) or it may have a dampening mechanism on the pressure (physically) or on the voltage output (likely an RC circuit). Figure A6 shows data from a typical measurement period (in this case the Nailor fixed damper at 3 degrees). The two sensors track each other as expected, but the Setra sensor shows much less "noise".

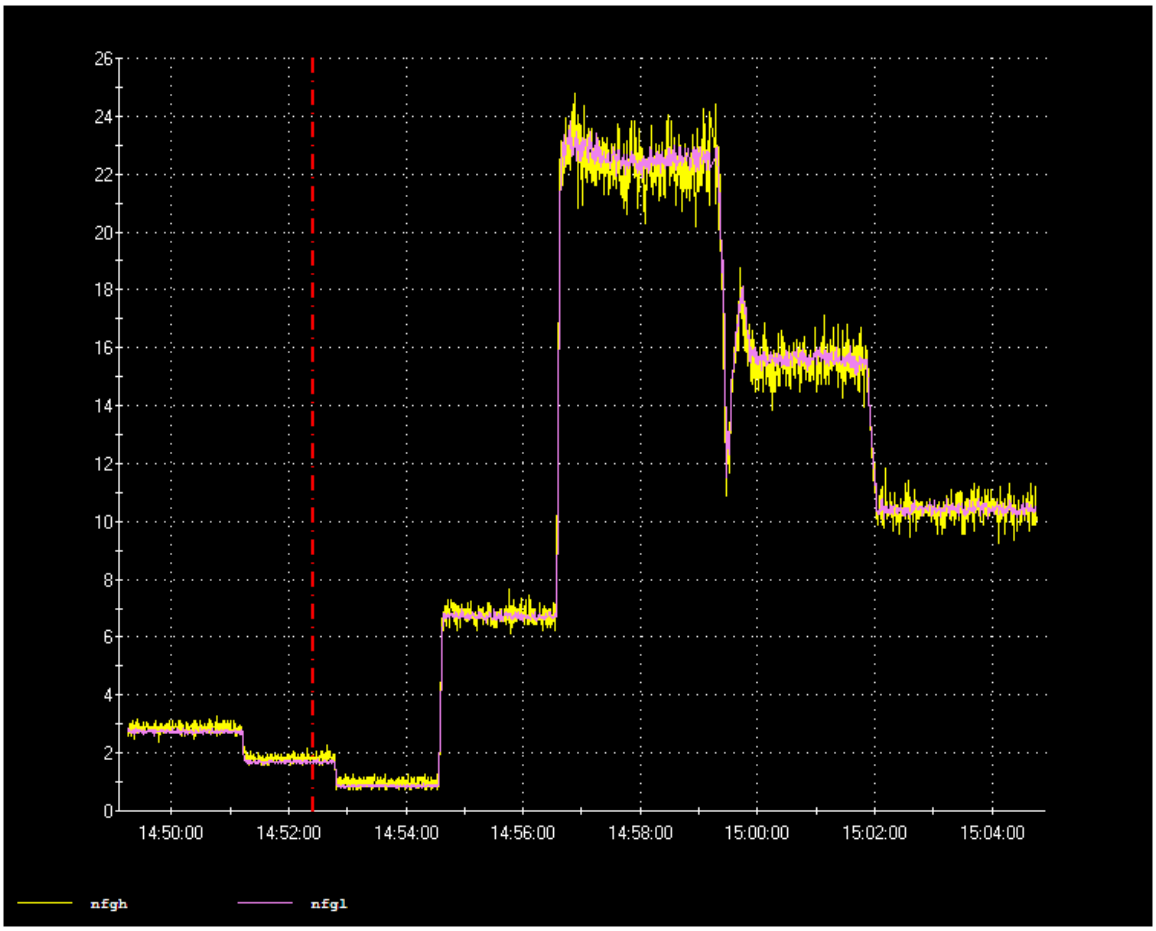

Figure A6: Behavior of the APT and Setra pressure sensor on the Nailor VAV flow grid during a Constant Pressure Test. The yellow, "noisy" signal is from the APT sensor. Note: This graph is produced by the APT software and is seen as the data is being taken.

Another indicator of the background noise is found in measurements taken to check the pressure sensor "zeros". These show that the Setra sensor is stable to within 0.0001 iwc (0.02 Pa). At the same time the APT varies by about a factor of 10 more. Both of these fluctuations are very small but they appear to be correlated, see figure A7, thus not random noise.

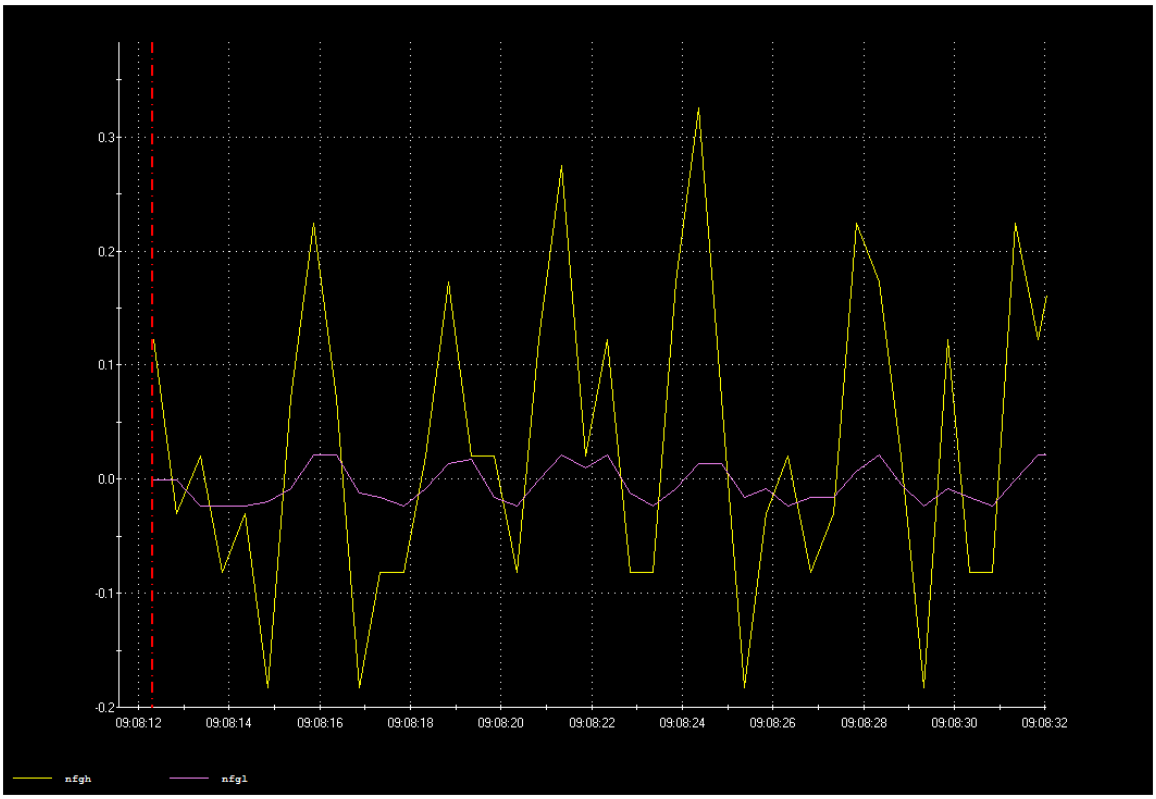

Figure A7: Background pressure fluctuations, possibly due to building vibrations or electrical noise in the data logger. (The vertical grid lines are space every 2 seconds. The horizontal grid lines are space every 0.1 Pa {about 0.0004 iwc}.)

These fluctuations are probably present in all buildings and/or controllers. The controllers need to smooth out these fluctuations in order to provide a smooth control of the damper positions in the VAV boxes.

Appendix B: Nailor Box details

Description of the Nailor VAV Box

The Nailor VAV box (model 30RW) has a single 8" inlet, a damper, and a 2 row hot water reheat coil before the exit. The damper consists of two opposed blades that are driven via a "gear" box with the damper shaft. The gearing is such that the full range of damper motion is achieved with a 45 degree rotation of the damper shaft. Nailor specifies a flow range of 150 to 1000 cfm when used with a digital controller. They have a "Diamond Flow Sensor" to pick up the velocity pressure of up to 1 iwc. They list the K factor as 1007 (making an Amplification factor of 1.927).

Figure 1. Nailor K-Factor Cutsheet

Nailor Industries Inc.

Installation and Operation Manual Diamond Flow Sensor K-Factors for VAV Terminal Units

Model Series:
3000 Single Duct
3210 Dual Duct
38S Underfloor Fan Powered

Inlet Size	Type	Duct Area (Sq. ft.)	K-Factor (cfm)	Velocity (fpm)
4	ROUND	0.087	214	2460
5		0.136	312	2294
6		0.196	494	2520
7		0.267	712	2667
8		0.349	1007	2885
9		0.442	1303	2948
10		0.545	1435	2633
12	OVAL	0.754	2184	2897
14		0.970	2746	2831
16		1.186	3729	3144
24x16	RECT.	2.667	6437	2414

Model Series:
3230 Dual Duct
3240 "Blendmaster" Dual Duct
35N Parallel Fan Powered
35S Series Fan Powered
35SST "Stealth™" Series Fan Powered
36VRR Round Retrofit

Inlet Size	Type	Duct Area (Sq. ft.)	K-Factor (cfm)	Velocity (fpm)
4	ROUND	0.087	214	2460
5		0.136	312	2294
6		0.196	494	2520
7		0.267	712	2667
8		0.349	1007	2885
9		0.442	1303	2948
10		0.545	1435	2633
12		0.785	2149	2738
14		1.069	3061	2863
16		1.395	4047	2901
18	OVAL	1.683	4986	2963

Model Series:
36VRS Square/Retangular Retrofit

Unit Size	Type	Valve Size (inches)	Damper (valve) area (Sq. ft.)	K-Factor (cfm)	Velocity (fpm)
7	SQUARE OR RECT.	5 x 5	0.174	479	2753
8		6 x 6	0.250	689	2756
9		8 x 6	0.333	919	2760
10		10 x 8	0.555	1531	2759
11		14 x 8	0.778	2150	2763
11A		18 x 6	0.750	2068	2757
12		12 x 10	0.833	2297	2758
13		18 x 10	1.250	3446	2757
14		18 x 12	1.500	4135	2757
15		20 x 14	1.944	5360	2757
15A		30 x 12	2.500	6892	2757
16		22 x 16	2.444	6739	2757
17		24 x 18	3.000	8270	2757
18		30 x 20	4.167	11486	2756
19		40 x 20	5.555	15315	2757

Model Series:
37S Low Profile Fan Powered
37SST Low Profile "Stealth™" Fan Powered

Inlet Size	Type	Duct Area (Sq. ft.)	K-Factor (cfm)	Velocity (fpm)
4	ROUND	0.087	214	2460
5		0.136	312	2294
6		0.196	494	2520
8		0.349	1007	2885
10		0.545	1435	2633
14x10	RECT.	0.972	2558	2632

Equations:

$Q = K \times \sqrt{\Delta P}$ $\quad \Delta P = \left(\frac{Q}{K}\right)^2$

Where: Q = Airflow Rate (cfm)
ΔP = Sensor Differential Pressure ("w.g.)
K = K-Factor Calibration Constant

The K-Factors tabulated in the above tables are the airflow required to produce a 1.0" w.g. differential pressure at the Diamond Flow Sensor.

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Flow Calibration (K factor)

The purpose of these tests is to examine the performance of the flow grid of a typical VAV box. We concentrate on two factors that might indicate that a single "K" factor might not be accurate enough to properly insure adequate air flow or lead to problems for the controls of these devices. These are: flows lower than the flow range listed by the manufacturer and the possible influence of the damper position, especially at low flows. Data was taken for flows as low as 20 cfm at several damper positions, concentrating on nearly closed dampers.

Two kinds of tests were performed on the VAV boxes without the controllers installed. One with dampers in fixed positions at various flows, and ones with "fixed" inlet static pressures where the damper position was varied. Data was collected for about one to two minutes at 2 Hz at each test point. Combined, these tests made more than 40,000 measurements for each VAV box.

The "K factor" [1] was calculated by fitting the flow (cfm) vs. square root of the pressure (iwc) data to a line. We obtained a slope (the K value) of 925 with an RMS error of 4. Limiting the fit to flows under 150 cfm (a flow grid pressure of 0.028 iwc) results in a K value of 921, a difference that is less than one cfm in this range.

Note that in a typical commercial application the "K factor" (or "Amplification Factor") is usually adjusted in-situ for the specific setup of each VAV box. These are typically determined by test-and-balancing crews by summing the flows from the registers and adjusting the "K" factor so that the flows match. This process ignores duct leakage downstream of the VAV box and relies on the accuracy of the flow hoods employed[2,3]. These factors can severely affect the process and result in a poorly determined K value.

Figure B1 shows the measured flows and flow grid pressures on a log log plot. We see that the specifications from Nailor (K=1007), the dotted line, would predict a flow which is larger, by 9%, than the result we obtained of K=925, for the same flow grid pressure. While these results indicate that a single calibration point taken above 0.01 iwc would obtain a calibration suitable for all flows this might not be the case for other inlet geometries. A fit to the data where any exponent is allowed yields an exponent of 0.5053. This fit is not distinguishable from the fit where we force the exponent to be 0.5 (square root flow).

[1] Definitions of "K" and other terms used are found in Nomenclature section of the main report.

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[2] Etur, E. 2003. "Evaluation of Flow Capture Technologies for Duct Leakage, Screening in Large Commercial Buildings". Ecole Nationale Supérieure d'Arts; et Métiers, Paris, France.

[3] *Diamond, R.C., C.P. Wray, D.J. Dickerhoff, N.E. Matson, and D. Wang.* **"Thermal Distribution Systems in Commercial Buildings"** 2003. LBNL-51860.

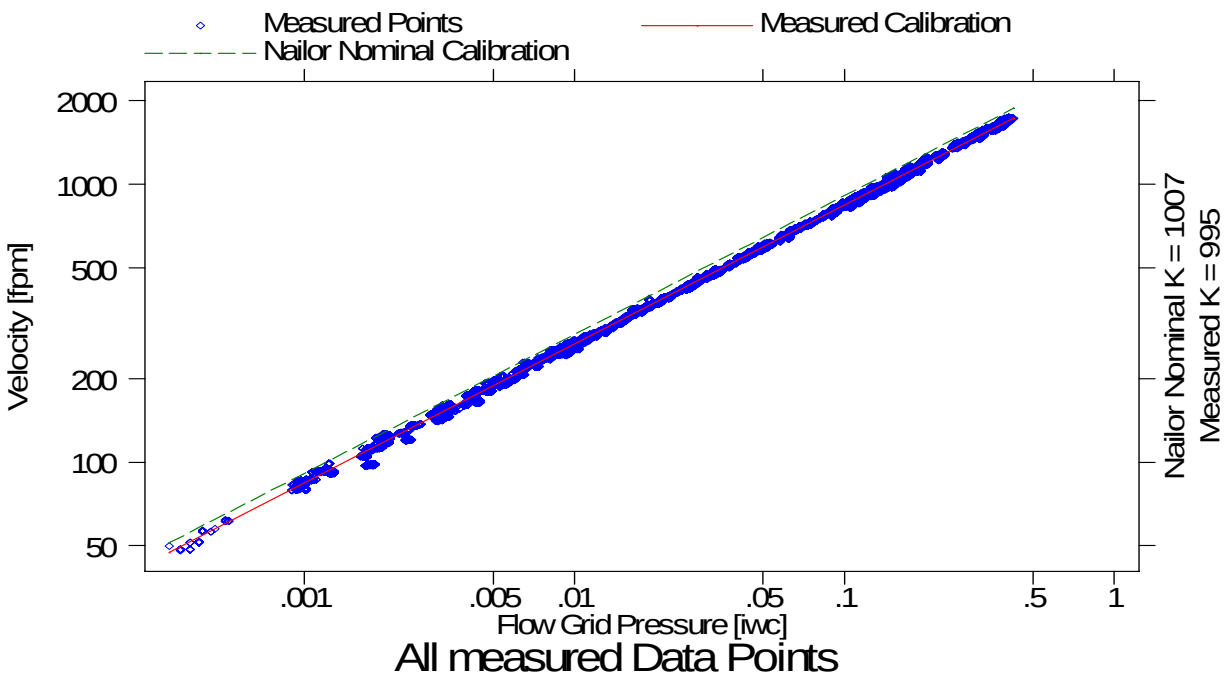


Figure B1, VAV flow performance from all measured data (fixed damper and fixed inlet pressure modes).

Fixed Damper Position Tests

Figures B2 to B7 are subsets of the data presented in Figure B1 with each figure showing the data for one damper shaft position. Lines corresponding to the nominal "K" value from Nailor and the K factor found from regression of all the measured points are also plotted.

Figure B6, at a damper shaft position of 1.5 degrees, shows a possible divergence from the fit line determined with all the data. Interestingly this is not seen in the fully closed damper data, figure B7.

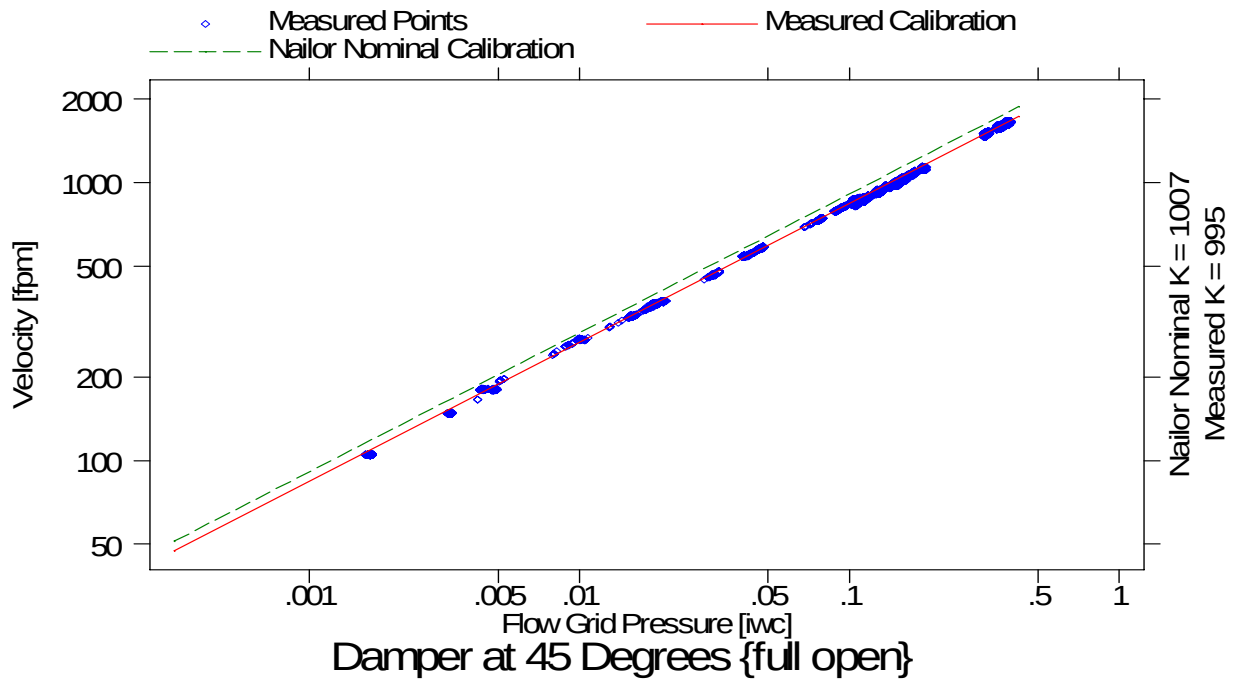


Figure B2: Subset of the data from the fixed damper shaft position at 45 degrees (fully open).

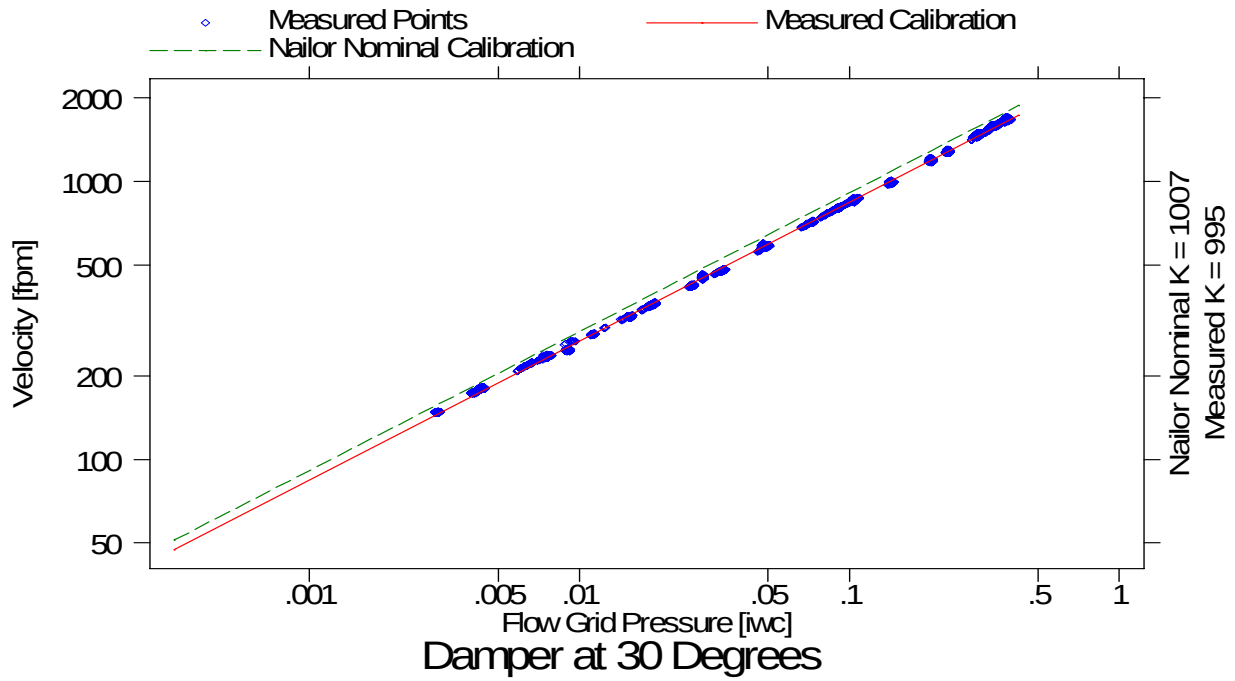


Figure B3: Subset of the data from the fixed damper shaft position at 30 degrees.

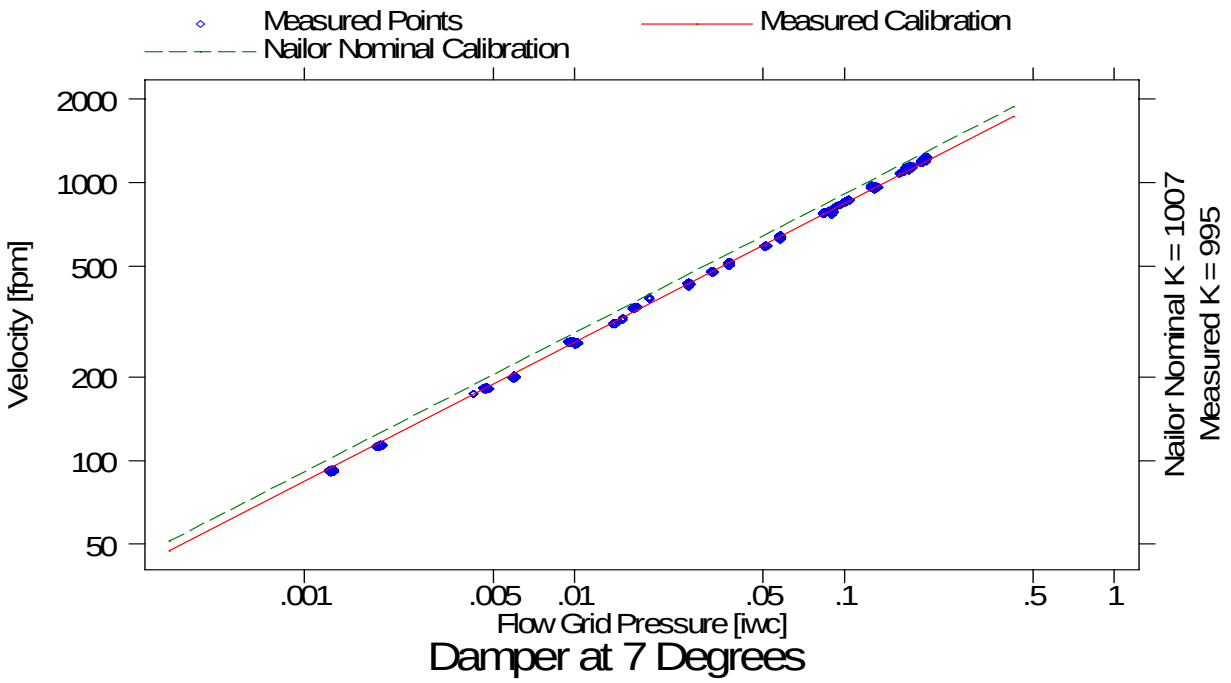


Figure B4: Subset of the data from the fixed damper shaft position at 7 degrees.

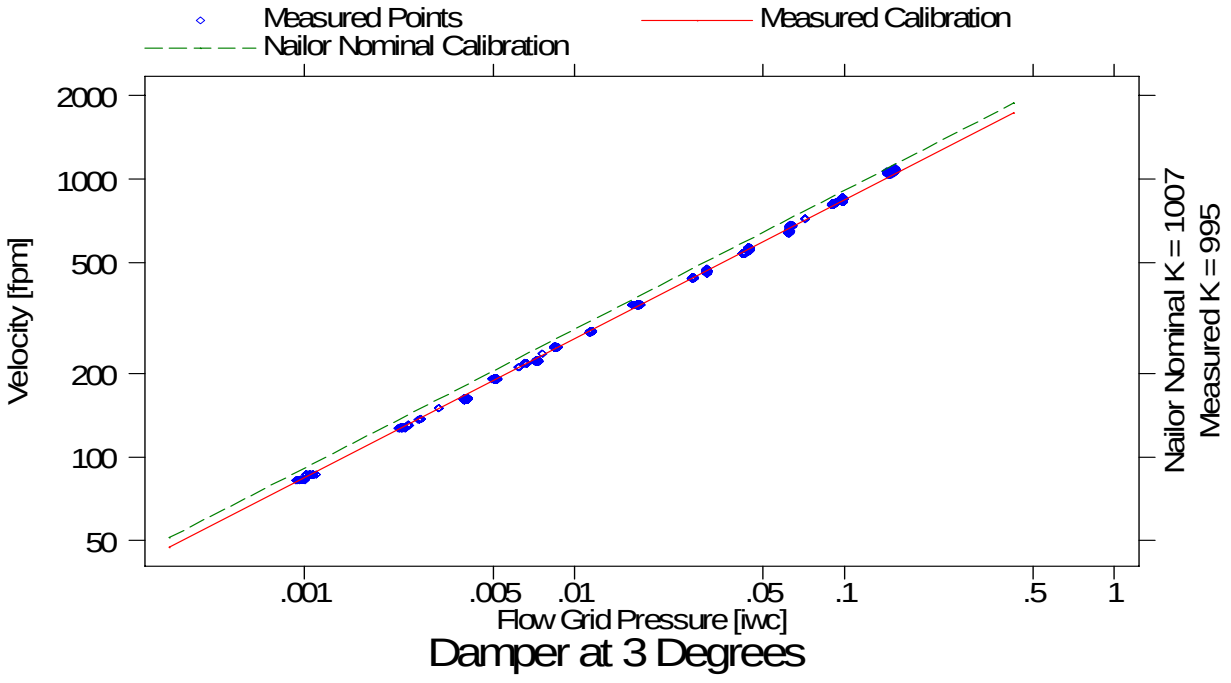


Figure B5: Subset of the data from the fixed damper shaft position at 3 degrees.

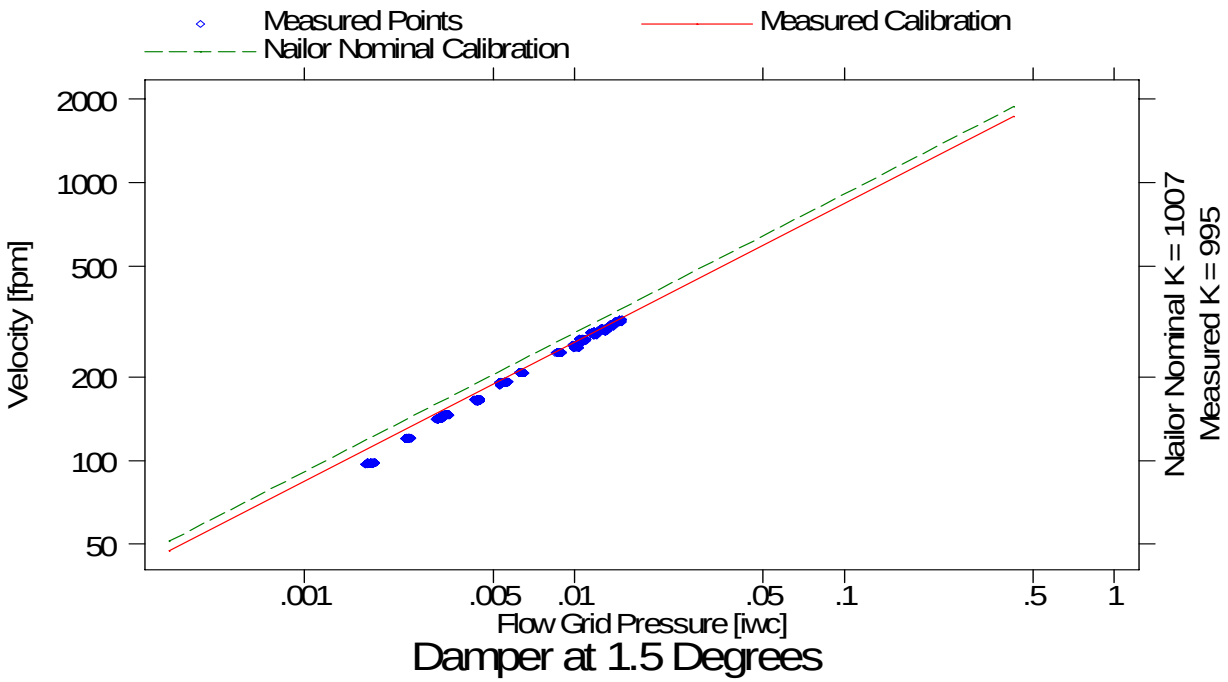


Figure B6: Subset of the data from the fixed damper shaft position at 1.5 degrees.

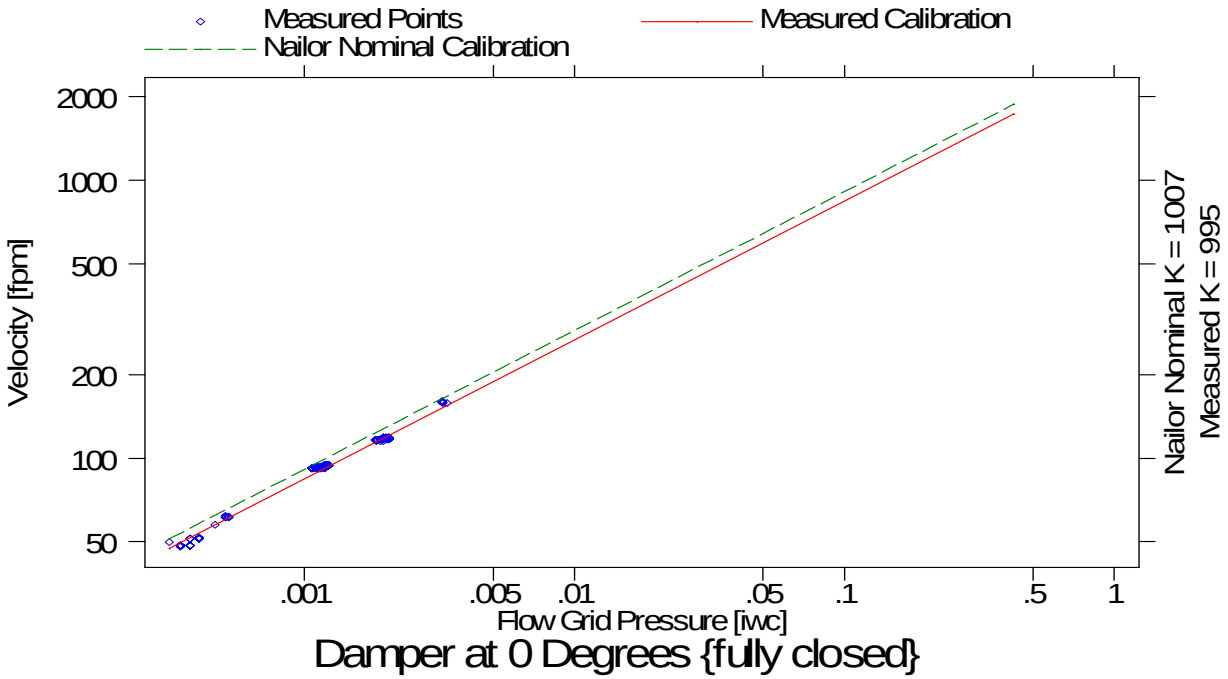


Figure B7: Subset of the data from the fixed damper shaft position at 0 degrees (fully closed).

The data from the measurements with the damper at 0 degrees can be used to calculate the leakage of the Nailor damper. Figure B8 shows this leakage curve.

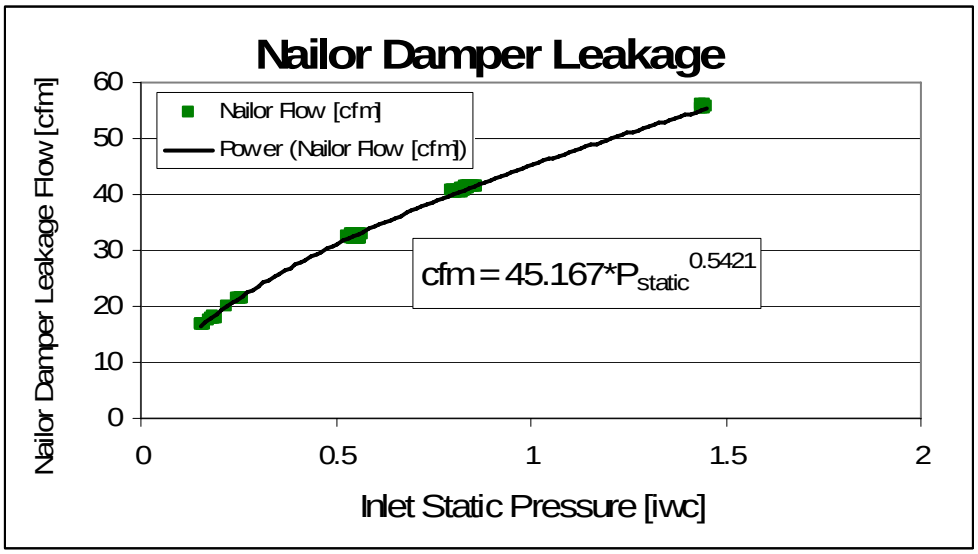


Figure B8: The leakage of the Nailor Damper when fully closed.

The damper leaks about 45 cfm when the inlet pressure is one inch of water. This corresponds to a pressure on the flow grid of about 0.002 iwc. Zeroing procedures that assume no damper leakage would be off by 0.002 iwc if the zeroing was done during these conditions. This will cause significant under estimation of the flow at low flows. This will result in the true flow being higher than the setpoint flow. The Titus damper had no measurable leakage.

Nailor Fixed Duct System Static Pressure Measurements

Measurements were also made at a series of constant inlet static pressures, where the software controlling the "Duct Blaster" fan was asked to maintain a constant VAV inlet pressure. The damper was then adjusted to vary the flow and a series of measurements was taken. The data is shown in Figures B9 to B15 for inlet pressures at 1.5, 1.00, 0.75, 0.66, 0.50, 0.25, and 0.10 iwc.

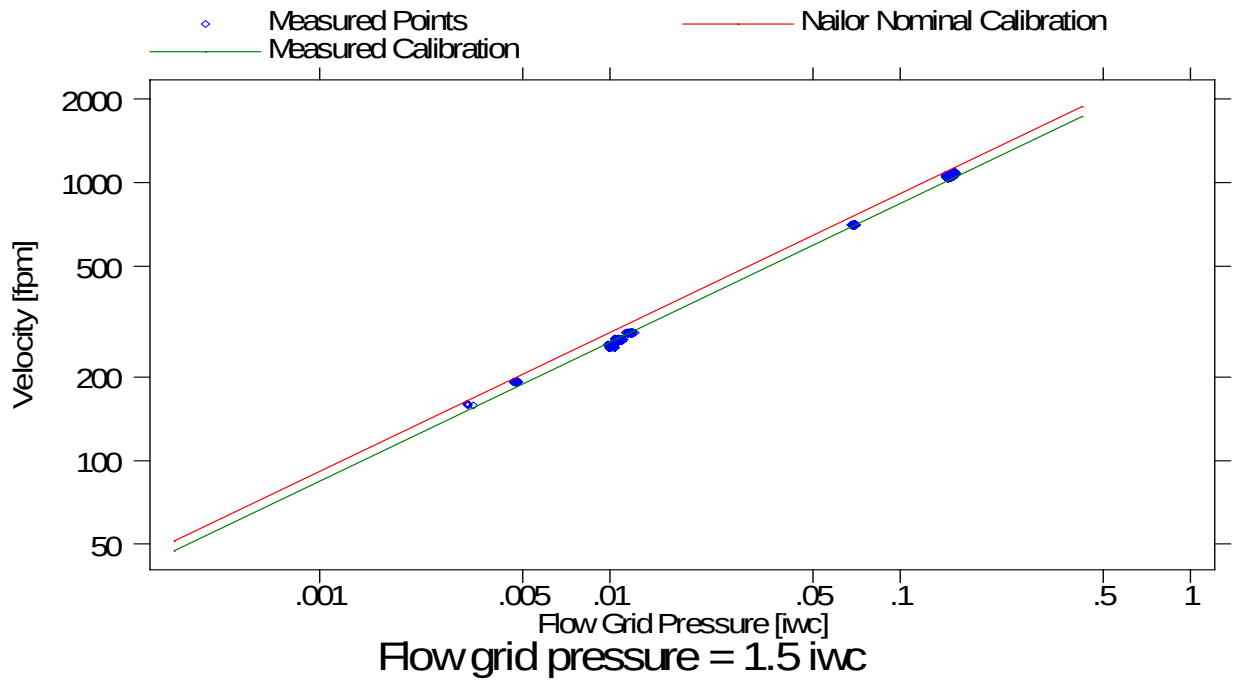


Figure B9: Calibration at various damper positions at an inlet static pressure of 1.5 iwc.

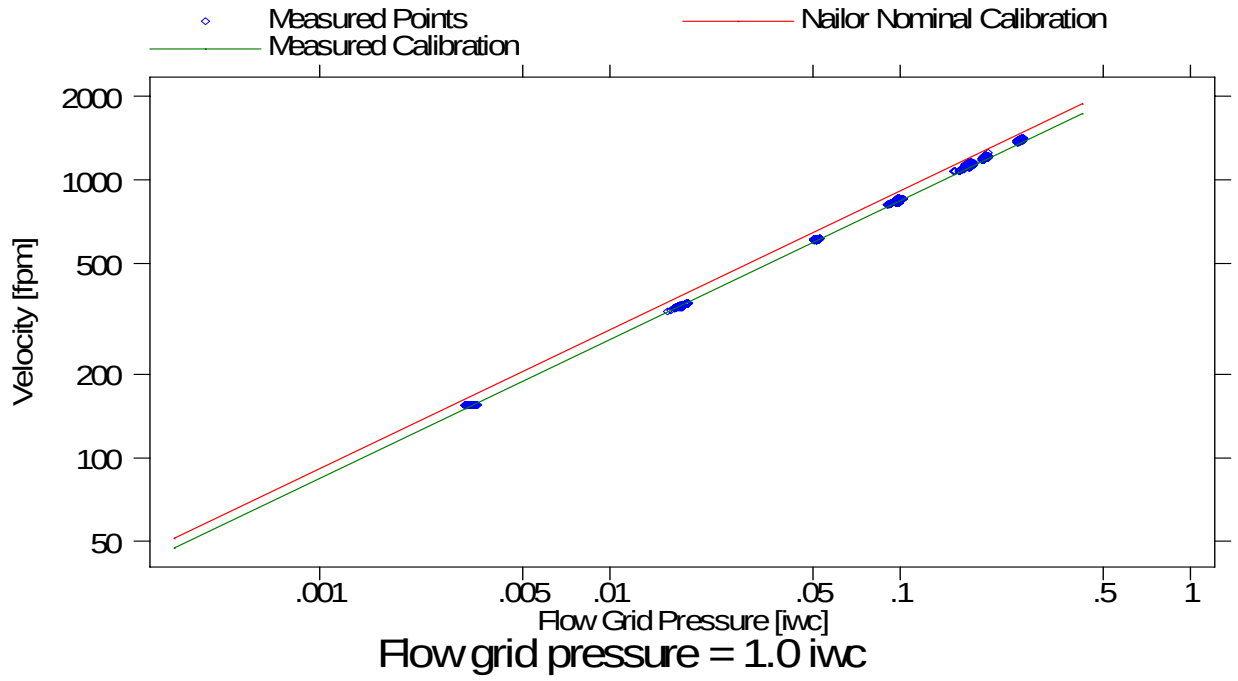


Figure B10: Calibration at various damper positions at an inlet static pressure of 1.0 iwc.

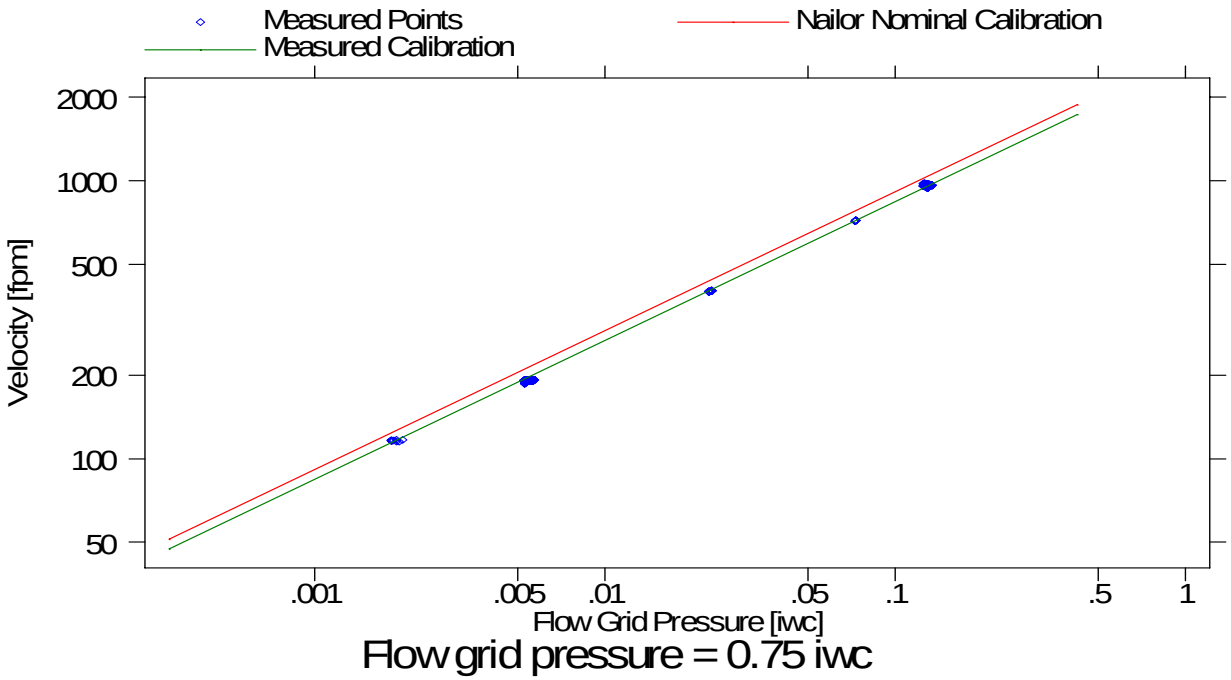


Figure B11: Calibration at various damper positions at an inlet static pressure of 0.75 iwc.

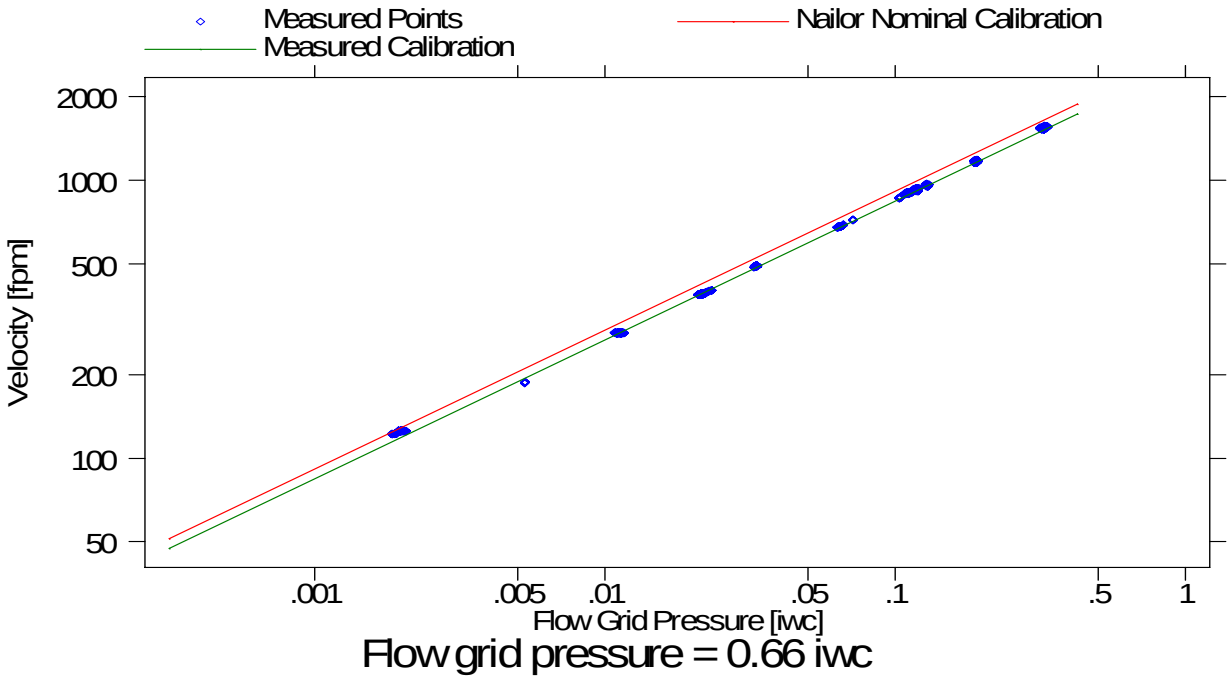


Figure B12: Calibration at various damper positions at an inlet static pressure of 0.66 iwc.

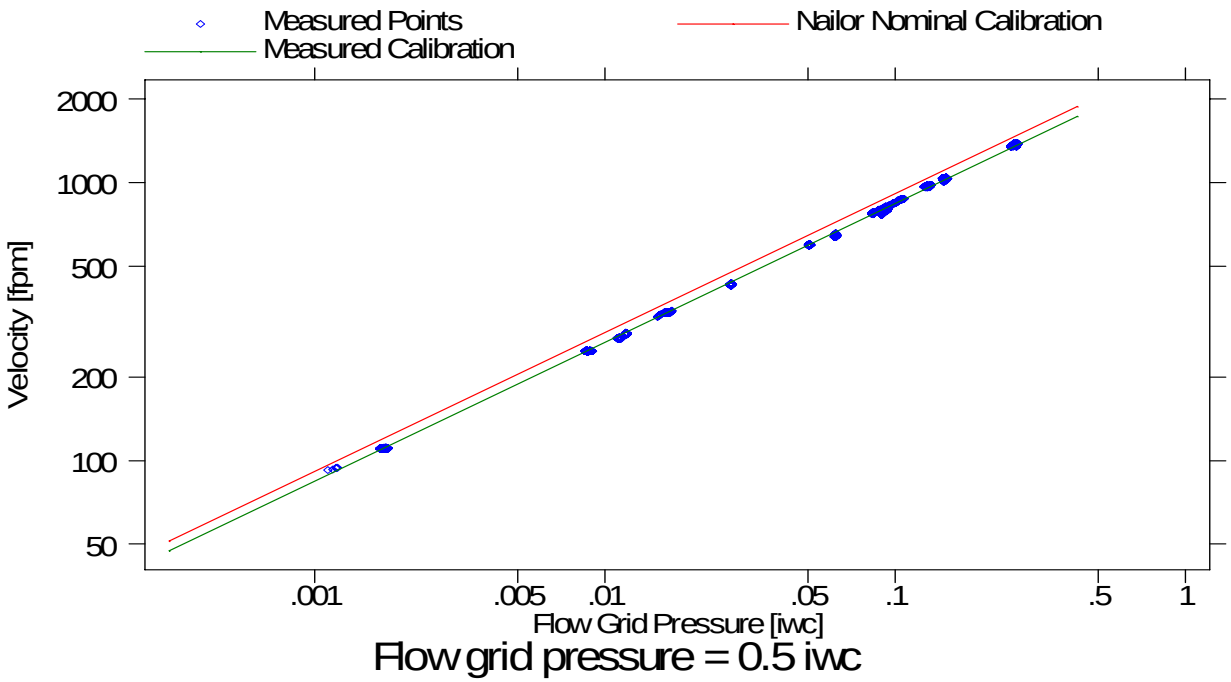


Figure B13: Calibration at various damper positions at an inlet static pressure of 0.5 iwc.

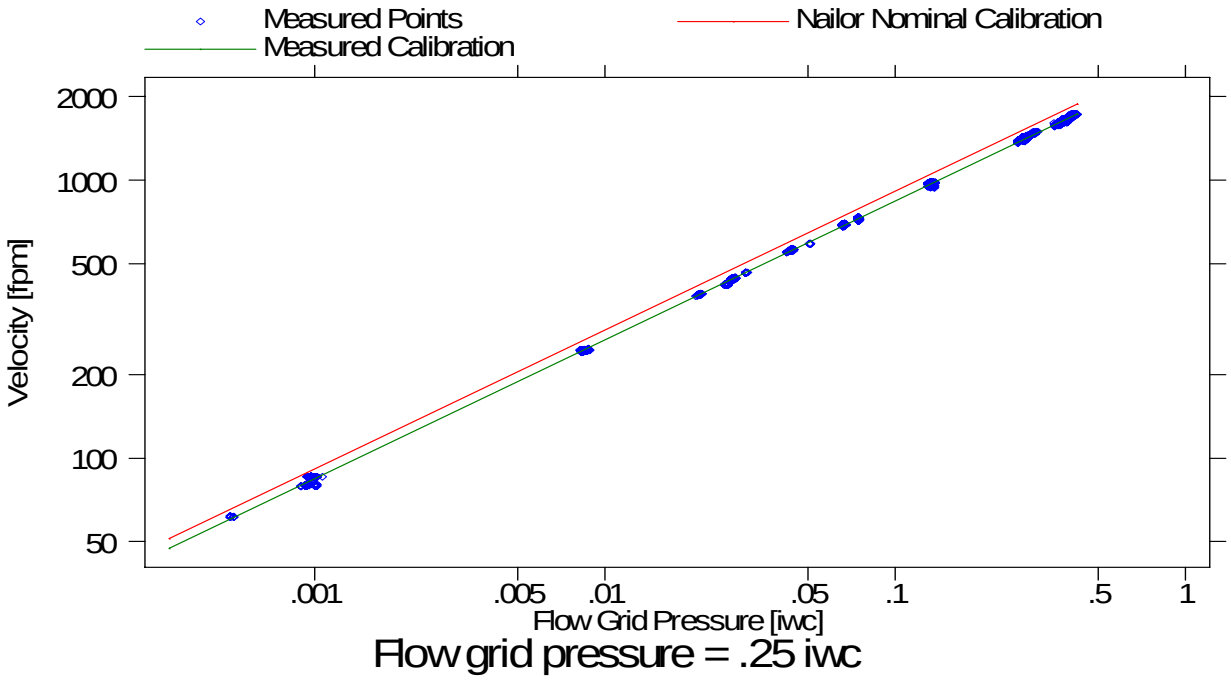


Figure B14: Calibration at various damper positions at an inlet static pressure of 0.25 iwc.

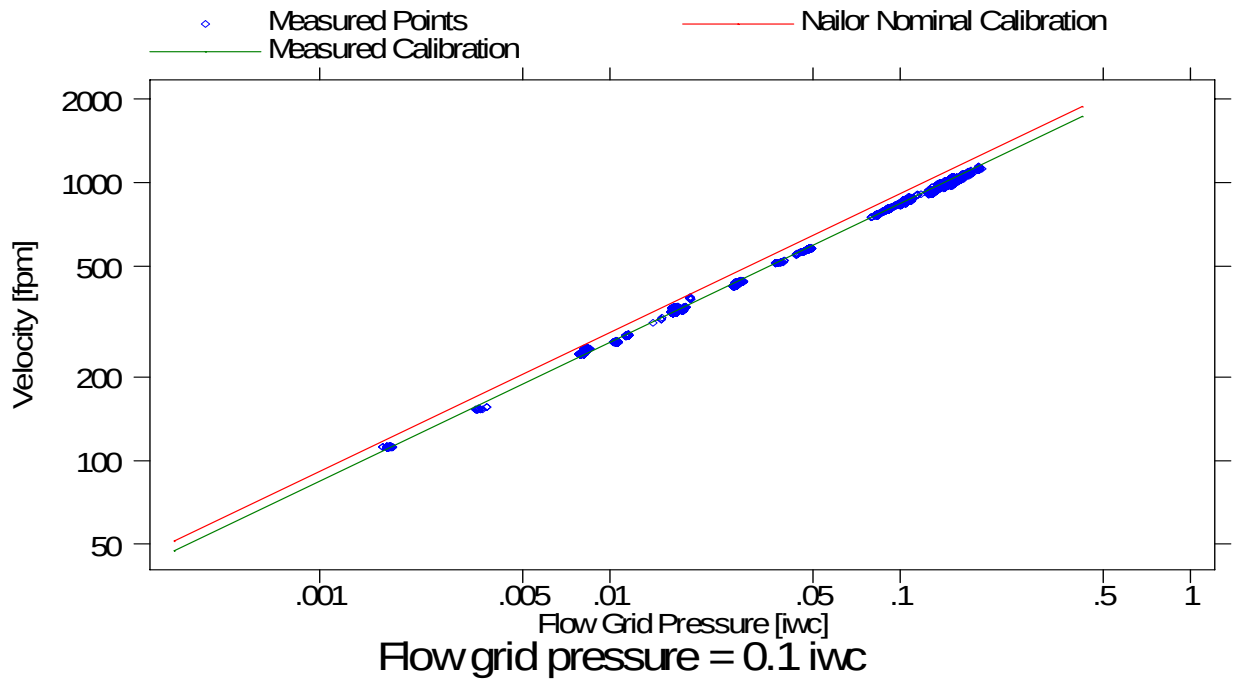


Figure B15: Calibration at various damper positions at an inlet static pressure of 0.1 iwc.

Appendix C: Titus Box details

See Appendix B for details about these measurements and discussion of relevant issues.

Description of the Titus VAV Box

The Titus VAV box (model DESV) has a single 8" inlet, a damper, and a 2 row hot water reheat coil before the exit. The damper consists of a single round sheet metal disk fixed to a central shaft. The edges of the damper have a rubber gasket which, when closed, make an air tight seal and no measurable leakage There is a mechanical stop at the fully closed position, although the gasket fully seals the inlet well before this stop (about 17.5 degrees). The damper is operated over a 90 degree range by rotating its central shaft. Titus specifies a flow range of 0 to 900 cfm. Titus did not specifically recommended a low flow limit when used with a digital controller. They have a "Multipoint center averaging inlet velocity sensor" to pick up the velocity pressure of up to 1 iwc. They list the K factor as 904 (making an Amplification factor of 2.39).

Titus The Leader in Air Management

Installation Manual KFACTOR-IOM-1.0 8-9-02

AeroCross™ K-Factors

AeroCross Sensor - Calibration Curves

Inlet Sensor Applications (All Units Except QCV)

Unit Size	Duct Area SQ FT	K-Factor CFM	K-Factor FPM	Sensor Qty.	Sensor Size
04	0.087	273	3138	1	4/5
05	0.136	390	2647	1	4/5
06	0.196	448	2286	1	6
07	0.267	667	2498	1	7
08	0.349	904	2590	1	8
09	0.442	1167	2640	1	9
10	0.545	1436	2635	1	10
12	0.785	1891	2409	1	12
14	1.069	3015	2820	1	14
16	1.395	3839	2752	1	16
20	0.778	2106	2707	1	8
22	0.778	2106	2707	1	8
26	1.000	2498	2498	1	8
40	2.667	7176	2691	2	14

Equations:

$$CFM = K\sqrt{\Delta P}$$

$$\Delta P = \left(\frac{CFM}{K}\right)^2$$

ΔP = Differential Pressure On AeroCross, IN WG

K = Flow Required To Produce A 1.0 IN WG Differential Pressure On AeroCross, CFM

Titus K Factor Tests Results

The "K factor" was calculated by fitting the flow (cfm) vs. square root of the pressure (iwc) data to a line. We obtained a slope (the K value) of 869 with an RMS error of 4. Limiting the fit to flows under 150 cfm (a flow grid pressure of 0.030 iwc) results in a K value of 877, a difference that is less than 1.5 cfm in this range.

Figure C1 shows the measured flows and flow grid pressures on a log log plot.
We see that the specifications from Titus, the dotted line, would predict a flow which is larger, by 4%, than the result we obtained for the same flow grid pressure. A fit to the data where any exponent is allowed yields an exponent of 0.4912. This fit is not distinguishable from the fit

where we force the exponent to be 0.5 (square root flow). However it looks like there is some under prediction at flows below about 75 fpm (~25 cfm).

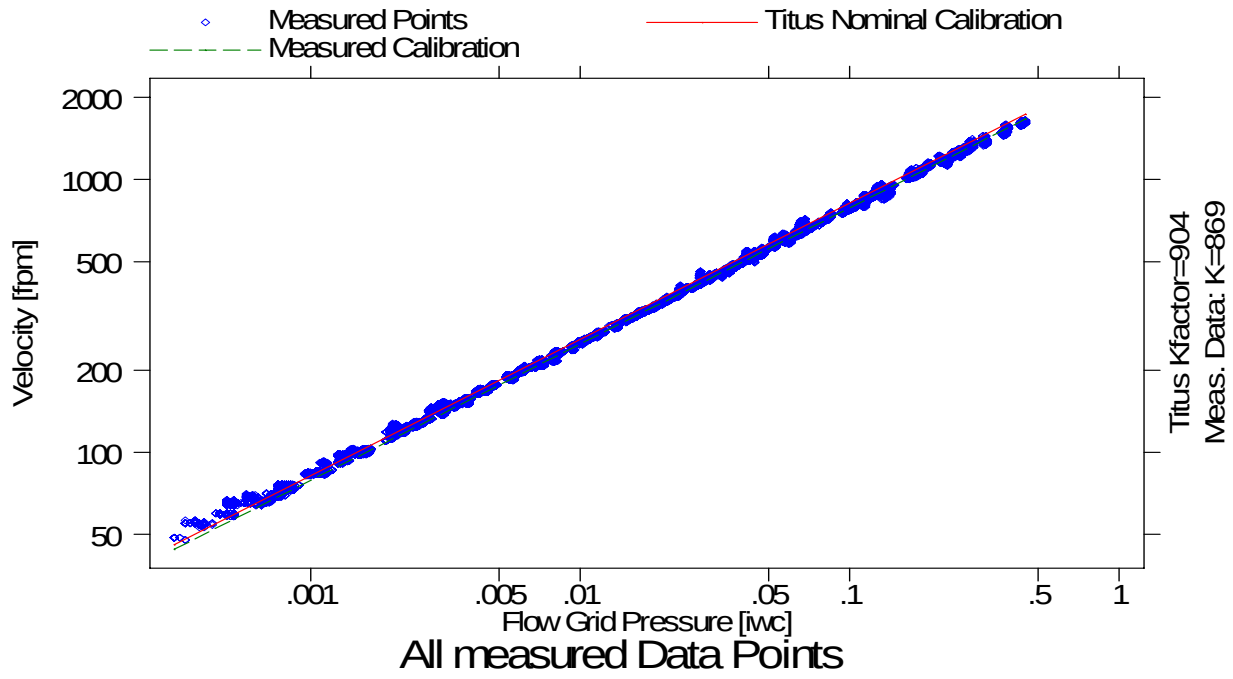


Figure C1, VAV performance from all measured data.

Fixed Damper Position Tests

Looking closely at a subset of this data, for each of the fixed damper positions, nothing is seen to distinguish the different damper positions. Figures C2 to C7 are subsets of the data presented in Figure C1 with each figure showing the data for one damper shaft position. Lines corresponding to the nominal "K" value from Titus and the K factor found from regression of all the measured points are also plotted.

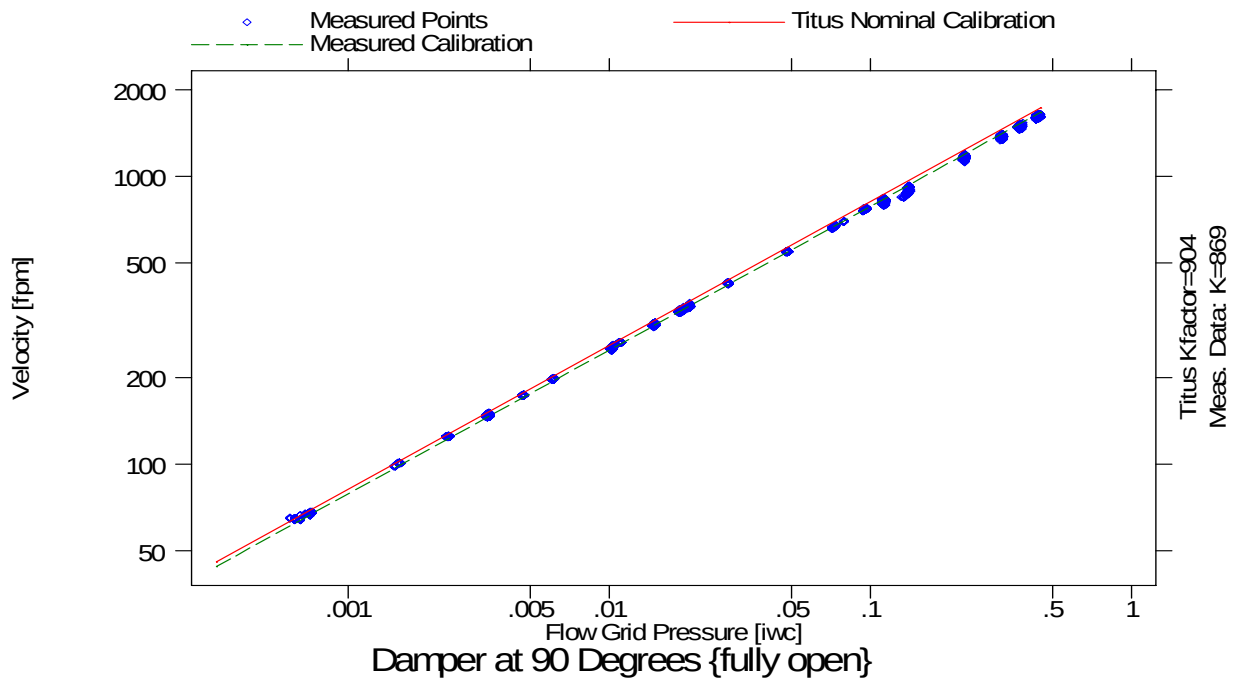


Figure C2: Subset of the data from the fixed damper shaft position at 90 degrees (fully open).

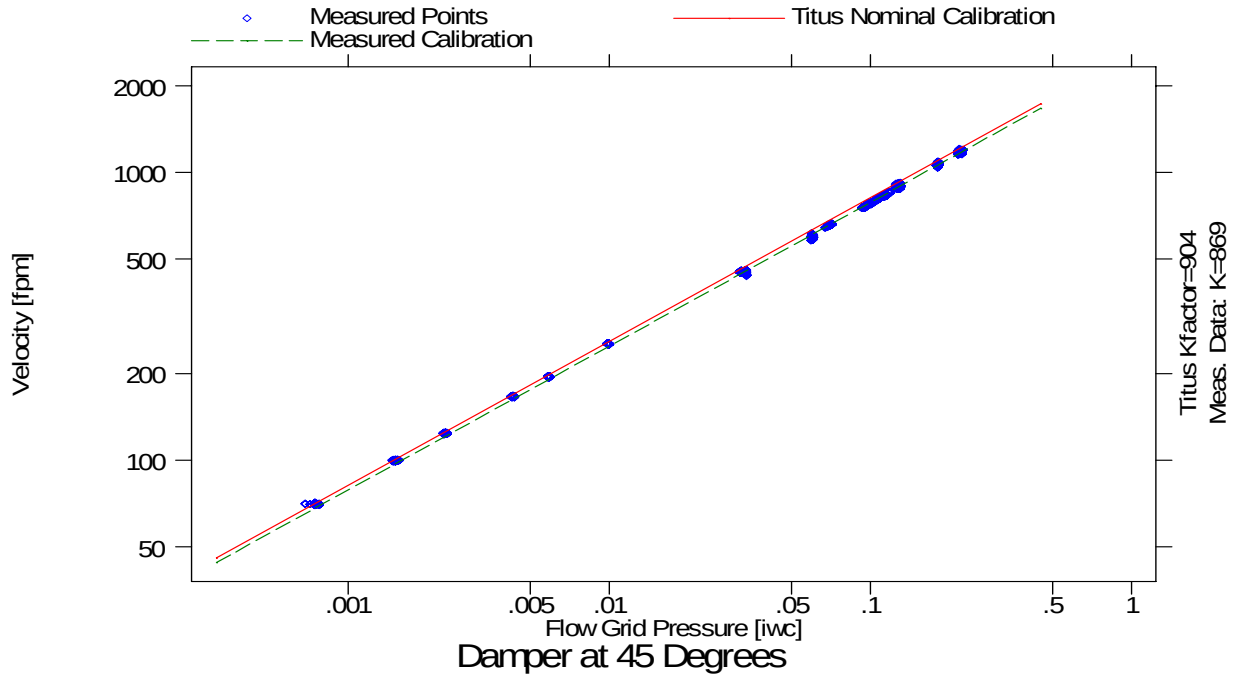


Figure C3: Subset of the data from the fixed damper shaft position at 45 degrees.

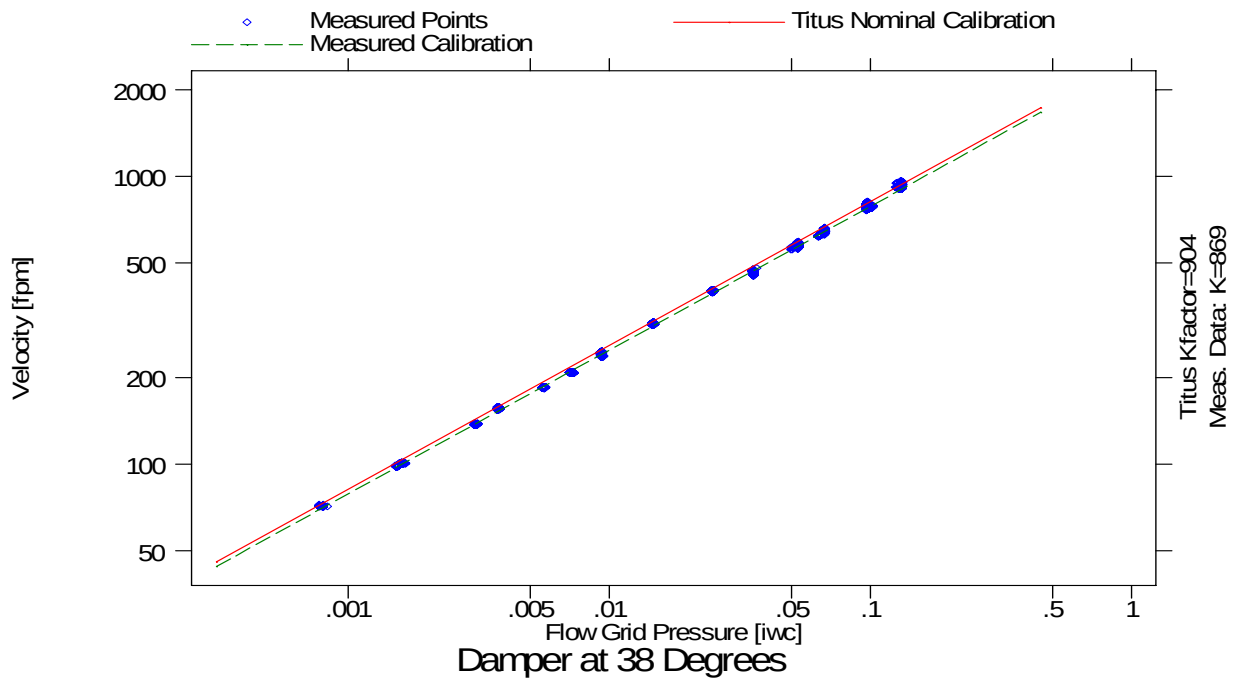


Figure C4: Subset of the data from the fixed damper shaft position at 38 degrees.

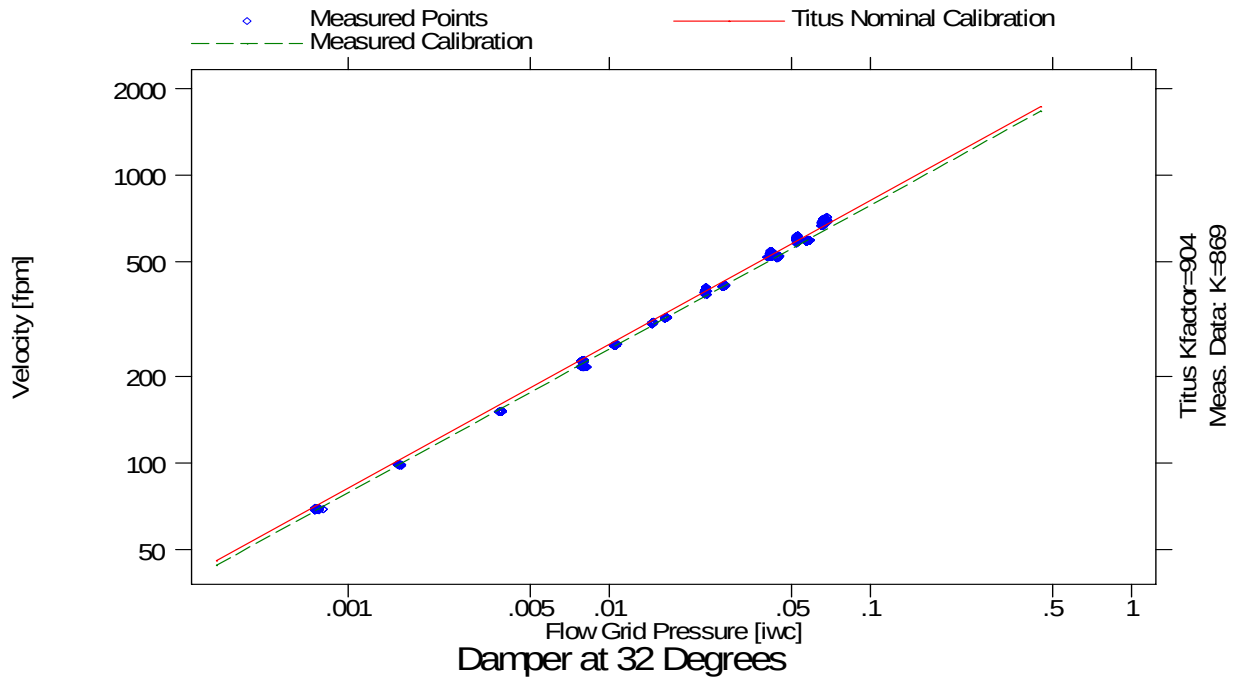


Figure C5: Subset of the data from the fixed damper shaft position at 32 degrees.

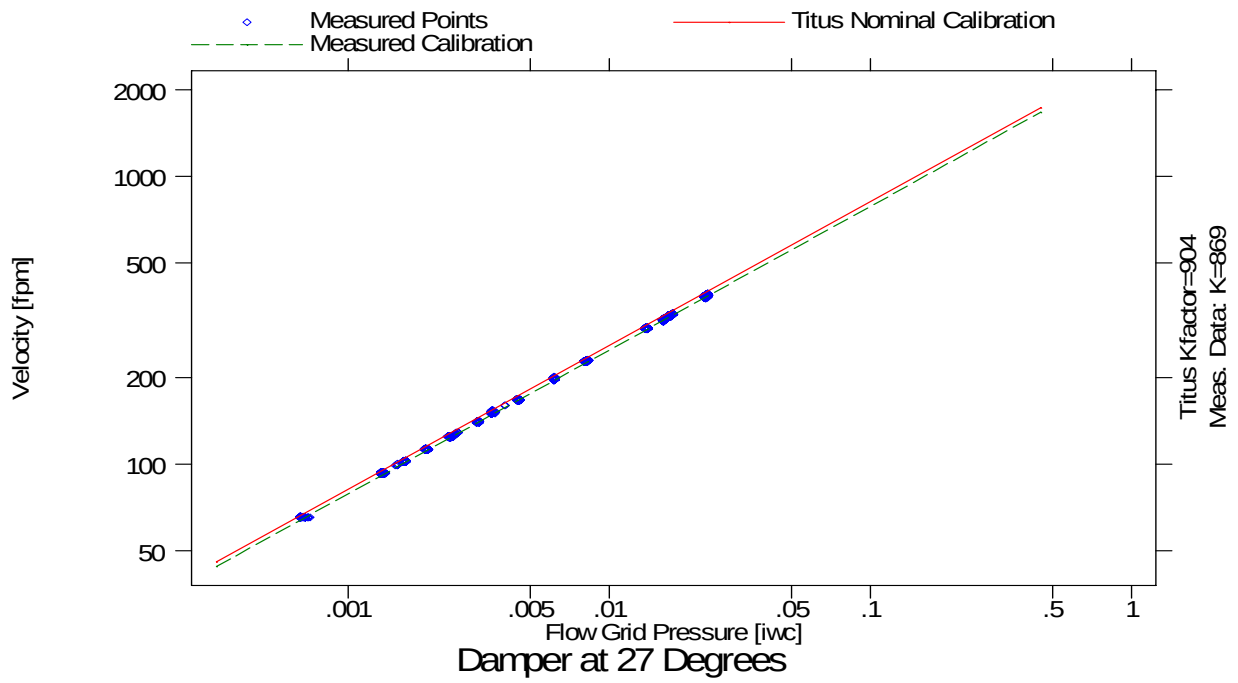


Figure C6: Subset of the data from the fixed damper shaft position at 27 degrees.

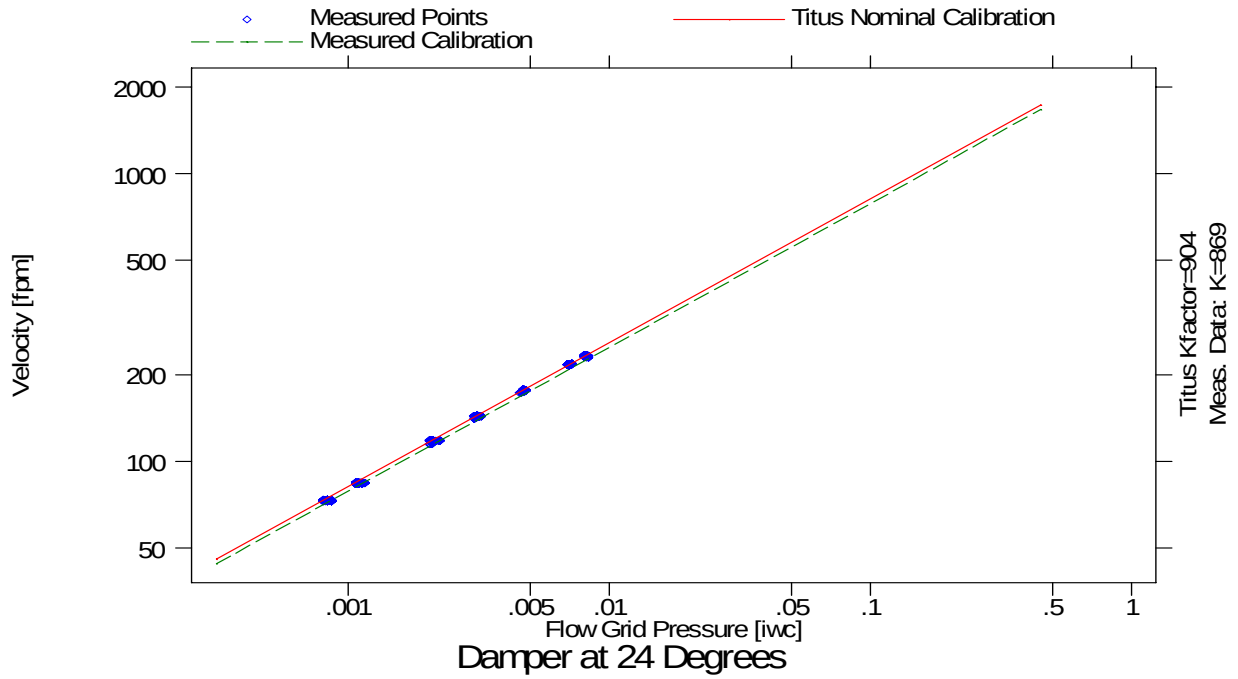


Figure C7: Subset of the data from the fixed damper shaft position at 24 degrees.

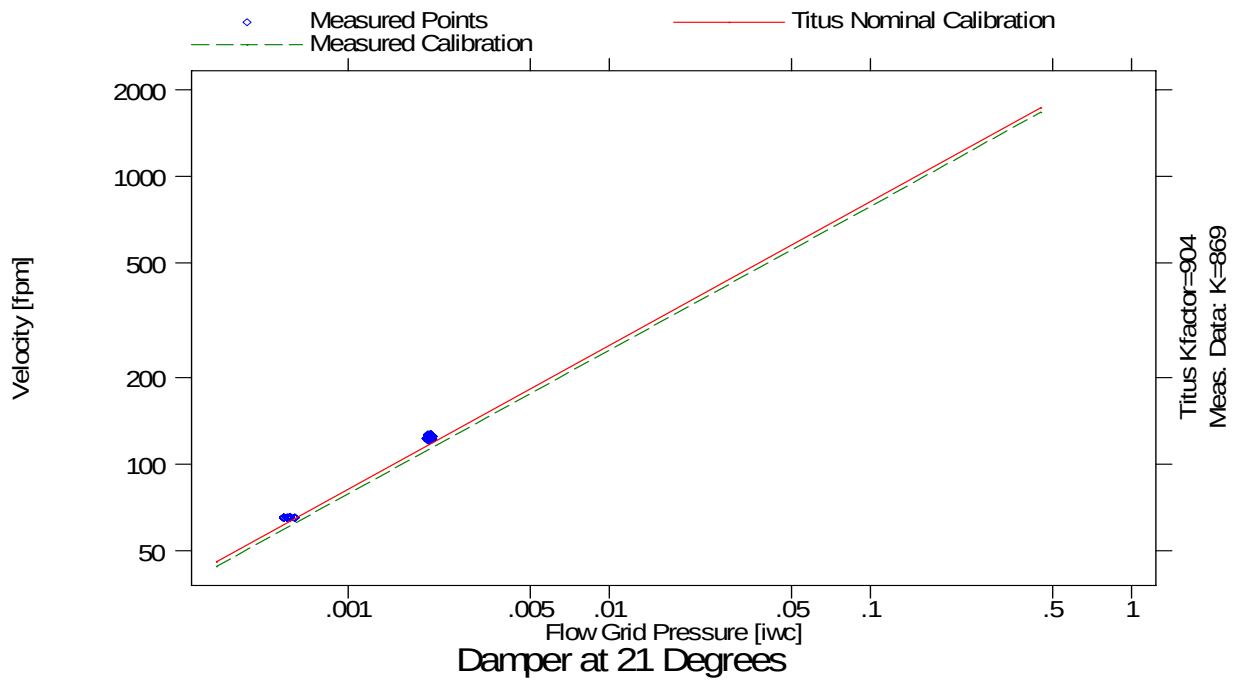


Figure C8: Subset of the data from the fixed damper shaft position at 21 degrees.

Fixed Duct System Static Pressure Measurements

Measurements were also made at a series of constant inlet static pressures, where the software controlling the "Duct Blaster" fan was asked to maintain a constant VAV inlet pressure. The damper was then adjusted to vary the flow and a series of measurements was taken. The data is shown in Figures C9 to C16 below is for inlet pressures at 2.0, 1.50, 1.00, 0.75, 0.66, 0.50, 0.25, and 0.10 iwc.

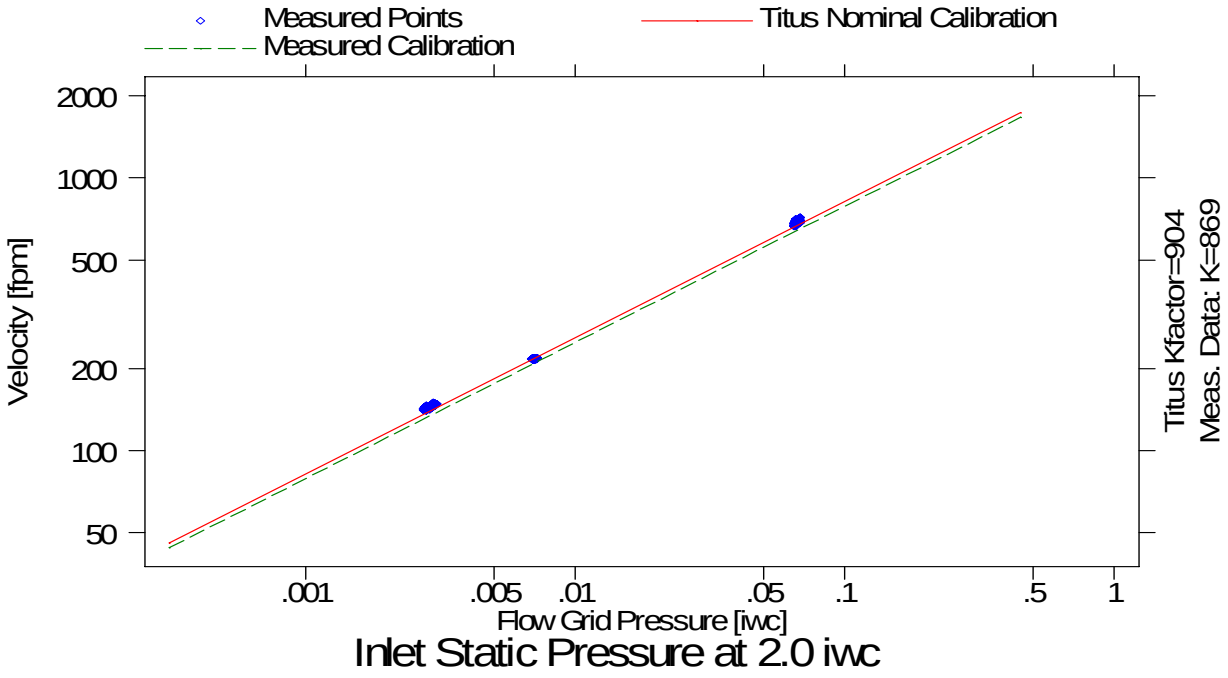


Figure C9: Calibration at various damper positions at an inlet static pressure of 2.0 iwc.

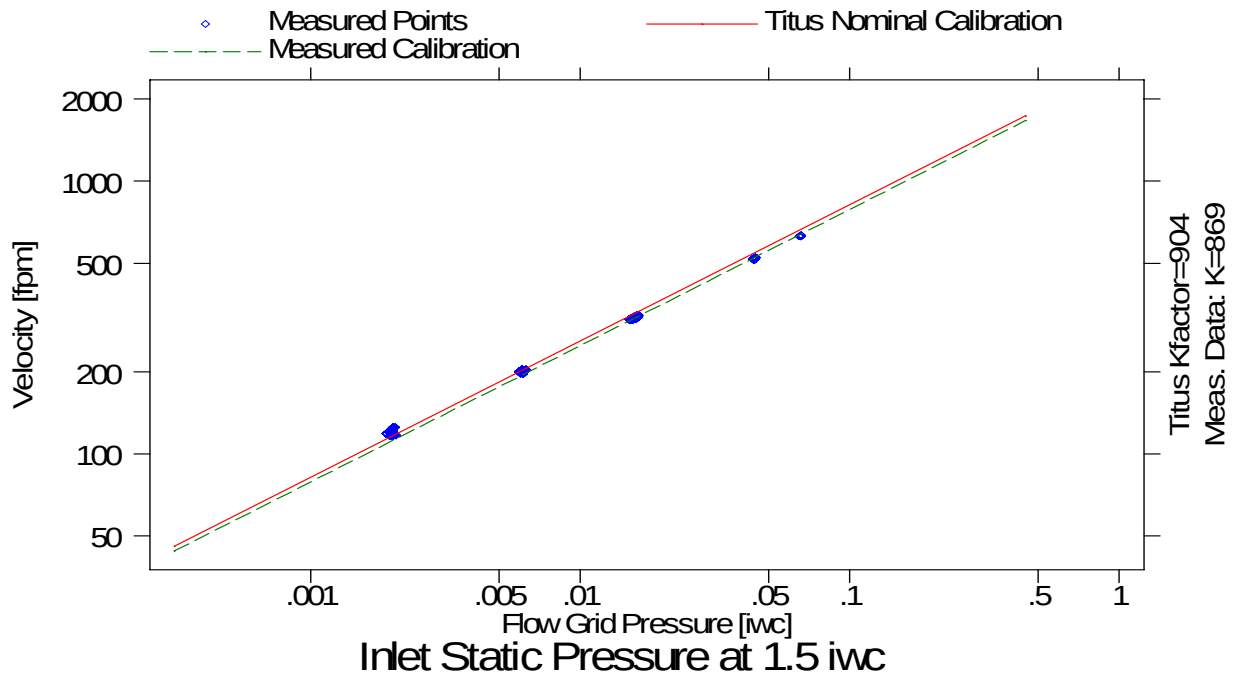


Figure C10: Calibration at various damper positions at an inlet static pressure of 1.5 iwc.

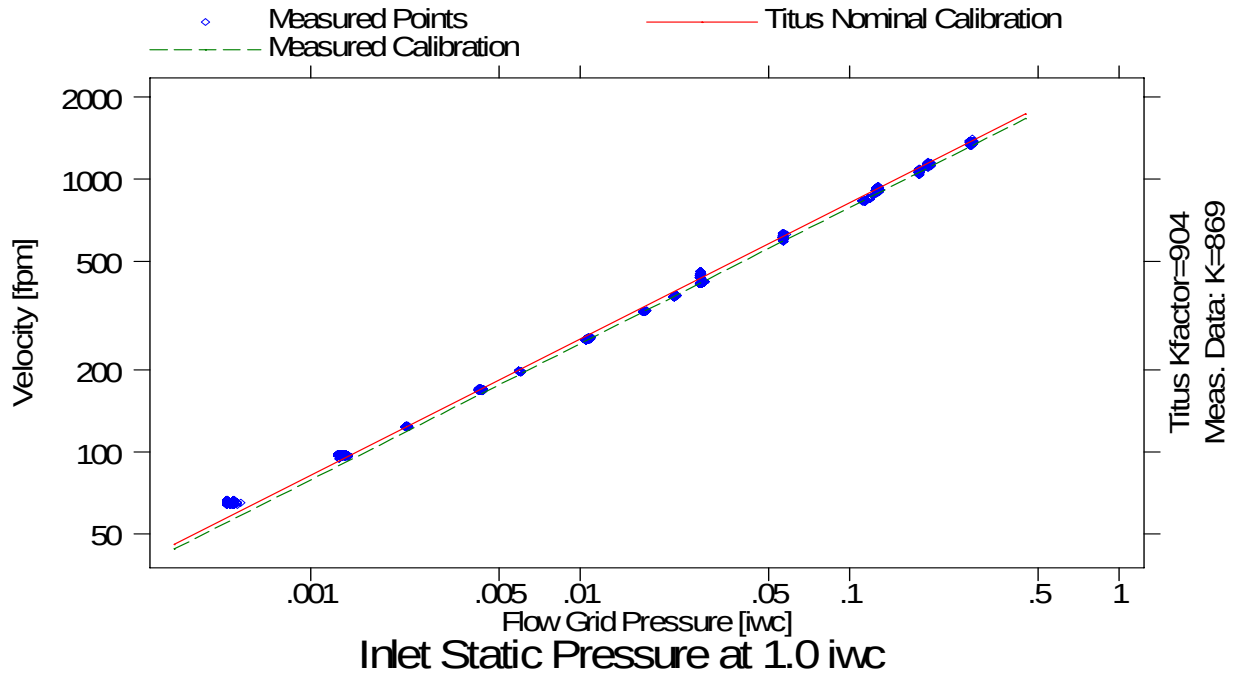


Figure C11: Calibration at various damper positions at an inlet static pressure of 1.0 iwc.

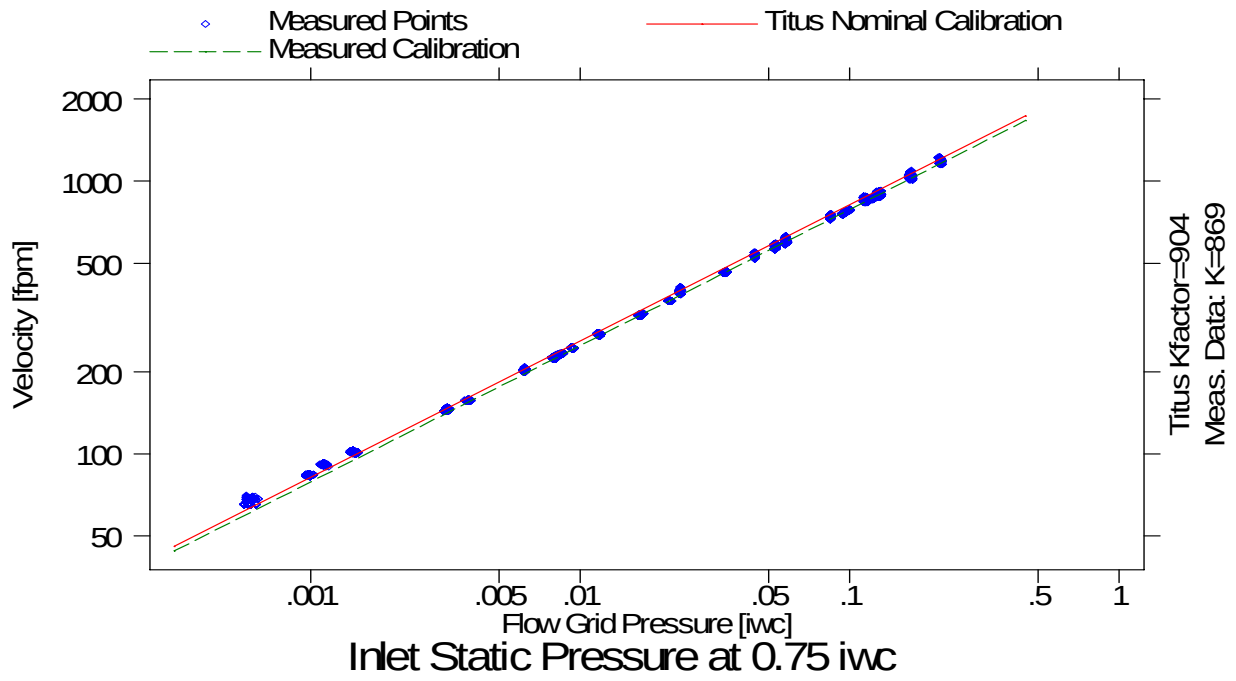


Figure C12: Calibration at various damper positions at an inlet static pressure of 0.75 iwc.

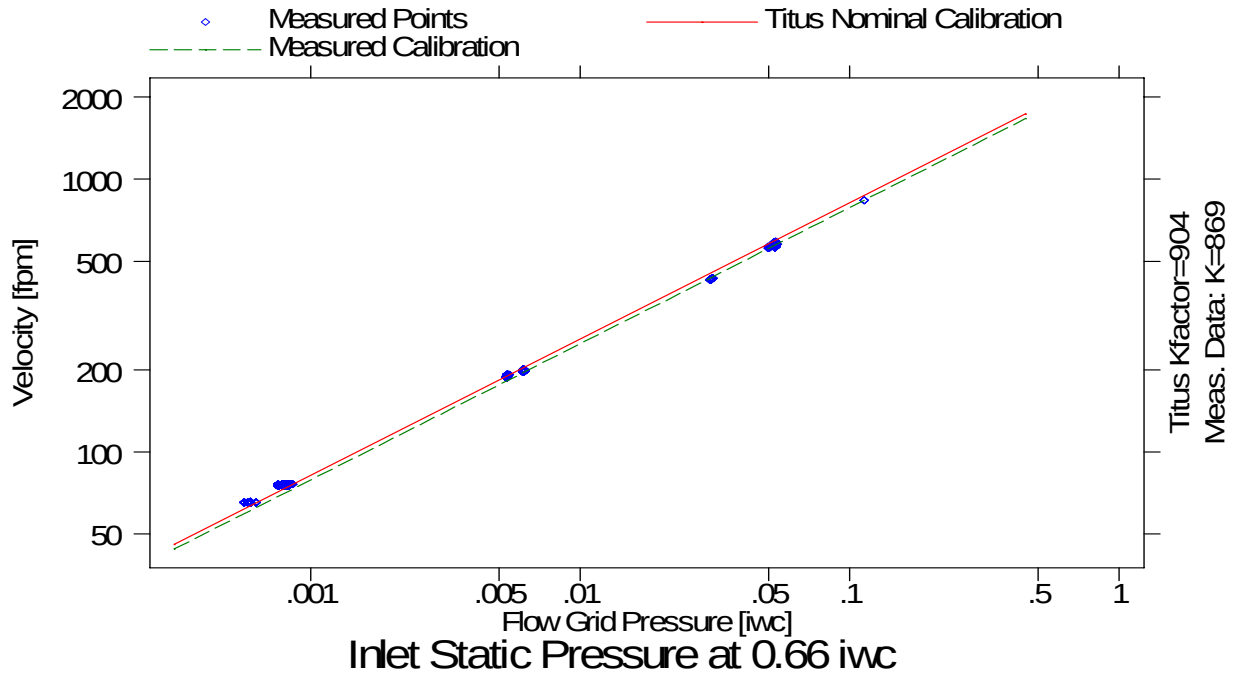


Figure C13: Calibration at various damper positions at an inlet static pressure of 0.66 iwc.

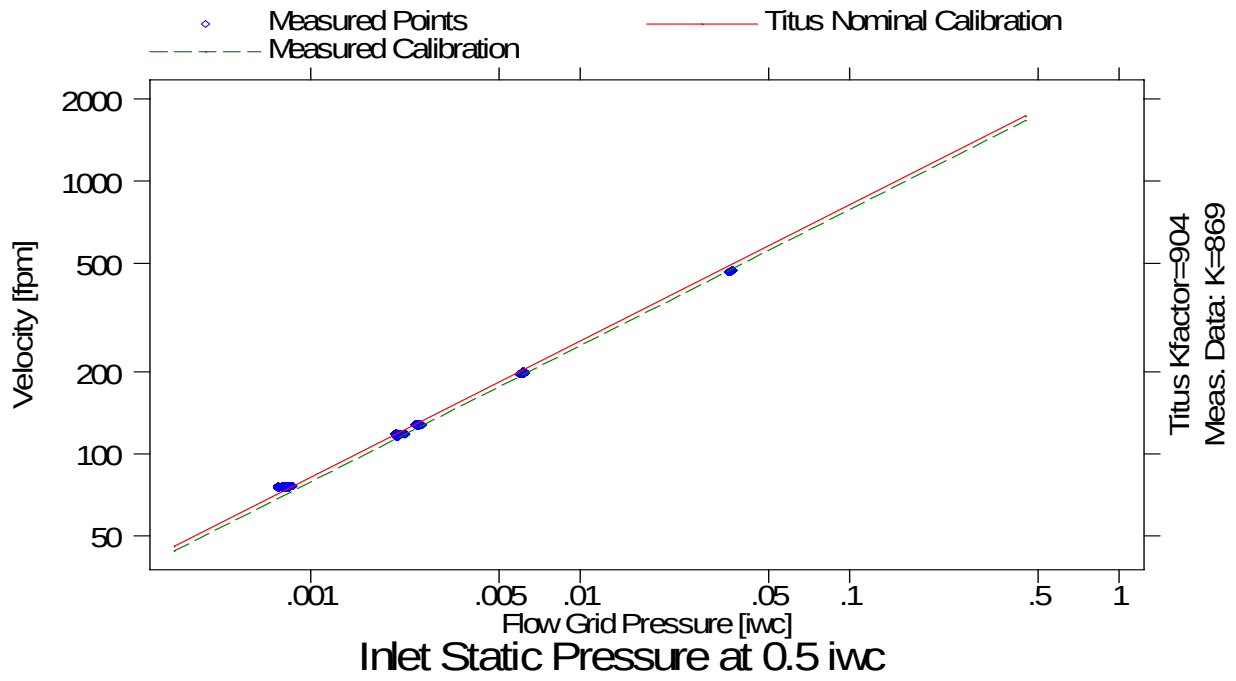


Figure C14: Calibration at various damper positions at an inlet static pressure of 0.5 iwc.

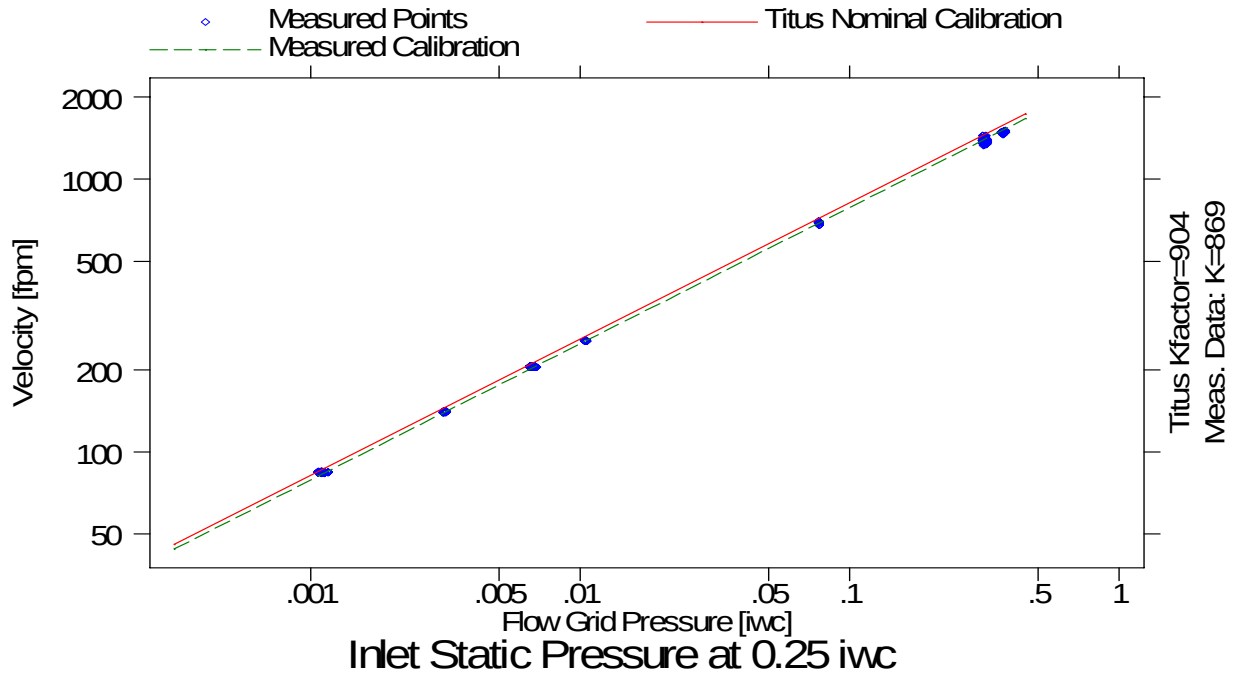


Figure C15: Calibration at various damper positions at an inlet static pressure of 0.25 iwc.

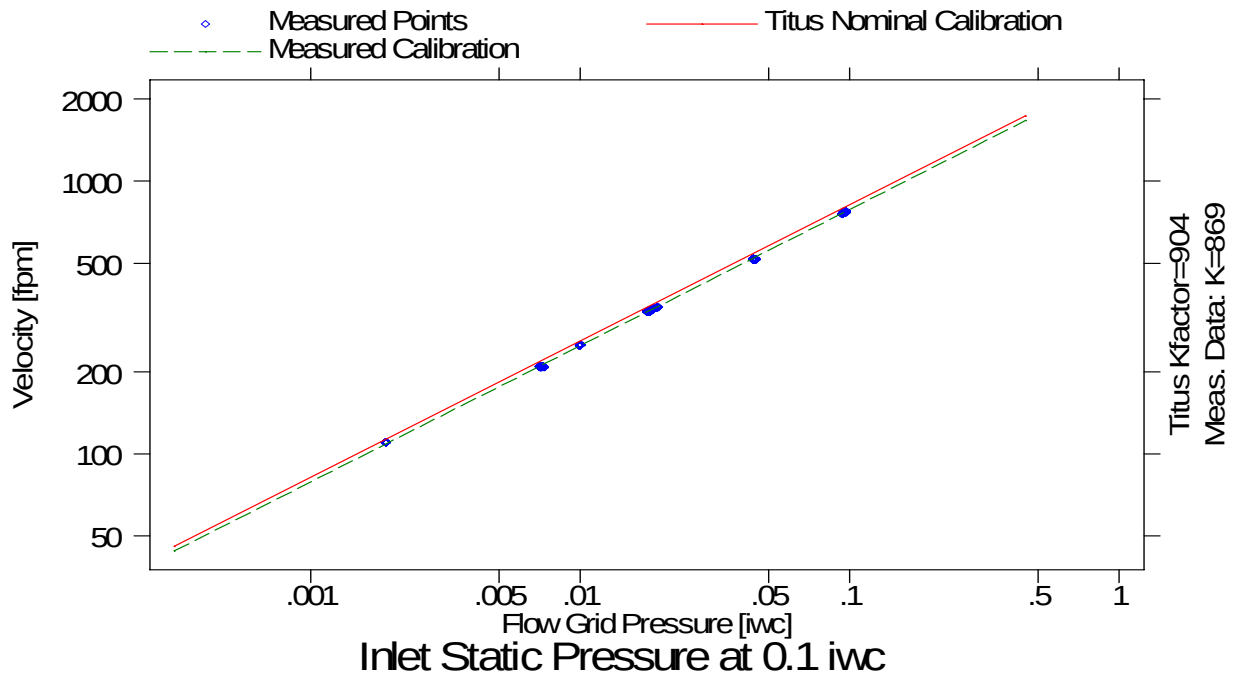


Figure C16: Calibration at various damper positions at an inlet static pressure of 0.1 iwc.

Appendix D: Siemens Controller

Summary

The controllers were tested to determine the accuracy of their flow measurement with static, quickly changing and normal operating conditions. The ability of the controller to obtain a set point flow was determined for a variety of set point flows at different inlet pressures. Measurements of damper position were made to assess excessive or unstable operation of the damper. While some measurements were made at high flows, these tests focus on low flows.

Static and dynamic flow grid pressures were simulated by using pressure generating devices, either the Setra Micro-Cal or similar manual means. Normal operating conditions were investigated by connecting the VAV boxes to the existing HVAC system and/or by using the "Duct Blaster" fan to generate constant inlet pressures. Tests at low flow set points spanning several days were made to evaluate the longer term issues including the problems associated with the zero drift of the pressure sensors used by the controllers to measure flow.

One problem seen with the controller is that the zero of the pressure sensor drifts. The worse case seen had a reported flow of 50 cfm when the reference flow was 0 cfm, and at another time the reference flow was 100 cfm. This error was found to be greatly reduced by installing and using an "Autozero Module" which permitted measurement of the zero more often than the default of twice a day. The zero drift was found to be extremely correlated to changes in ambient temperature and a temperature compensated pressure sensor would also likely help reduce the error caused by zero drift.

The controller was able to stably control the flow (what the controller thought the flow was) at any requested set point flow at any inlet static pressure large enough to make that flow with the damper fully open. The frequency with which the damper was adjusted to maintain the set point flow was somewhat higher at low flows, but not excessively so.

Description of the Siemens Controller

The Siemens controller tested was an APOGEE Products End Devices and Controllers Terminal Box Controllers model #540-100 (Siemens document number 149-171). Accuracy specifications are difficult to find and the following specifications are from a Siemens web document[*]. This document lists the accuracy, assuming an amplification factor of 1, as ±5% from 300 to 4000 fpm, and ±15% from 200 fpm to 300 fpm. No accuracy is given for velocities below 200 fpm. The resolution is listed as 4 fpm. In flow units for our VAV boxes, with an amplification factor of about 2, this becomes ±5% from 73 to 970 cfm, and ±15% from 49 to 73 cfm. The accuracy below 49 cfm is not listed.

[*] Unfortunately the web address and title of this document was not saved and this section of the Siemens web site appears to be permanently broken. It is believed to have been in http://www.sbt.siemens.com/bau/products/HVACend/TEC.asp and then select Terminal Box Controller, All.

The controller operated a GDE131.1P damper actuator (see Siemens document number 155-188). This has the "3-position" control; those are "rotate right", "rotate left", and "stay". The damper position is calculated from the amount of time spent rotating and is reset whenever the command sends the damper to the limit of its travel.
The Siemens software "Insight" was used to send commands to the controller and log trend data. This software was able to tend data in two modes: a time based mode and a change of value (COV) mode. The fastest time based trending was once a minute, while the COV trending could record up to every second. Both possible trend logs had a 4 cfm resolution of flow and a 0.4 % of full scale for the damper position. (Both values are 0.4% of full scale.)

By default the zero of the pressure sensors was measured every 12 hours. This was done by requesting the dampers to fully close and assuming that the resulting pressure was at zero flow. The flow and damper position were not logged when this zeroing was done.

During installation of the controllers the flow zero is measured and then the K factor was adjusted to obtain the flow as measured by the reference flow meter at about 600 cfm. However, in typical commercial installation, this adjustment is usually done based on the test and balancing measurements, i.e. with a flow hood. Some of the flow hoods that are used in these measurements have been shown to have questionable accuracy[†]. These measurements also ignore duct and VAV box leakage, so the K factor adjustment procedure is expected to produce a K factor different from the factory value whenever duct/VAV box leakage is present. For the purpose of insuring that adequate fresh air is delivered to the occupied zone, this adjustment procedure, specifically not including duct and box leakage flows, if it could be done accurately, is good.

Siemens Accuracy

The flow accuracy of the controller was determined by providing the controller with a constant low noise pressure signal and comparing the controller flow, from the trend logs, to the flow determined with the reference pressure sensors. The Setra Micro-Cal was used for this purpose. The Micro-Cal can be programmed to produce a series of equally spaced pressures. It does this by moving a small piston to change the volume of a reference volume plus the tubing which connects to the controller pressure sensor (and in our case the reference pressure sensors). It uses its own internal pressure sensor to monitor the pressure and thus control the piston. If the controller or the tubing used to attach it to the Micro-Cal leaks, the piston must be continuously moved in order to maintain a constant pressure. The Micro-Cal used in these measurements was an early model and occasionally the piston would "stick". Setra has changed the design to reduce this problem.

[†] Wray, C.P.; R.C. Diamond, and M. H. Sherman (2005) *Rationale for Measuring Duct Leakage Flows in large Commercial Buildings* Lawrence Berkeley National Laboratory, Berkeley, CA, LBNL # 58252

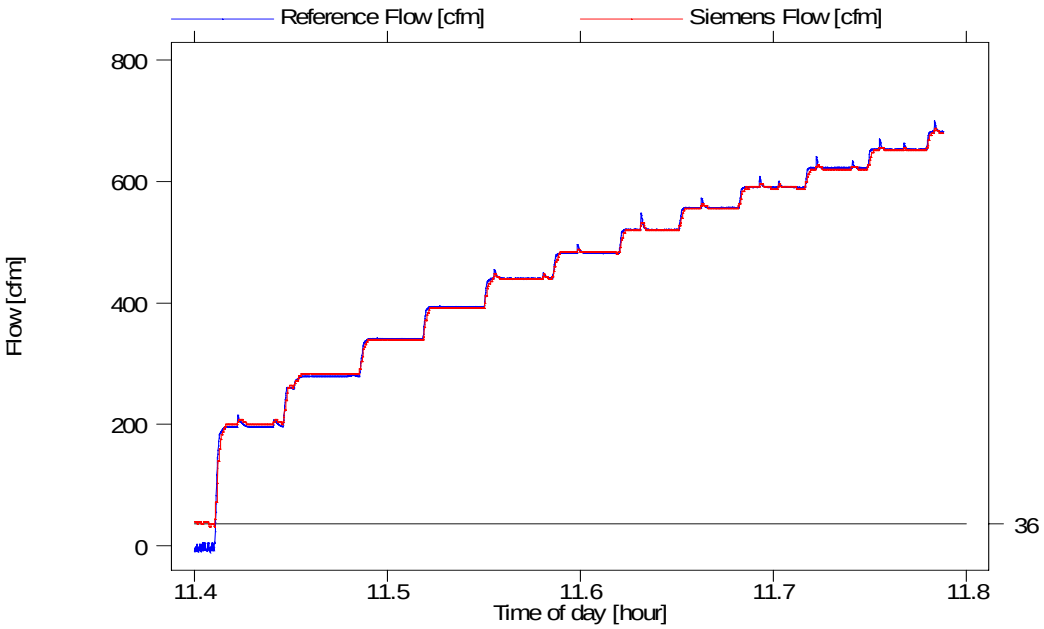


Figure D1: Typical Data for accuracy tests.

Figure D1 illustrates data typical for the Siemens controller. The flow output by the Siemens controller closely matches the reference flow (calculated from the flow grid pressure by the reference pressure sensors) except at zero flow where the Siemens controller shows a marked offset of 36 cfm. An expanded discussion and measurements of the zero offset are presented at the end of this appendix. Because of the square root dependence of flow on pressure this offset has a much smaller affect at higher flows (pressures). Notice that the steps in flow are uneven, while the steps in pressure were 0.05 iwc (12.5 Pa). Occasionally seen are spikes in the data that correspond to the sticking of the Micro-Cal piston. Figure D2 shows a close up showing an "ideal" transition from about 340 cfm to 392 cfm. Horizontal lines have been drawn at 392 and 396 cfm, 4 cfm apart, the logging resolution of the Siemens controller. The transition to 440 cfm shows the problem of the sticky piston both at the beginning and at the end of this flow.

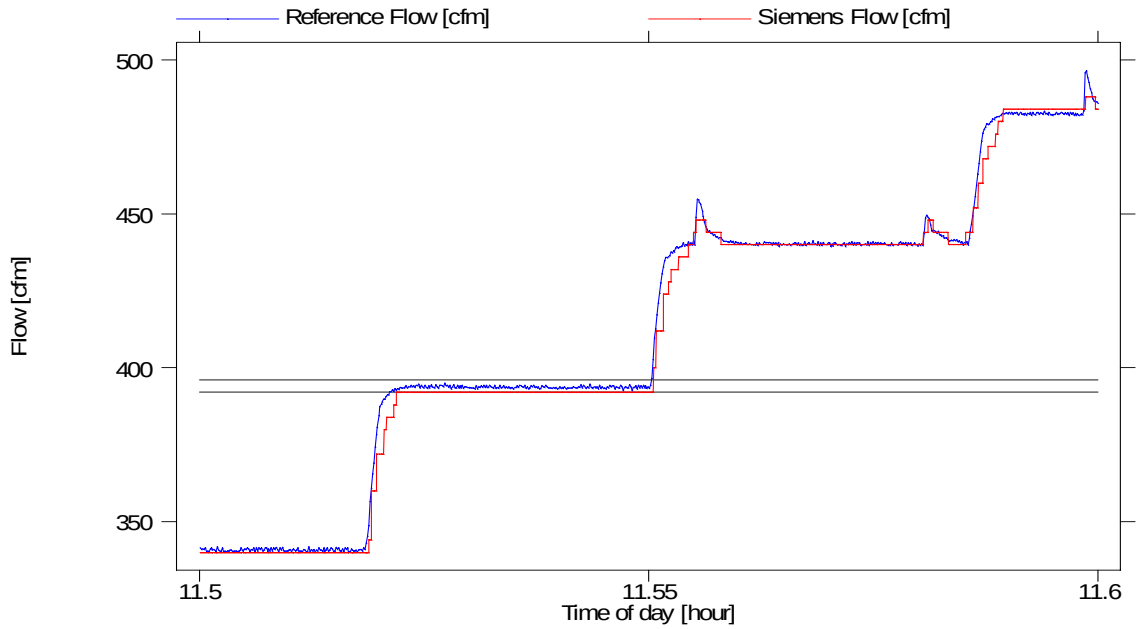


Figure D2: Close up of flow regime transition. Note the spike at about 11.56 when the Micro-Cal piston stuck.

Data similar to that taken as shown in Figures D1 and D2 were obtained for many different pressure (flow) ranges. Data was selected, as shown in Figure D3, from these measurements when the flows, as measured by the controllers, were stable. The data from these stable flow periods was then averaged together and the difference between the controller flow and the reference flow was then calculated. The flow data from the two controllers is labeled as the "Nailor" or "Titus" VAV box as this is the assignment that will be used in tests when the controller is connected to a VAV box, however at this point the two Siemens controllers being tested are connected in parallel to the Micro-Cal.

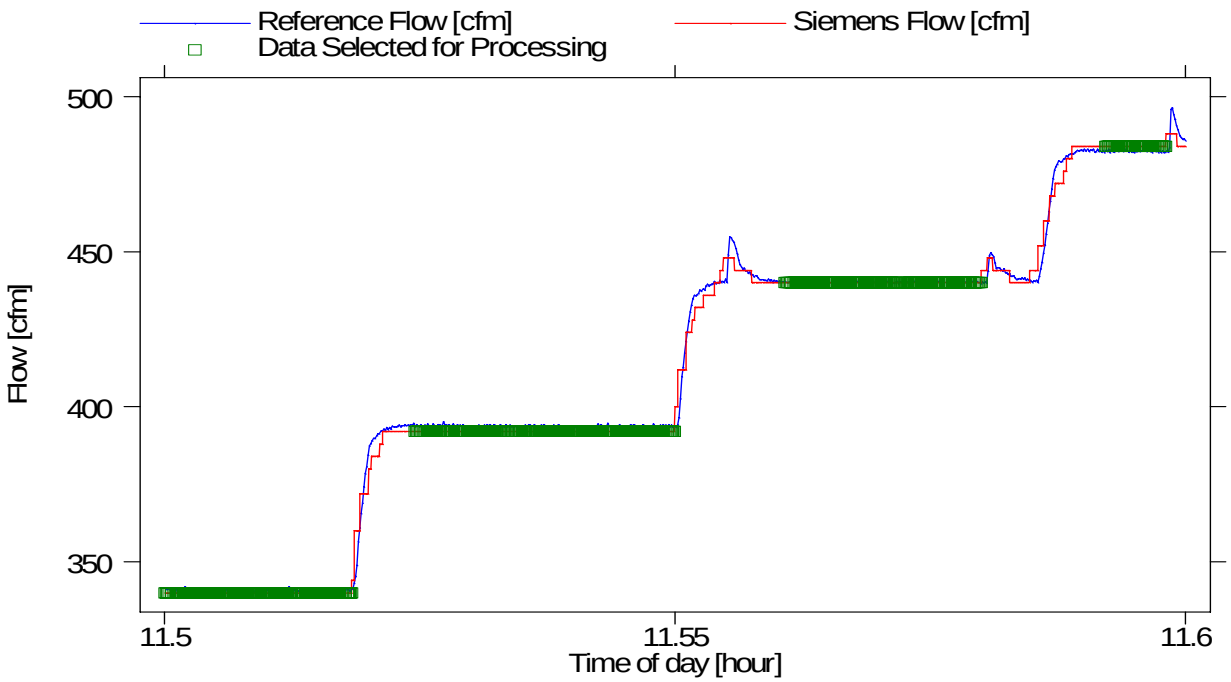


Figure D3 Data marked with the green "o" are determined to be stable flow and have been selected to be used in the analysis of accuracy of the controllers.

Figures D4 shows the error in the Siemens controller flow for the two different VAV box controllers.

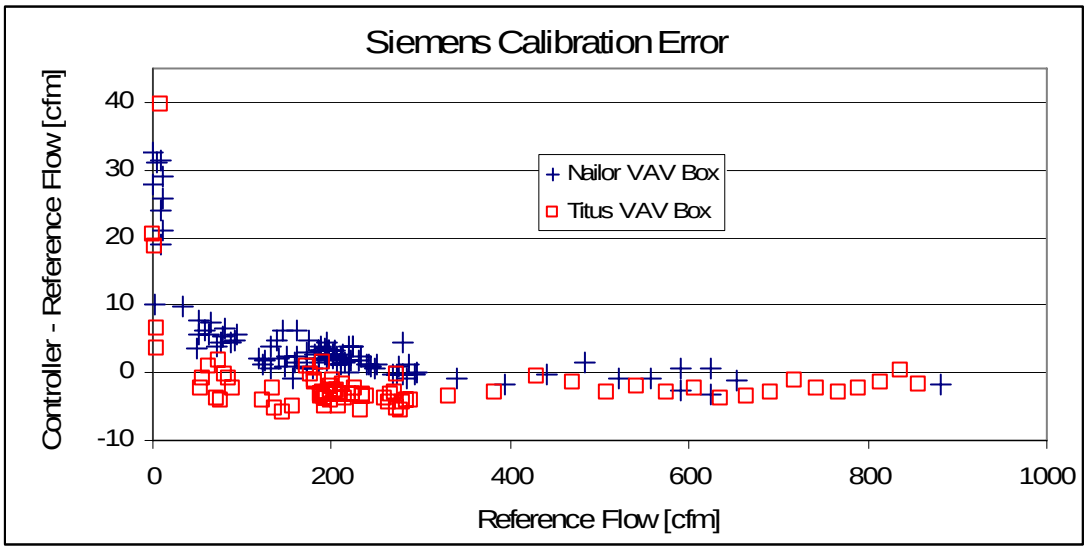


Figure D4: Controller error for the Nailor box

The error is generally within the expected -4 cfm to +4 cfm band for flows above 150 cfm. Almost all the errors for flows between 50 and 150 cfm fall within a -4 to +8 cfm band. Because

the controller zero has significantly drifted, the flows below 50 cfm have large errors. The drift experienced during this measurement time resulted in a calculated flow that was higher than the reference flow, but it could just as easily have been lower.

Siemens Resolution

The resolution of the pressure sensor (or other types of velocity sensors) used in a controller will limit how low a flow it can measure. Typical controllers, like the Siemens unit studied, can measure 1 iwc full scale or more. For the moment let us assume that the pressure sensor (not a hot-wire type sensor) can be perfectly zeroed, thus one would have zero flow at 0 bits from an ADC of any resolution. For the VAV boxes studied the K-factor is about 1000, thus the lowest, non zero, flow that can be measured would be 62.5 cfm for an 8 bit converter, 15.6 cfm for 12 bits, and 3.9 cfm for a 16 bit controller. Figure D5 shows the flow error corresponding to one bit high and one bit low for an 8, 12, and 16 bit controller. The first point plotted is the flow at the least significant bit (a count of 1).

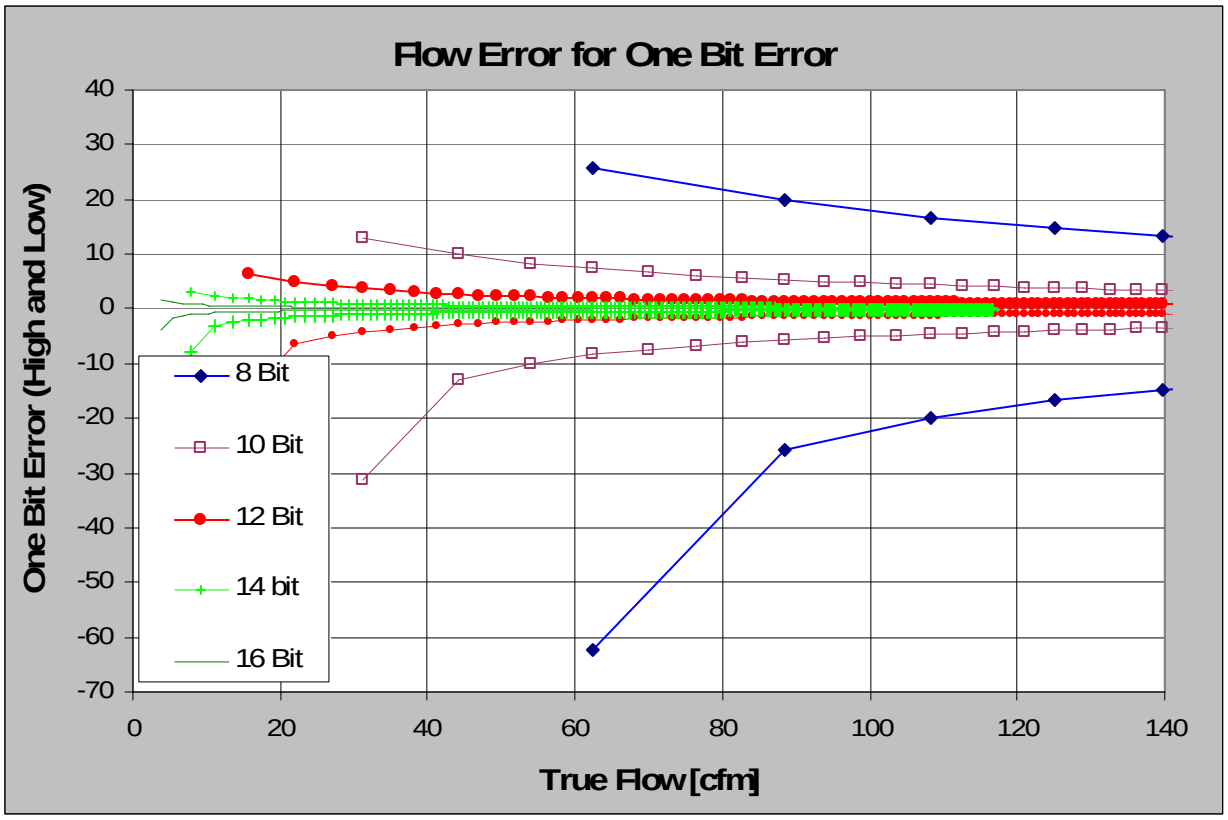


Figure D5: Flow error for one bit high and low, assuming a 0 to 1 iwc generic pressure sensor for a VAV box with a K-value of 1000, for an 8, 12, and 16 bit analogue to digital converter.

Clearly an 8 bit controller will not be able to measure flows below 80 cfm, while a 16 bit controller has the resolution to accurately measure flows down to 10 cfm or lower. In practice a single reading of the flow is unlikely to be used to control the damper of the VAV box. A running average of the pressure combined with a dead band will be used to limit the number of

times the damper position is adjusted. For a discussion of damper controls see the ASHRAE Seminar by Steve Tom, 2003[‡].

The value of the flow reported by a DDC system is not a snapshot at a particular time, with all the noise that this might have, but a value that has been conditioned by some sort of digital filter. Thus the value reported for flow can have values that are intermediate to those corresponding to the ADC counts. The Siemens controller/software reports flow with 4 cfm increments. The data presented here is done in units of cfm instead of pressure because of this limitation of the controller software, but the controller saw a generated pressure, not a flow. Because the software filters the reported flow it is not possible to determine the resolution of the ADC used from the trended data, and the resolution could not be found in the literature.

The experimental plan to investigate the resolution of the controllers was to slowly vary the pressure around a fixed pressure using the Setra Micro-Cal. However the Micro-Cal operates somewhat differently. The Micro-Cal starts at 0 pressure and then quickly changes the pressure to a starting value, it then makes a series of up to 21 equal pressure steps to a final pressure, and optionally, steps back down to the starting pressure, and then goes back to 0 pressure. The time spent at each pressure can be controlled to a point, with the shortest time being the amount of time that its internal pressure sensor takes to make a valid measurement. The longest time is perhaps a minute. For some reason it spends more time at each pressure point when lowering the pressure than when increasing the pressure. Occasionally the piston that makes the pressure sticks and causes a pressure spike. The Micro-Cal then must make adjustments for this pressure spike with the result being that more time is spent at this step than other steps.

Zero drift in the data can be corrected by equation D1 (assuming both input flows are positive):

$$Q = sign(Q_{raw}^2 - Q_{zero}^2) * \sqrt{abs(Q_{raw}^2 - Q_{zero}^2)} \qquad \text{Equation D1}$$

Where Q_{raw} is the flow reported by the controller

Qzero is the flow reported by the controller when the flow is zero

Q is the corrected flow

And the function sign() returns 1 if positive or -1 if negative

The reference, raw and corrected "flows" are shown in Figure D6. We see excellent tracking of the reference flow by the zero corrected Siemens flow, even at flows less than 100 cfm.

[‡] Tom, Steve, ASHRAE Seminar 37, Winter Meeting 2003
VAV Damper Control, Longer Life and Better Precision
http://tc14.ashraetcs.org/presentations/VAV%20Damper%20Control.ppt#349

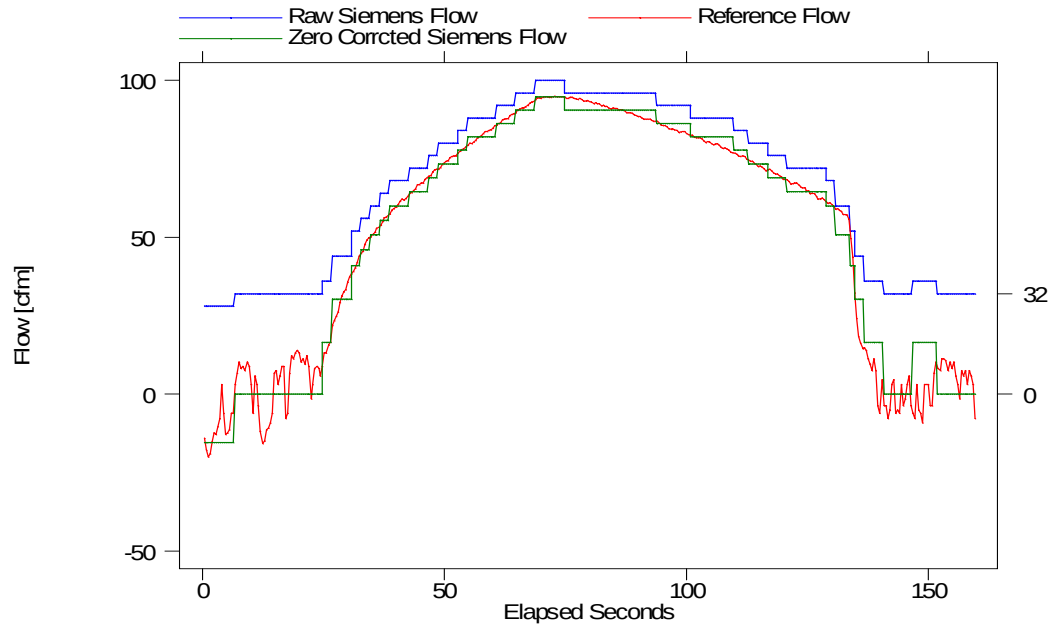


Figure D6: The zero corrected controller flow (using equation D1 with Q_{zero}=32) shows excellent tracking of the reference flow at low flows.

Figure D7a and D7b show the same kind of data at higher flows. The slower, and thus smoother, response of the controller flow is evident in these figures.[§]

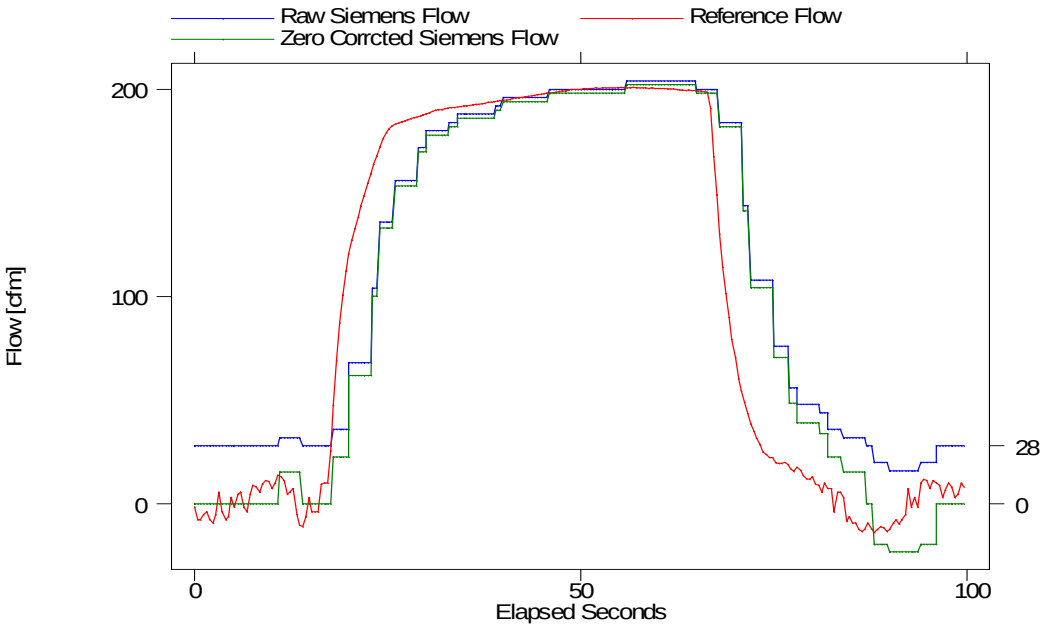


Figure D7a: The controller flow tracks the reference flow (pressure) with a small delay due to filtering (smoothing) of the data.

[§] There may be up to two seconds difference between the two data collection systems, the Siemens software, and the APT data collection.

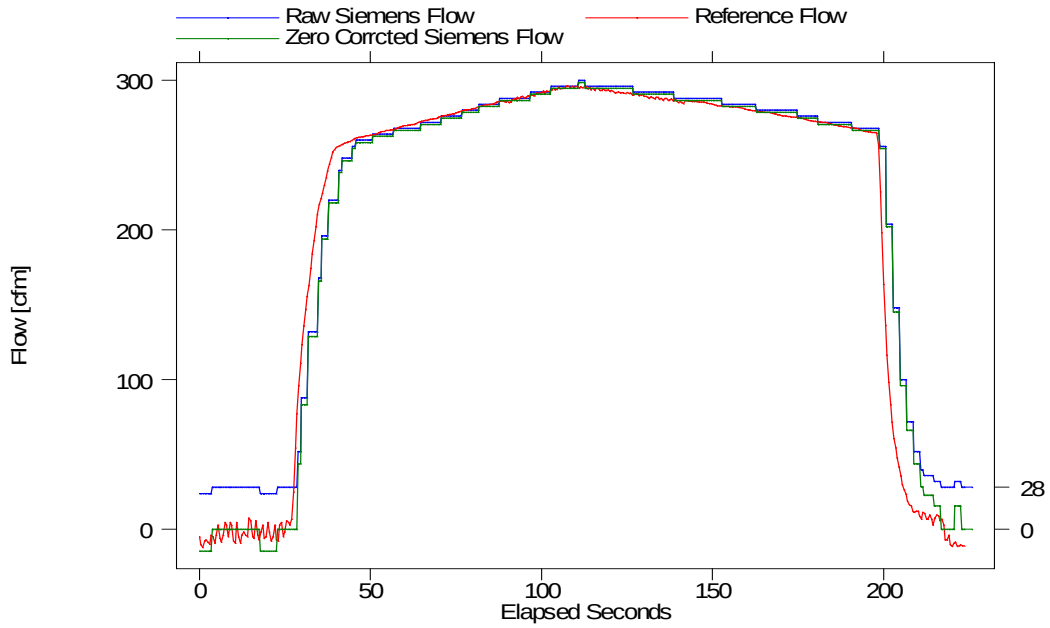


Figure D7b: The zero corrected controller flow shows excellent tracking of the reference flow at mid range flows.

The resolution of the Siemens controller will not impede an accurate measurement of the flow, if the flow (pressure) zero is correct, down to 50 cfm or lower.

Siemens Stability

The measurements for stability are designed to see how the controller's measurement of pressure and calculation of flow respond to rapid changes in the input pressure. The original experimental design was to use the Micro-Cal to generate these pressure changes, but the Micro-Cal can only generate one half cycle and then returns to zero pressure. A manual method was developed instead by making a closed system of tubing connecting the controller pressure sensor to the reference pressures. The pressure in the tubing was then adjusted to a target value by adjusting the insertion length of the tubing on the connection fittings; for instance by pushing the tube onto a "T", the pressure could be increased. The target pressure was then cyclically varied by pinching the tube using a vice-grip.** Figure D8 shows the controller response to a small pressure variation (from 0.016 to 0.08 iwc [4 to 20 Pa])at about 0.1 Hz. The controller flow variation has a smaller amplitude and a slight offset in time compared to the reference "flow".

** One should note that the pressure in a closed system such as this is very sensitive to changes in temperature, and it is very difficult to maintain a constant pressure.

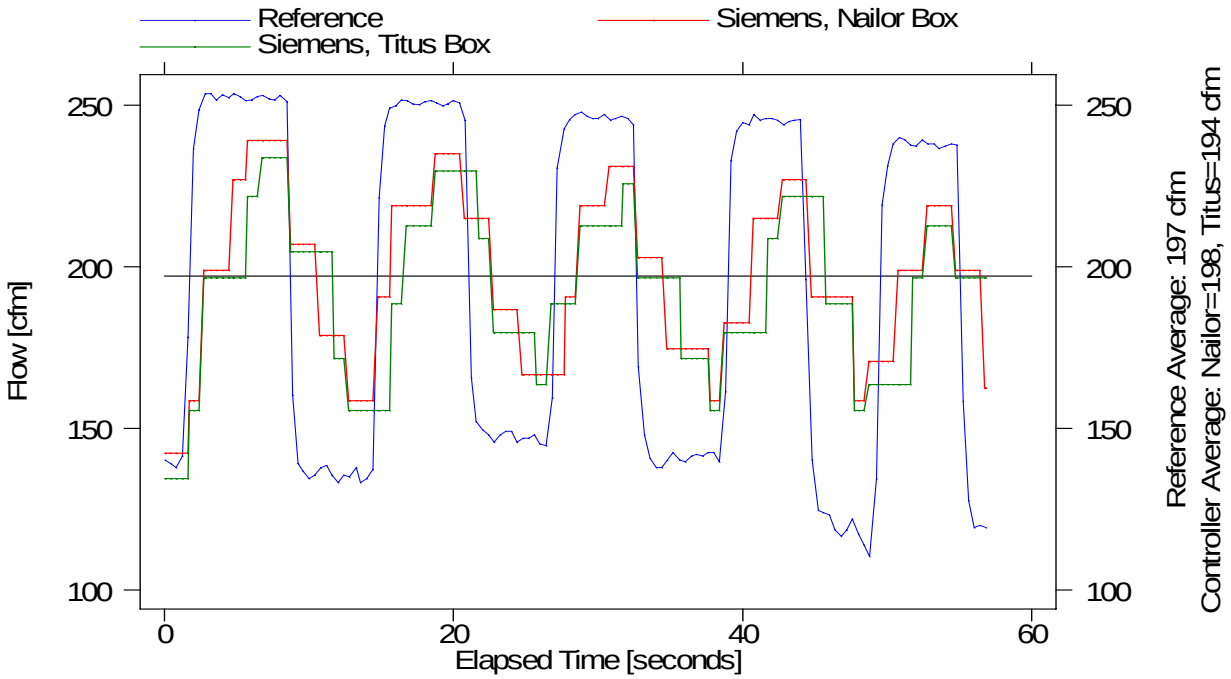


Figure D8: About 0.1 Hz cycling of the pressure signal at a pressure corresponding to an average flow of 195 cfm. The Siemens flow shows a small attenuation of the cyclic portion of the signal.

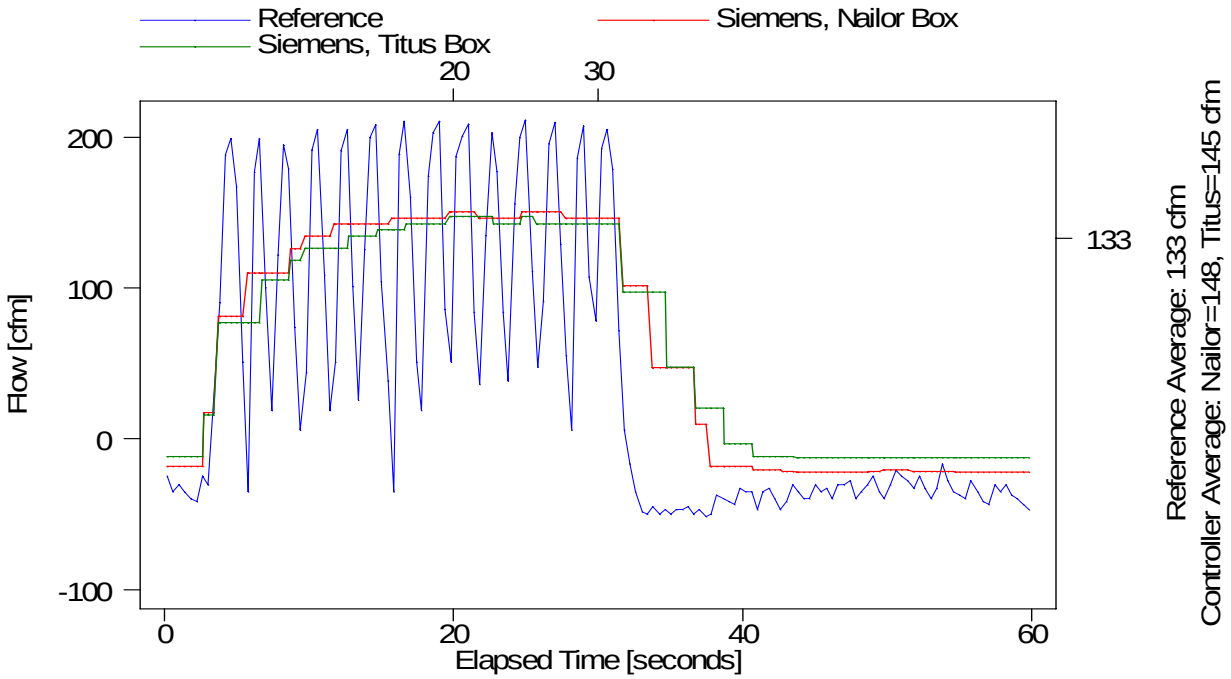


Figure D9: 0.5 Hz cycling of the pressure signal at a pressure corresponding to an average flow of 135 cfm.

Figure D9 shows the controller response to a small pressure variation (from 0.0012 to 0.056 iwc [0.3 to 14 Pa]) at about 0.5 Hz. At this frequency the controller flow output can no longer track

individual cycles but is an acceptable smoothing. The average flow, calculated between 20 and 30 seconds, is somewhat higher (145 and 148 cfm) than the reference flow (133 cfm). Figure D10 shows a similar test made at higher "flows". In this case all the average flows, measured between 17 an 55 seconds, have the same value of 471 cfm.

Note that the pressure before and after the cycles in Figure D9, is negative, as evidenced by the negative reference flows. The Siemens controller/software could not calculate a flow less than 0. Any negative value was set to 0. When the pressure signal is noisy enough there can be negative values in the noise. By setting these values to 0 the calculated average is higher than the correct average. This possibly explains the slightly higher controller values seen in Figure D9. If so the actual pressure sensor used in the controller would have to have a faster response time than the reference pressure sensors. This was impossible to directly determine.

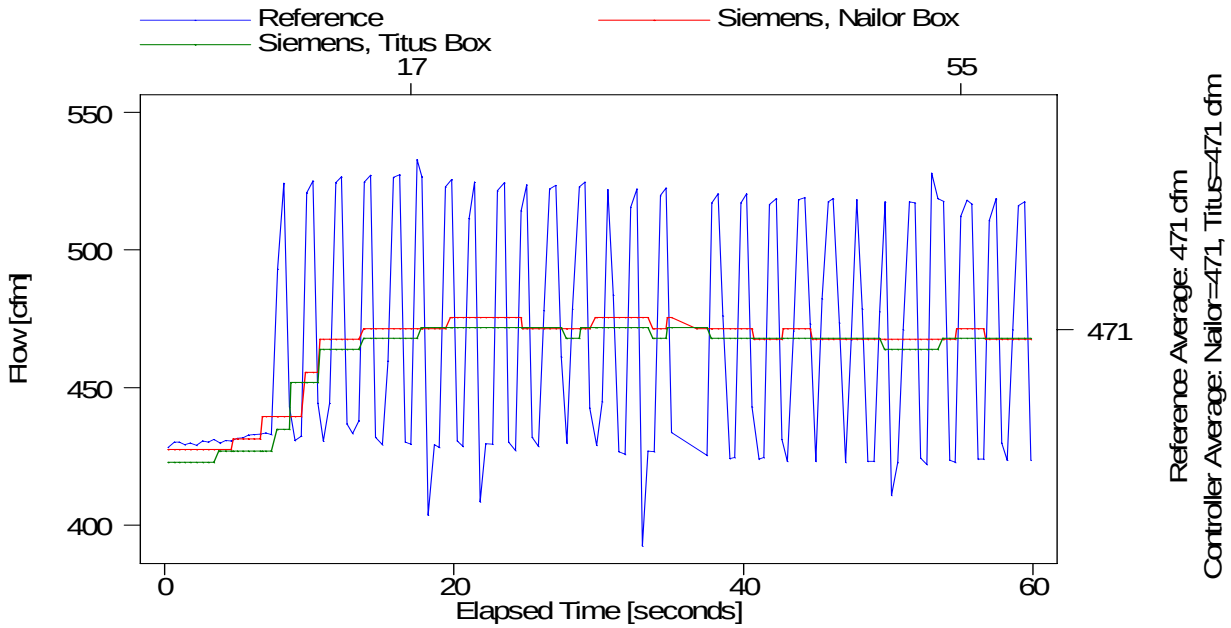


Figure D10: At higher flows the average of each flow has the same value of 471 cfm.

Note the pause in the reference flow at about 35 seconds where the APT rezeroed the pressure sensors.

Siemens Complete System Test: Titus VAV box

The complete system tests are intended to show how the controller operates with real flows instead of the pressures generated by the Micro-Cal or by manual means. In these tests the controller was allowed to move the VAV damper to adjust the flow to the flow set point. Tests were made at several different duct inlet static pressures. Each static pressure was maintained for about 10 minutes after the controller had stabilized the flow after a step change in the static pressure. Figures D11 to D15 show the flows, damper position and inlet duct static pressure for various flow set points for the Titus VAV box. The vertical lines delimit the time periods used in the analysis. The data may have gaps in time where the flows went beyond the plot limits. Table D1 summarizes this data.

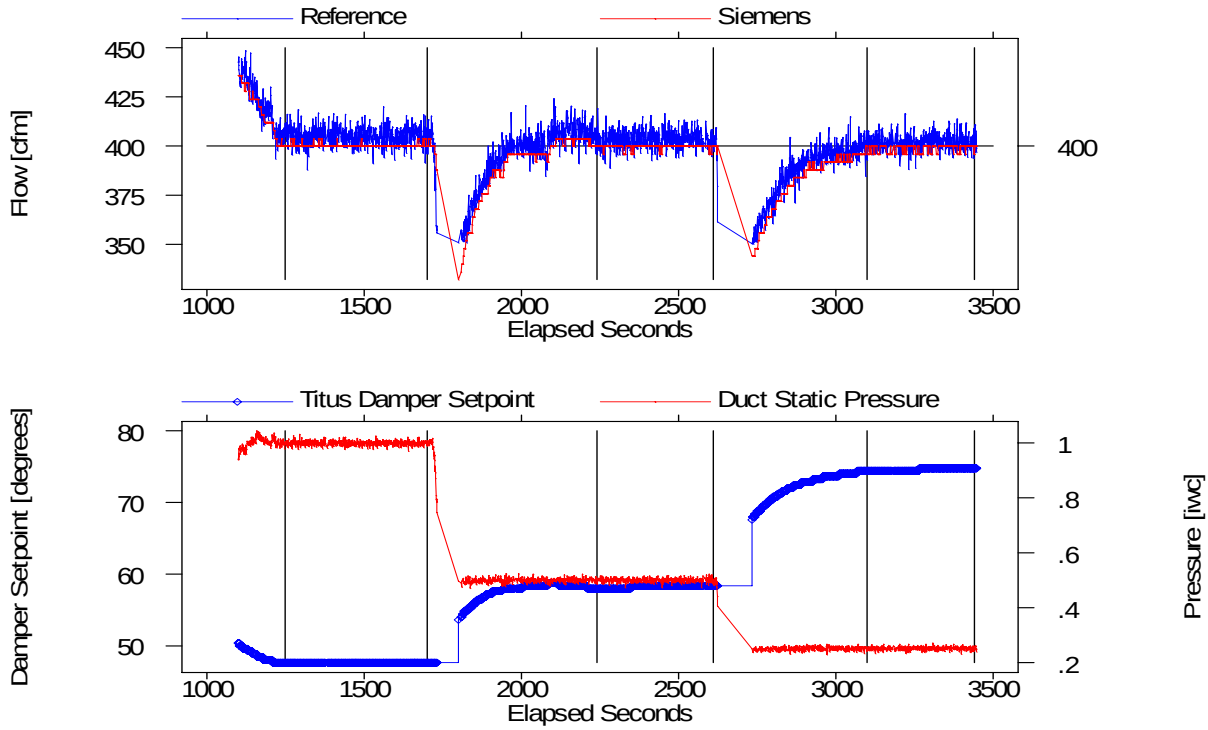


Figure D11: Controller response to changes in the duct inlet static pressure at a set point of 400 cfm on the Titus VAV box.

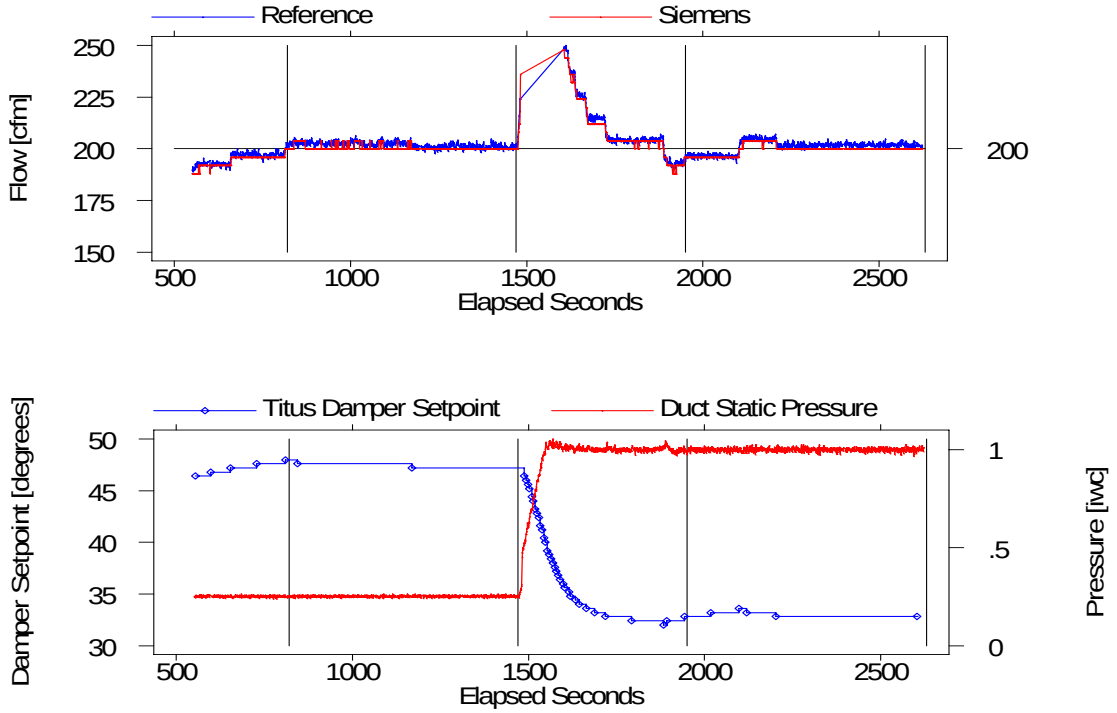


Figure D12: Controller response to changes in the duct inlet static pressure at a set point of 200 cfm on the Titus VAV box.

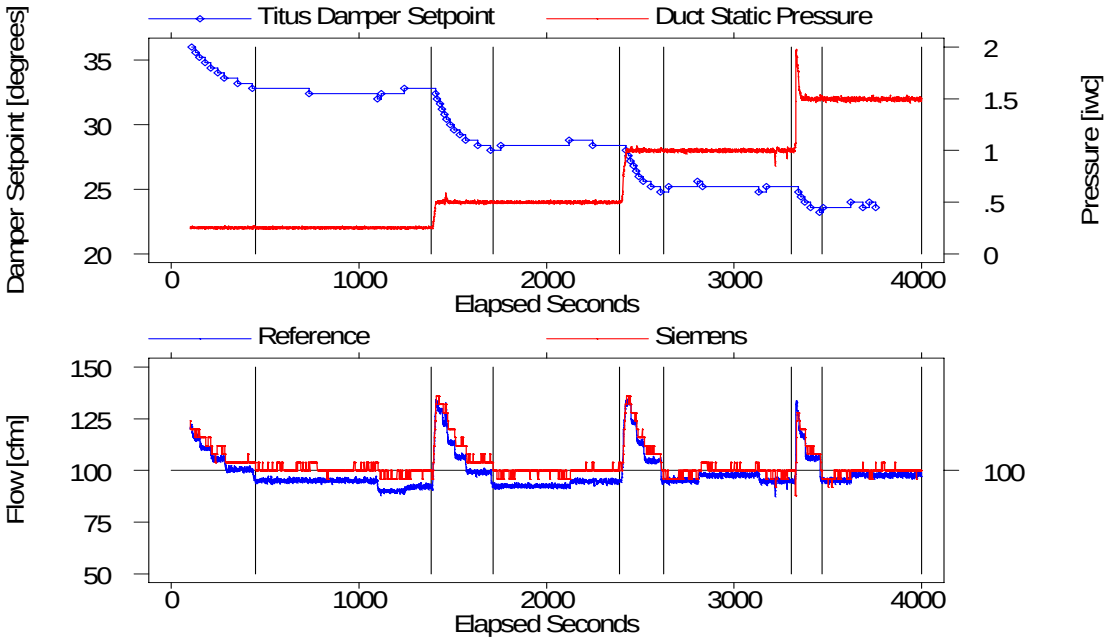


Figure D13: Controller response to changes in the duct inlet static pressure at a set point of 100 cfm on the Titus VAV box. Recall that this damper is fully sealed at about 17.5 degrees.

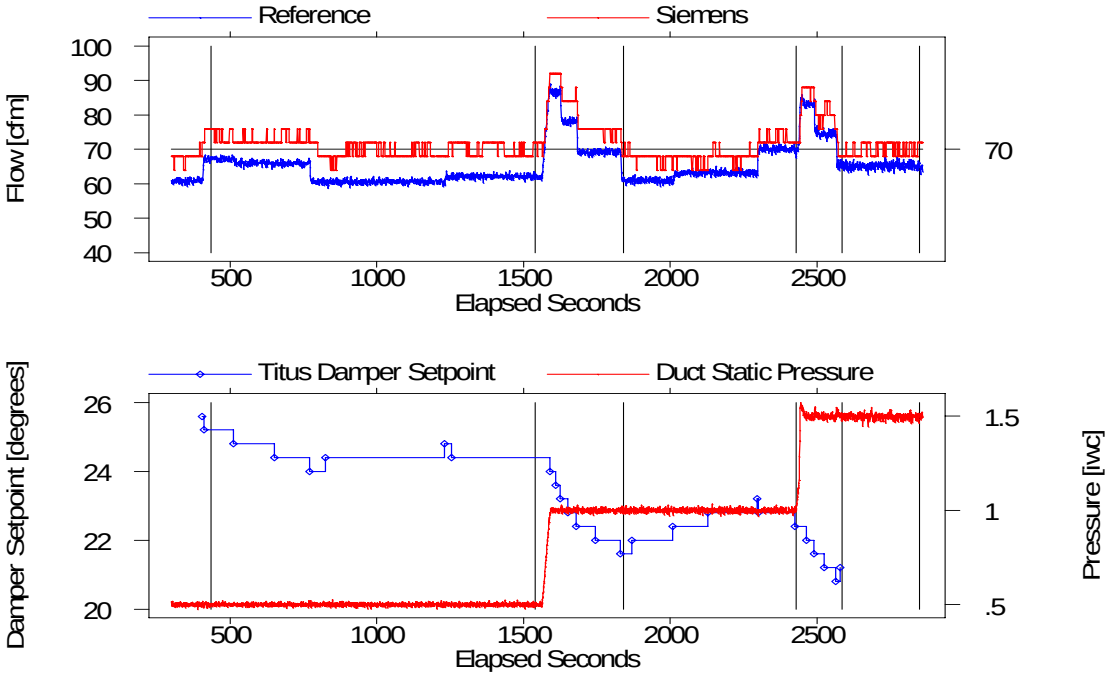


Figure D 14: Controller response to changes in the duct inlet static pressure at a set point of 70 cfm on the Titus VAV box.

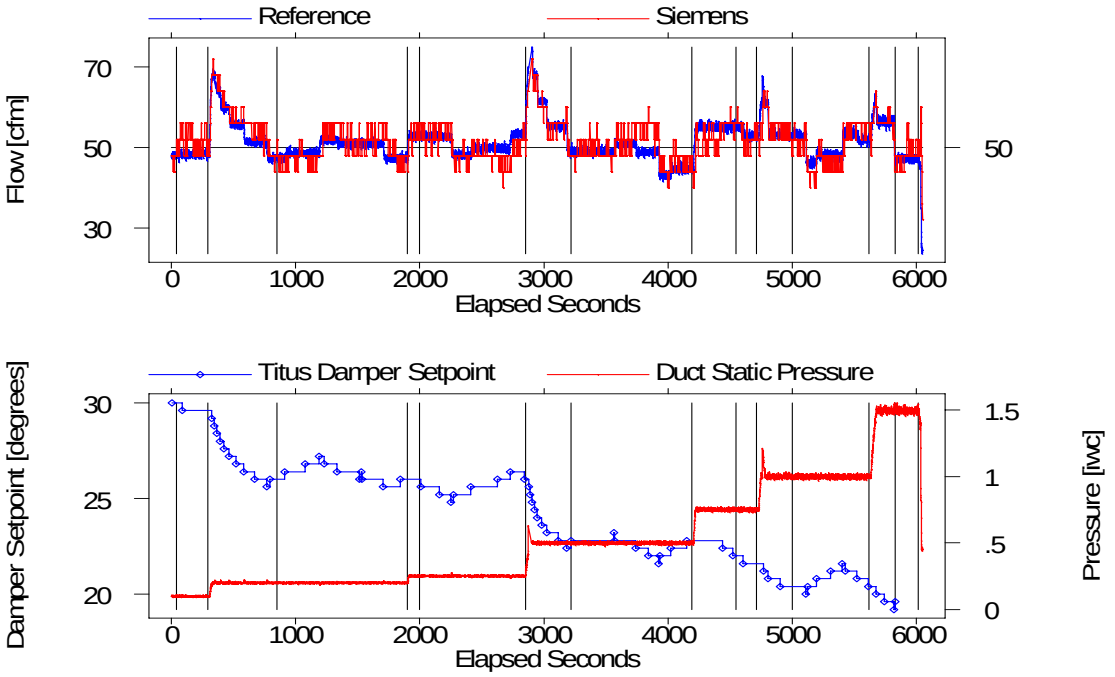


Figure D15: Controller response to changes in the duct inlet static pressure at a set point of 50 cfm on the Titus VAV box. The controller had been rezeroed just before these measurements. Recall that this damper is fully sealed at about 17.5 degrees.

A summary of these data is in Table D1. The controller was able to adjust the average controller measured flow to within 3 cfm of all set points. At the lowest set point tested, 50 cfm, the average flow over all tests was 50 cfm. The differences between the set point and the reference flow for individual tests (different inlet duct static pressures) could be due to the relatively short duration of each test. The only significant difference between the set point flow and the reference flow occurs at the 70 and 100 cfm set points. The controller pressure sensor was auto zeroed just before the tests at 50 cfm and about two hours before the 70 cfm tests, and 3.5 hours before the 100 cfm tests. It is possible that the controller pressure sensor had drifted enough to be the cause of the observed differences between the controller and reference flows.

There is a general trend for more frequent damper adjustments at lower flows. Longer term tests will need to be made in order to see if these adjustments settle down.

Table D1: Siemens Controller Operating the Titus VAV Box.								
Set point Flow [cfm]		Duct Inlet Static Pressure [iwc] 0.1	0.2	0.25	0.5	0.75	1	1.5
400	Siemens Flow [cfm]			399	400		400	
	Reference Flow [cfm]			402	404		405	
	Damper Position [degrees]			74.6	58.3		47.6	
	# of Damper Adjustments			1	1		0	
	Damper Adjustments/hour			11	11		1	
	Minutes of Data			5.7	6.1		7.5	
200	Siemens Flow [cfm]			201			200	
	Reference Flow [cfm]			202			201	
	Damper Position [degrees]			47.5			32.9	
	# of Damper Adjustments			2			5	
	Damper Adjustments/hour			12			28	
	Minutes of Data			10.8			11.2	
100	Siemens Flow [cfm]			100	100		99	99
	Reference Flow [cfm]			94	93		96	97
	Damper Position [degrees]			32.6	28.4		25.2	23.6
	# of Damper Adjustments			4	3		5	5
	Damper Adjustments/hour			16	17		27	35
	Minutes of Data			15.6	11.2		11.3	8.8
70	Siemens Flow [cfm]				71		69	70
	Reference Flow [cfm]				63		64	65
	Damper Position [degrees]				24.6		22.5	1.2
	# of Damper Adjustments				6		6	0
	Damper Adjustments/hour				21		38	1
	Minutes of Data				18.4		9.8	4.4
50	Siemens Flow [cfm]	52	50	50	50	53	50	48
	Reference Flow [cfm]	48	50	51	48	54	50	47
	Damper Position [degrees]	29.7	26.3	25.6	22.5	21.7	20.9	19.6
	# of Damper Adjustments	1	10	8	8	1	8	0
	Damper Adjustments/hour	15	35	35	31	24	47	1
	Minutes of Data	4.1	17.5	14.1	16.2	2.6	10.3	3.1

Siemens Complete System Test: Nailor VAV box

The Nailor VAV box was tested in a similar manner as described above for the Titus VAV box complete system test. A potentially important difference between the boxes is that the Nailor damper travel is 45 degrees instead of the 90 degrees for the Titus. This might lead to control problems. As seen in Figure D16, the response time of the controller to a step change in inlet pressure seems somewhat longer than for the Titus VAV box.

Figure D17 to D19 and Table D2 shows the results of these tests.

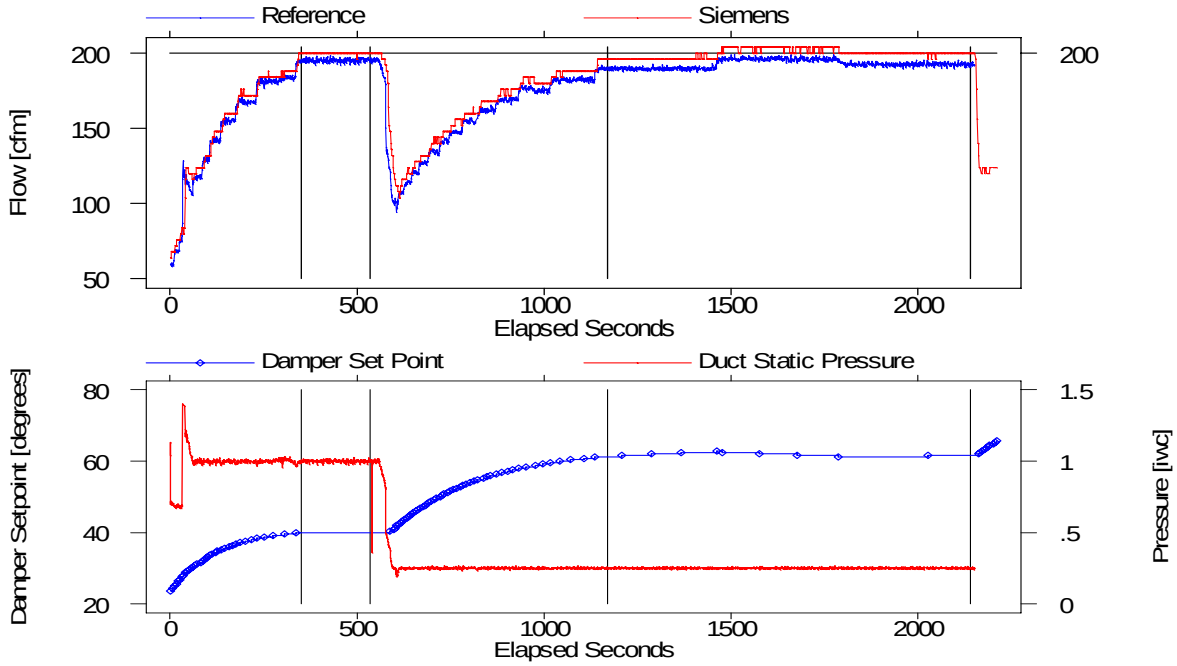


Figure D 16: Controller response to changes in the duct inlet static pressure at a set point of 200 cfm on the Nailor VAV box.

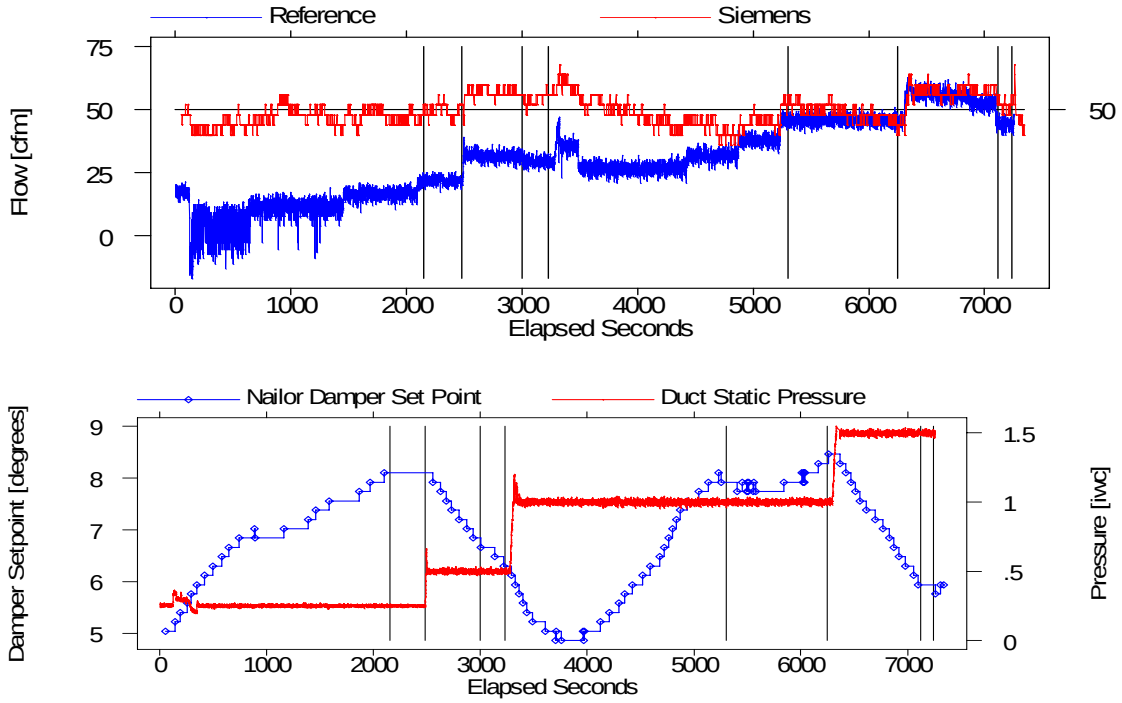


Figure D 17: Controller response to changes in the duct inlet static pressure at a set point of 50 cfm on the Nailor VAV box. It is not clear if the damper position has stabilized after a step change in inlet static pressure except for the time from about 5200 to 6200 seconds.

Table D2: Siemens Controller Operating the Nailor VAV Box.
Shaded cells indicate too few damper adjustments to determine rate.

		Duct Inlet Static Pressure [iwc]				
Set point Flow [cfm]		**0.15**	**0.25**	**0.5**	**1**	**1.5**
200	Siemens Flow [cfm]		200		200	
	Reference Flow [cfm]		193		195	
	Damper Position [degrees]		27.8		18	
	# of Damper Adjustments		9		0	
	Damper Adjustments/hour		34		N/A	
	Minutes of Data		16.1		3.1	
70	Siemens Flow [cfm]	70				
	Reference Flow [cfm]	77				
	Damper Position [degrees]	34.2				
	# of Damper Adjustments	31				
	Damper Adjustments/hour	3				
	Minutes of Data	662				
50	Siemens Flow [cfm]		50	54	49	51
	Reference Flow [cfm]		22	30	46	44
	Damper Position [degrees]		8.1	6.6	7.9	5.9
	# of Damper Adjustments		1	3	16	1
	Damper Adjustments/hour		N/A	62	64	N/A
	Minutes of Data		5.5	3.8	15.8	2

Data for the 70 cfm set point was taken from an 11 hour section of the long term test shown in Figure D18. The data for a set point of 50 cfm at 0.5 iwc inlet pressure did not have a stable damper position, however the flows appear stable.

Siemens Long Term Control and Pressure Sensor Zero Drift

To more fully evaluate the long term behavior of the controllers a several day test was performed over a weekend. The set point of both controllers was set to 70 cfm and the reference flow meter fans (the Duct Blasters) were set to a low speed. The Nailor fan was set to maintain an inlet static pressure of 6 Pa, while the Titus fan was set at a low uncontrolled speed. The amount that the damper had to move to maintain the set point flow is typical of normal operating conditions at very low inlet static pressures for the Nailor VAV box but is not typical for the Titus VAV box because of the drift in the static pressure. The HVAC system is normally off during the weekend, but it came on during this Saturday daytime and again on Monday morning. When it is on, the inlet static pressure is increased and the dampers close to maintain the set point flow.

Figure D18 shows the controller reported and reference flows as well as the inlet static pressure and VAV box damper positions during the long term test. The controllers perform a zeroing

function at about midnight and noon each day. This is done by fully closing the dampers and assuming that the resulting pressure signal is at zero flow. Any leakage at the dampers will cause an error which will scale with the square root of the inlet static pressure. Visual inspection showed that the Titus damper was very tight but the Nailor damper had some leakage. See Appendix C for measurements of the Nailor damper leakage.

We can calculate the amount of the zero drift, from the last time the controller performed its zeroing, from the difference between the measured reference flow grid pressure and the controller pressure derived from the reported flow. The resulting values, shown in Figure D18, show a very strong correlation to the ambient temperature. The "zero drift" is reset to approximately 0 when the controller measures its "zero" at about midnight and noon each day. On average the pressure sensor zero was incorrect by .0024 iwc for the controller sensor looking at the Nailor VAV box, and .0035 iwc for the controller sensor looking at the Titus VAV box. The maximum drift seen during this test is for the controller of the Titus VAV box after about 1.3 days and is about -0.01 iwc. At this time the reference flow is 108 cfm and the reported flow is 50 cfm. Table D3 shows the flow error that this pressure zero offset would have produced at other "controller reported flows".

Table D3: Flow Errors at Different Pressure Offsets.

Offset Pressure [iwc]	Reported Flow [cfm]	Correct Flow [cfm]	Flow Difference [cfm]
-0.01	25	99	-74
(largest seen)	50	108	-58
	100	139	-39
	150	178	-28
	300	315	-15
	600	608	-8
	900	905	-5
-.003	25	58	-33
(average error)	50	73	-23
	100	113	-13
	150	159	-9
	300	305	-5
	600	602	-2
	900	902	-2
+.003	25	0	25
	50	0	50
	100	85	15
	150	140	10
	300	295	5
	600	598	2
	900	898	2
+0.01	25	0	-25
	50	0	-50
	100	28	72
	150	115	35
	300	284	16
	600	592	8
	900	895	5

-

If the temperature had been rising thus causing the zero drift to be a positive 0.01 iwc there would in reality have been no flow until the "controller reported flow" was greater than 96 cfm.

Also of concern is the amount that the damper is adjusted to maintain the flow. During the long term test the damper was adjusted at an average rate of 10 and 5 changes per hour for the Titus and Nailor VAV box respectively. This is a lower rate than were seen during the short term tests and is probably a result of the very low inlet static pressures used in these tests. The Nailor VAV box damper is adjusted less frequently because its inlet static pressure was kept at a nearly constant value, while the inlet static pressure for the Titus VAV box varied.

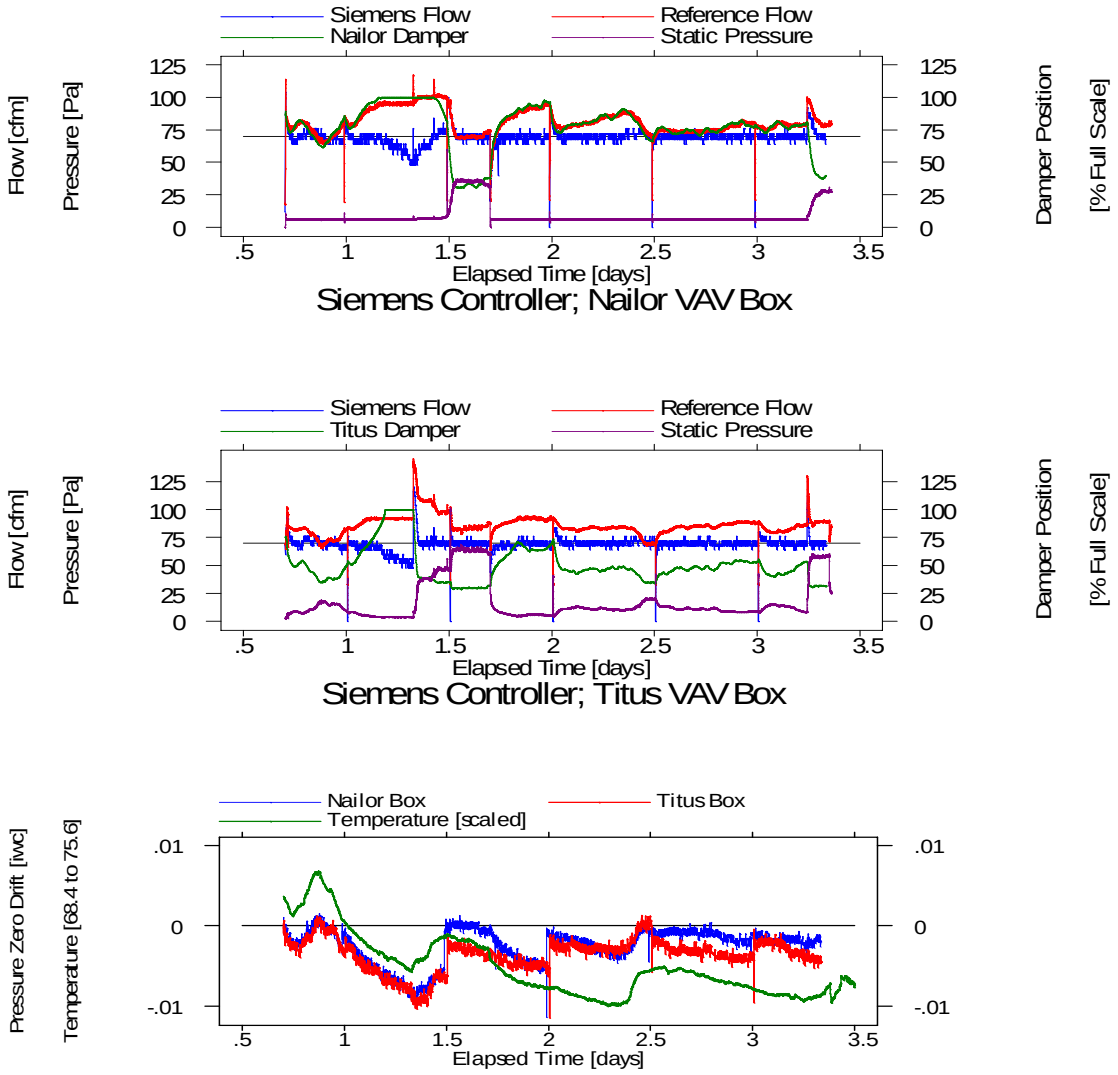


Figure D18: Long term controller test at 70 cfm set point. Flows, static pressures, damper positions, zero drift and temperature. By default the zero is measured twice a day and the reference and controller flows closely agree right after these times.

Pressure Shorting Valve

These controllers have an option of using a pressure shorting valve to measure the zero more frequently without disturbing the flow. Figure D19 shows the use of this option with the zero being measured every hour (any interval could be selected). The flows are maintained at the target of 70 cfm (the controller flow value in the trend log goes to zero when the pressure sensor zero is being measured but the flow does not) and the pressure "zero drift" is much smaller than without this option. The only significant pressure "zero drift" occurs when the temperature suddenly drops, but the values recover the next time the zeros are measured.

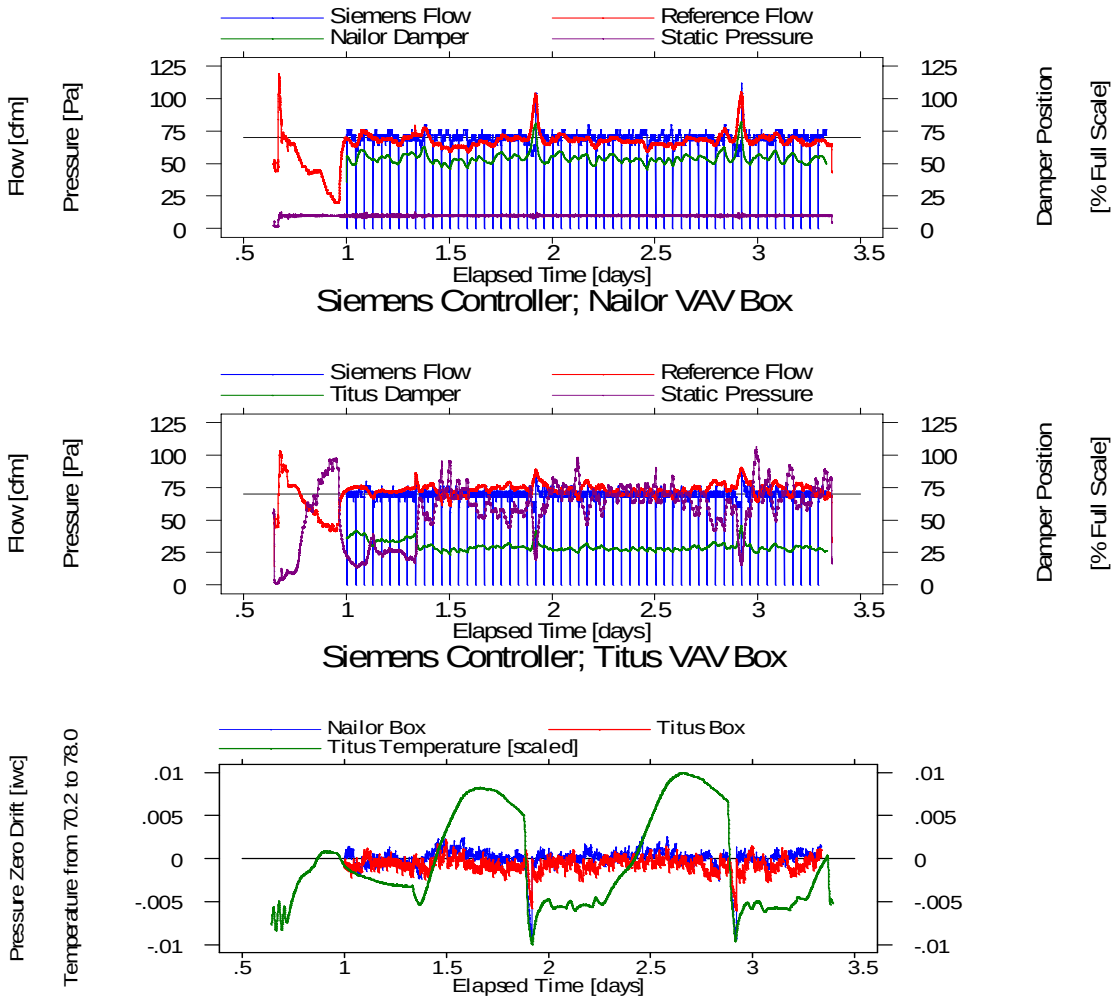


Figure D19: Long term controller test at 70 cfm set point. Flows, static pressures, damper positions, zero drift and temperature with the "auto zero" option.

Appendix E: Alerton Controller

Summary

The Alerton controller uses the pressures generated by the flow grid in the VAV box to induce a small flow across a hot wire type sensor in their controller. This air speed is then appropriately scaled to determine the flow rate of the VAV box. The specifications usually are sited as an equivalent pressure (i.e. 1.25 inches of water full scale).

The difference between the reported flow and the reference flow show that these sensors are not linear at low flows. Because of this nonlinearity the calibration procedure can play a significant role in determining the overall accuracy. The normal in-situ calibration consists of two flows: one at zero and one at the maximum flow. The controller is highly accurate at the two calibration points. Between these two points it under estimates the actual flow. Thus there is little risk of not providing the required ventilation even if the minimum flow setpoint is set exactly to the ventilation requirement because the controller always provides a little more air than it thinks it is providing.

Alerton has an optional multipoint calibration, which was not evaluated. ALC, which also uses a hot wire sensor, uses a four point calibration. It is assumed here that the Alerton controller with a multipoint calibration would perform similarly to the ALC (see Appendix G).
These sensors seem stable, they do not show signs of "zero" drift, and with the proper in-situ calibration can be expected to measure low flows with reasonable accuracy. They do exhibit a tendency to over predict the flow during periods of rapid flow cycling, however, this is not expected during normal operation.

The control uses a dead band of 3% of range. This might be unacceptably large for low flow applications. For example, suppose the design flow was 600 CFM, the minimum flow was 60 CFM, the current setpoint was 60 CFM, the controller measured flow was also 60 CFM, and the duct pressure was falling. The flow would have to drop below 44 CFM before the controller would open the damper to restore the 60 CFM setpoint.

When allowed to operate the dampers these controllers were able to achieve and stably hold the set point flows (at the flow the controller believed was correct) from the tested maximum flow rate of 300 cfm down to 20 cfm (~0.0005"). (Typical design flows for an 8" VAV box range from 400 to 800 CFM.) It is assumed that these controllers would control larger flows in a similar manner. Tests where the dead band was narrowed (by lowering the "cooling maximum" flow) show possible instability, though stability was always achieved in our tests.

Description of the Alerton Controller

The controller evaluated is a Model VAV-SD. This does not use a pressure sensor to measure the flow grid pressure but instead uses this differential pressure to produce a flow past an integral hot wire, or film, anemometer (henceforth referred to as a hot wire).

This was unknown to the installer and is not easily apparent from their specifications. Information from the spec sheet for the "VAV-SD" (found on the Alerton web site at www.alerton.com/Products/BACtalk/datasheets/LTBT-VAV-SD.pdf) says: *"Pressure sensor 0–1.25 inches water column differential pressure sensor."* While another web page (www.alerton.com/Products/BACtalk/Field_Controller_Level/VAV-SD.asp) describes it differently*; the "VAV-SD contains an integral airflow sensor*". No accuracy specification could be found. The controller operates a damper actuator model # LM24-3-T US. These were connected to model # BT1-100 communications box.

The associated software used was Envision for BACtalk. This version did not have a COV trending option. The maximum rate it could trend at was every ten seconds. The flow was recorded with a one cfm resolution. The control scheme incorporates a dead band where the damper is not adjusted if the flow is within 3% of the "max" to "min" range. Table E1 shows some ranges used and the associated dead band.

Table E1: Dead Band size at different settings.

Minimum [cfm]	Maximum [cfm]	Dead band [cfm]
0	1000	30
500	1000	15
40	140	3
70	1000	28
0	300	9

In practice, the range, and thus the dead band, is going to be fixed. The "max" and "min" settings affected the control parameters and the response to a step change in inlet static pressure or flow set point as will be shown below.

The flow zero was measured during installation and is assumed not to drift; it is not measured again by the software during normal operation. A calibration factor is assumed for the VAV box size and an "amplification factor" is used to make corrections for local conditions. The use of the term "amplification factor", used by the installer, for this value is confusing, as this is not the pressure amplification factor. The single (non-zero) flow calibration was done at a flow of about 150 cfm for these measurements, but some of the following results have been adjusted to the values that they would have been if this calibration had been made at 500 cfm. The differences between these calibrations are discussed below.

During installation the damper movement was timed from fully closed to fully open. This timing interval was used to control the damper and its position is determined based on the timed movements. The damper position was reset whenever the damper was asked to move to the fully close position, but there was no scheduled time when this would be done. (It is possible that a schedule would be made in a normal setup to fully close the damper.)

Accuracy

The Setra Micro-Cal could not be used to make requested pressures as it had been used in the Siemens analysis. This is because the Alerton controller allows flow from the upstream to the downstream side of the flow grid. To the Micro-Cal this appears as a leak, and it is unable to keep up with this "leak flow". Instead all measurements were made with actual flows. Because there is flow from one side of the flow grid to the other this pressure difference is less than if there were no flow. While this pressure difference drop is probably too small to measure, it was decided to only use the flows determined by the "duct blaster" for determining the reference flow values. These pressures are somewhat "noisier" than the pressure signals generated by the Micro-Cal, and required longer averaging times to obtain similar accuracy. Measurements of the accuracy of the controller were made in a similar manner as described in Appendix D.

The two point calibration of the Alerton sensors, at zero and one other flow, results in large flow discrepancies at other flows because of large nonlinear behavior. Figure E1 shows the error in the calibration if the single, non zero, calibration flow point is made at a high flow (450 to 600 cfm) for the two VAV boxes. The error shows a general under prediction of the flow at values less than the calibration point.

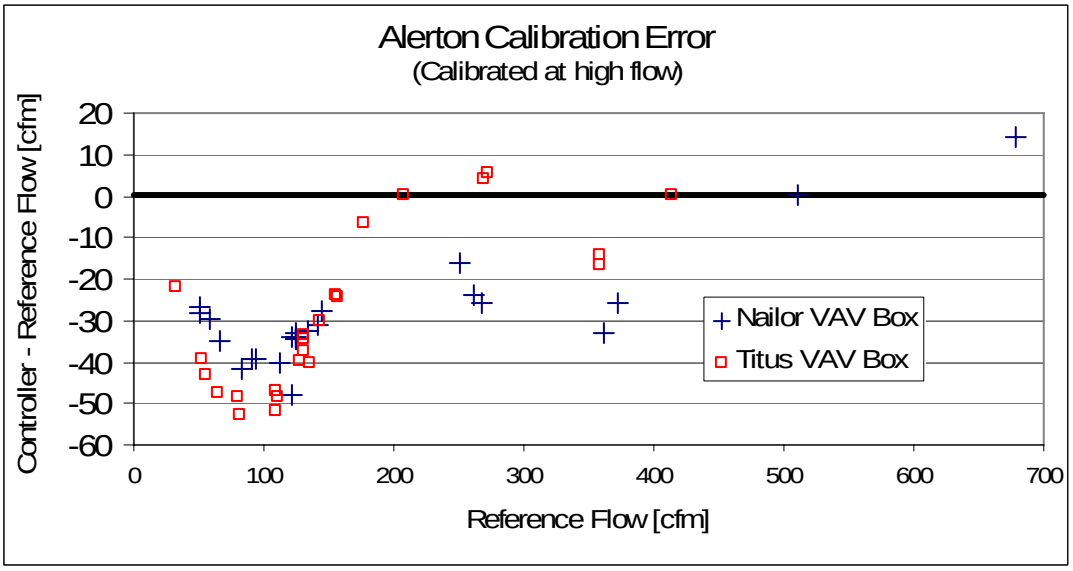


Figure E1: Calibration errors of the Alerton Controller for the Nailor and Titus VAV box flow grids if the calibration procedure uses high flows (about 500 cfm).

The above results are presented as if the single non-zero calibration flow point was at the high end of the expected range (400 or 500 cfm). The actual calibration points used were at about 150 cfm, as this was expected to result in better low end performance. Figure E2 shows the calibration errors if the 150 cfm value is used for the calibration. The maximum error below 150 cfm has been reduced by a factor of two, but the high flow errors are dramatically larger.

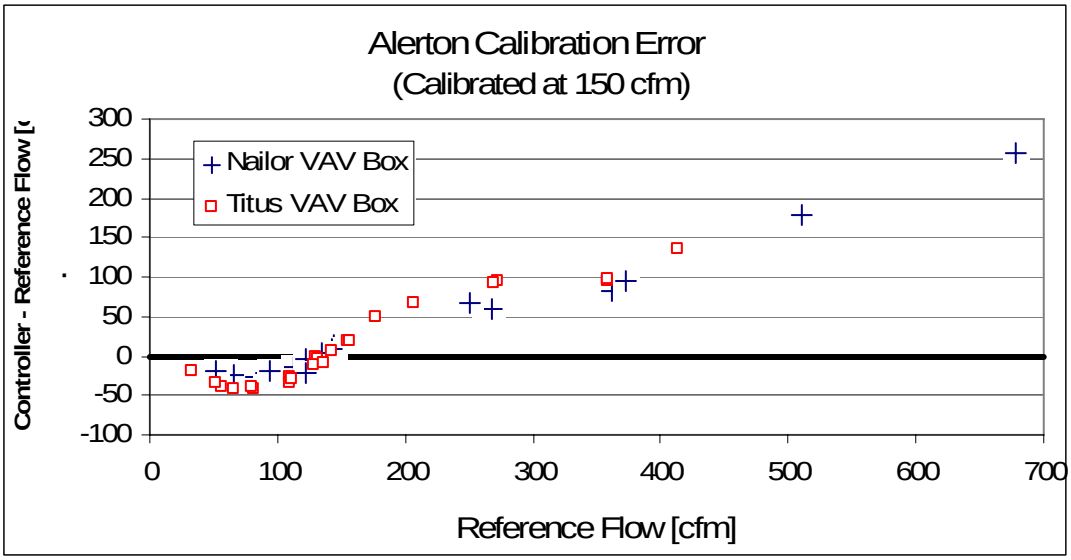


Figure E2: Calibration errors of the Alerton Controller for the Nailor and Titus VAV box flow grids if the calibration procedure uses low flows (about 150 cfm).

Resolution

Alerton says that the resolution of the hot wire is equivalent to about 16 bits. If the sensor was linear the resolution would be less than 0.1 cfm. We know that the sensor is not linear so the true resolution will vary with flow. The trend log recorded the flows as integers so it is not possible to determine the exact resolution but it was certainly better than 1 cfm.

Evidence of the resolution of the flow near 50 cfm can be seen at the end of this appendix in Figure E6

Stability

The response of the reported controller flow during periods of cycling flows was investigated at several flow ranges for several frequencies and amplitudes. This data was taken by varying the reference flow, whereas the corresponding data for the Siemens controller was taken with the Setra Micro-Cal. It was not possible to use the Micro-Cal because of the flow-through design of the Alerton flow sensor. Both methods result in similar information about the stability of the controller, but the Micro-Cal could only produce one cycle. Figure E3a shows an overview of these tests at about 125 cfm, while Figures E3b to E3e are close-up views of this data. Figure E3f shows a section of the test at 50 cfm.

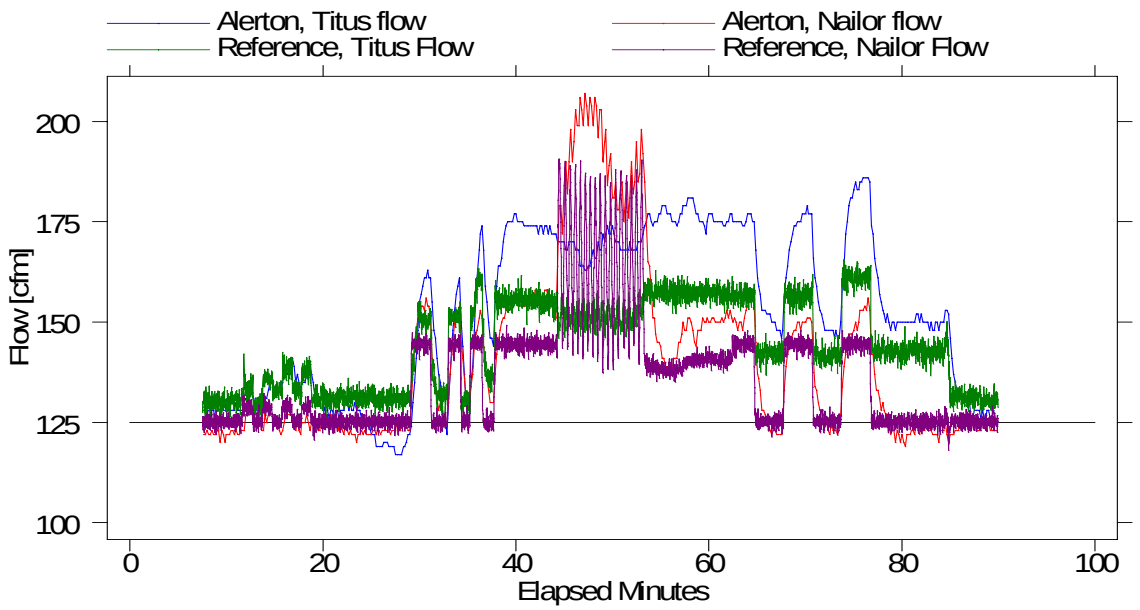


Figure E3a: Test of Alerton Controller response to cyclical changes of the reference flow. Four amplitude and frequency modes are show.

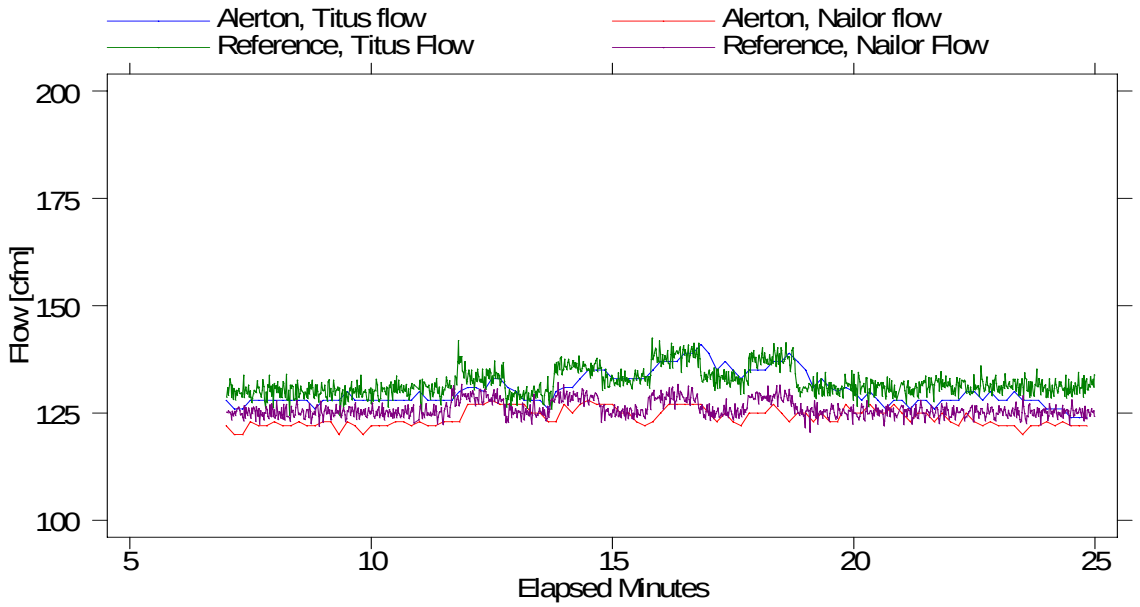


Figure E3b: Test of Alerton Controller response to cyclical changes of the reference flow.

At this frequency and amplitude the Alerton response appears to easily track the changes to the flow.

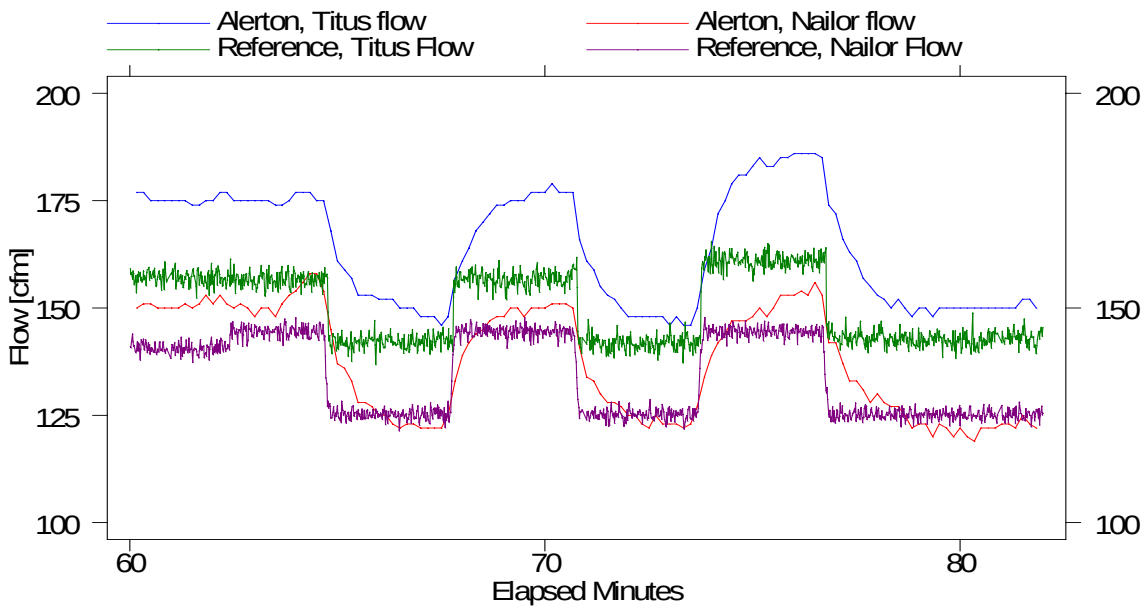


Figure E3c: Test of Alerton Controller response to cyclical changes of the reference flow.

This figure shows the lowest frequency of the four modes that were tested. The Alerton response appears to reach the new values at this frequency, about 1 cycle in 5 minutes.

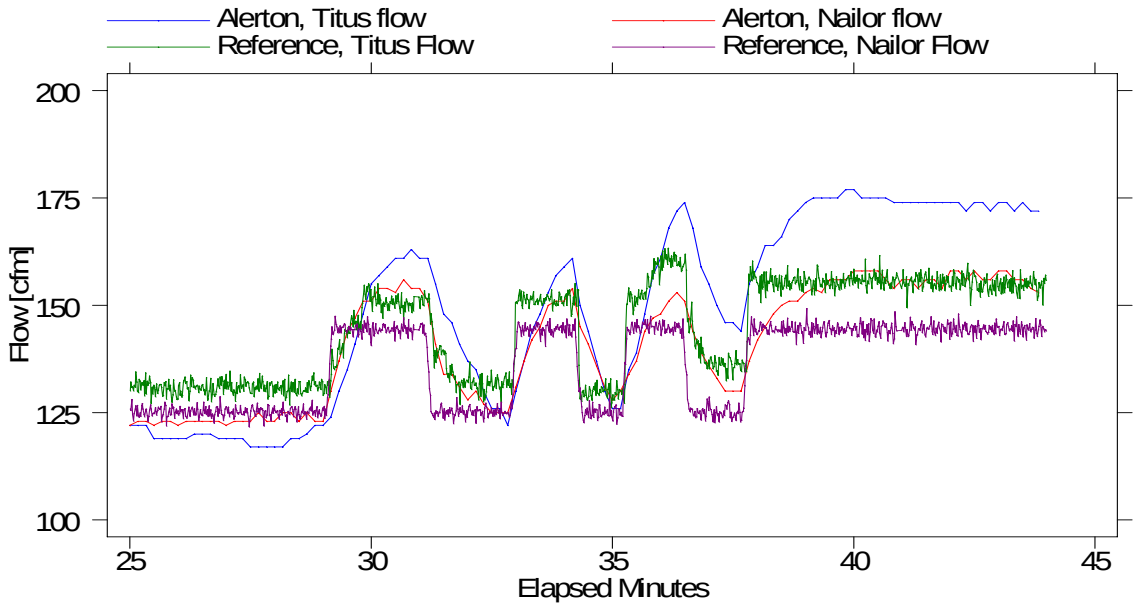


Figure E3d: Test of Alerton Controller response to cyclical changes of the reference flow.

At this frequency, about 1.5 cycles in 5 minutes, the Alerton response can not quite keep up with the changes to the flow.

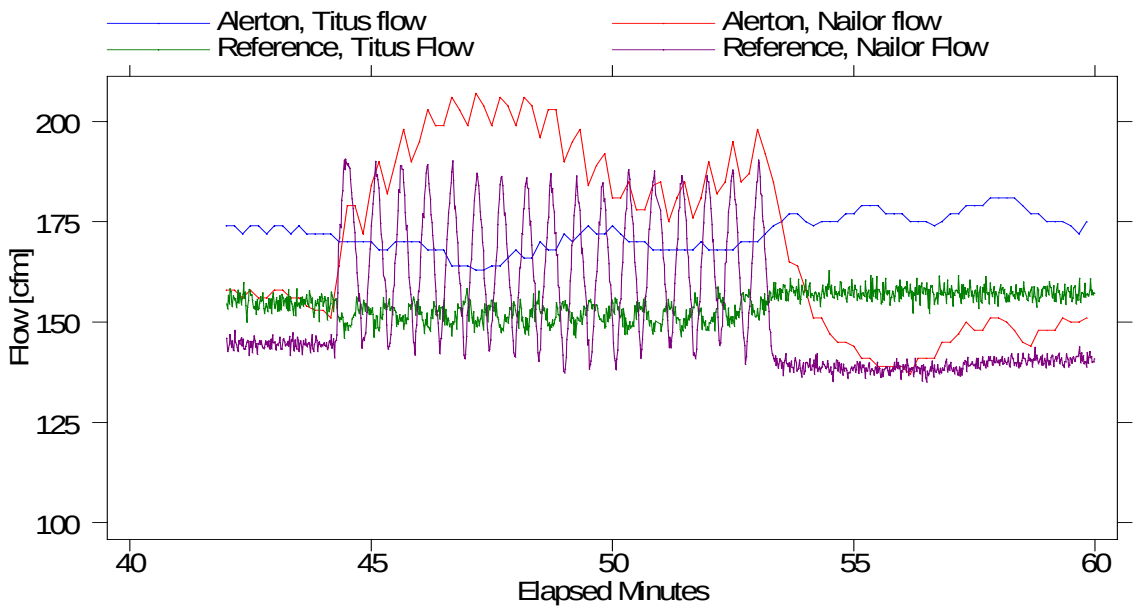


Figure E3e: Nailor flow cycled at almost 2 cycles per minute. (The Titus "duct blaster" fan speed was not adjusted but its flow varies due to changes in line voltage.)

It appears that the Alerton controller will over predict the flows during periods of rapid flow change. However at lower flows, shown in Figure E3f, a similar amplitude response was found but in this case there is no significant over prediction of the flow caused by the flow cycling.

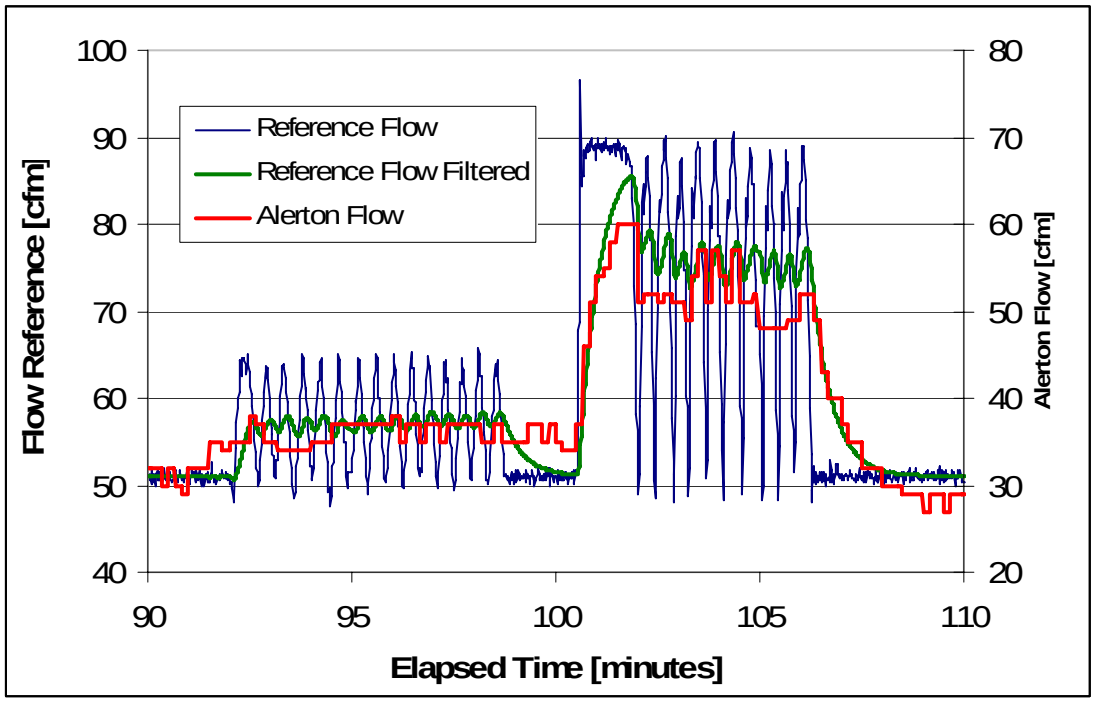


Figure E3f: Alerton controller response to cycling flows at about 50 cfm. The Alerton flow scale is on the right axis with a 20 cfm offset (using the "150 cfm" calibration).

A line showing a filtered value for the reference is included because it appears that the Alerton flow is over predicting the flow from elapsed times of about 102 to 107 minutes. The filtering helps to show that this is not the case, but rather, the flow fluctuation is not symmetric and the Alerton controller is successfully tracking the average flow.

Alerton Complete System Tests: Changing Inlet Static Pressure

The complete system tests are intended to show how the controller operates when the controller is allowed to move the VAV damper to adjust the flow to achieve the flow set point. Tests were made at several different duct inlet static pressures for each of several flow set points. Each static pressure was maintained for several minutes to allow the controller to stabilize the flow at the new static pressure. A second series of test were made where the flow set point was changed at a constant inlet static pressure.

Complete System Test at a flow set point of 300 cfm

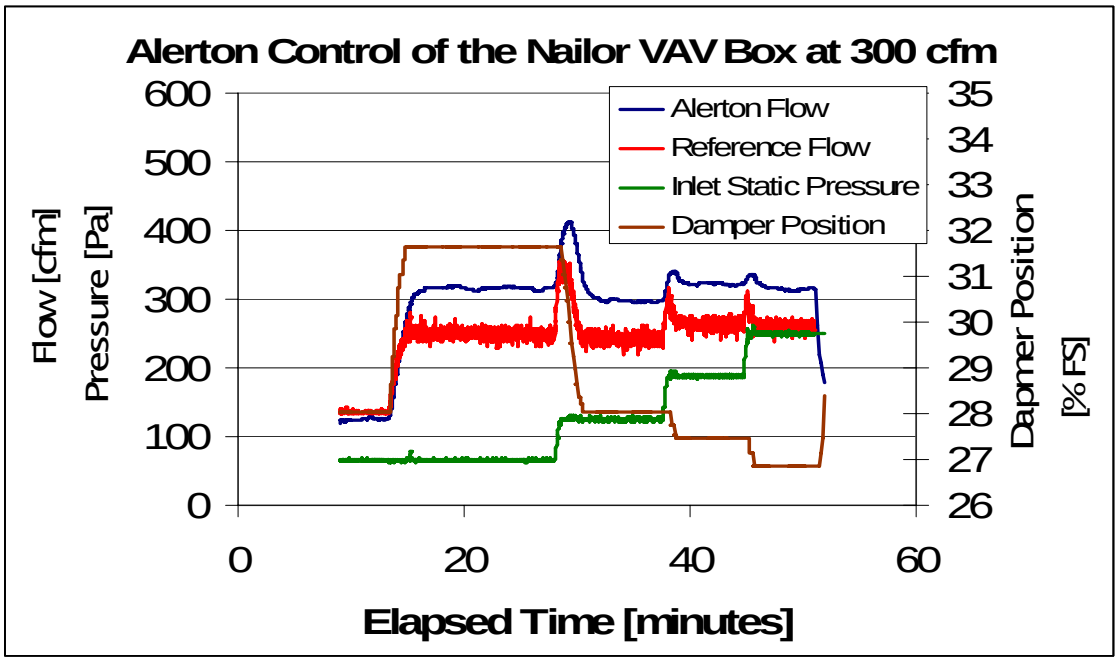


Figure E4a: Alerton controller response for the Nailor VAV Box to changes in inlet static pressure for a flow set point of 300 cfm.

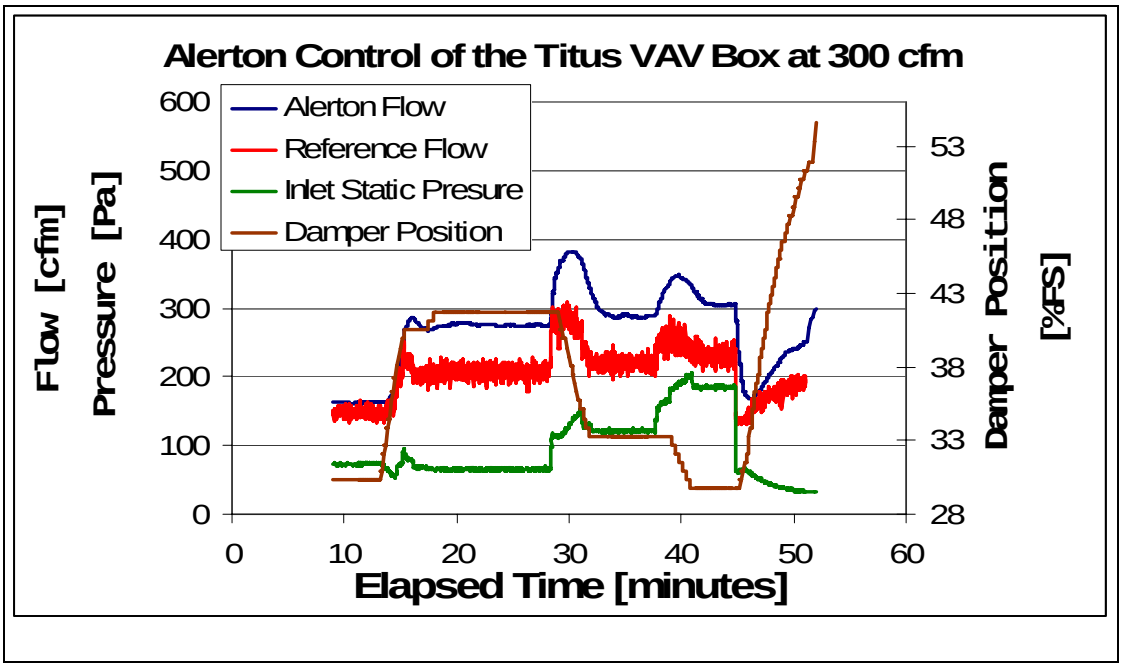


Figure E4b: Alerton controller response for the Titus VAV Box to changes in inlet static pressure for a flow set point of 300 cfm.

Complete System Test at a flow set point of 150 cfm

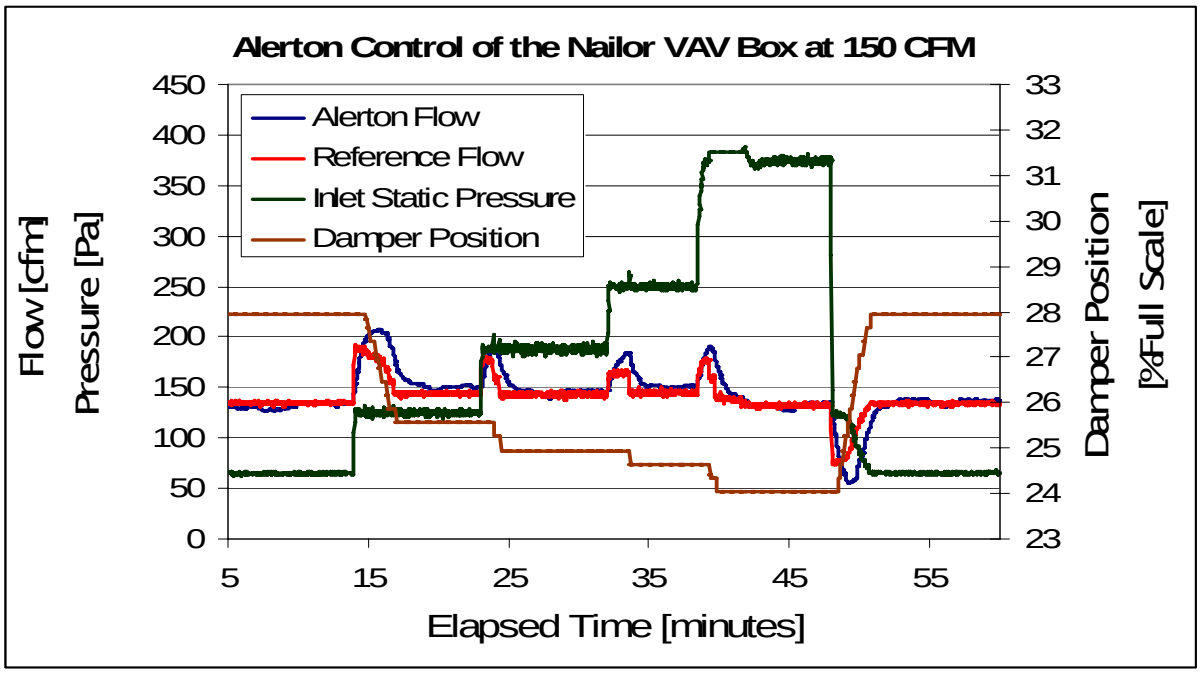


Figure E4c: Alerton controller response for the Nailor VAV Box to changes in inlet static pressure for a flow set point of 150 cfm.

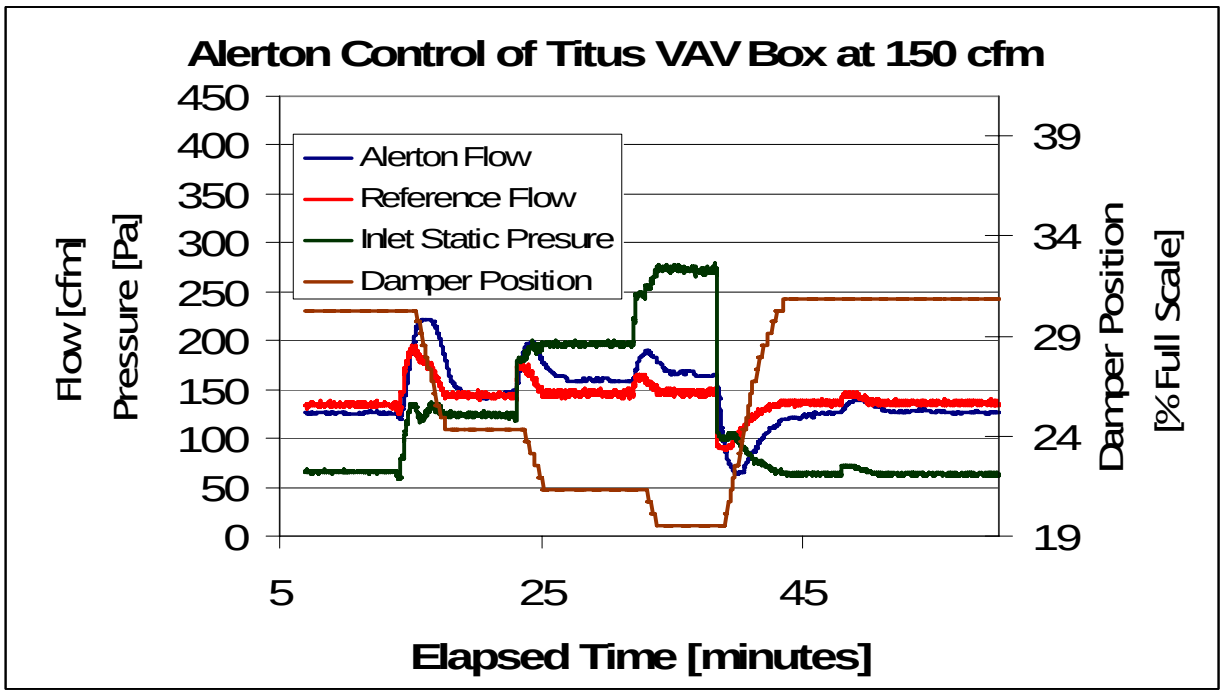


Figure E4d: Alerton controller response for the Titus VAV Box to changes in inlet static pressure for a flow set point of 150 cfm

Complete System Test at a flow set point of 70cfm

Figures E4e and E4f show the Alerton response to a step change in inlet static pressure at a set point flow of 70 cfm. The tests were made with two different max flow points, resulting in two different dead bands, initially 3 and then 30 cfm. The tight dead band results in some overshooting and subsequent ringing of the flow, which is then damped out. This behavior is more clearly seen in figures E4g and E4h.

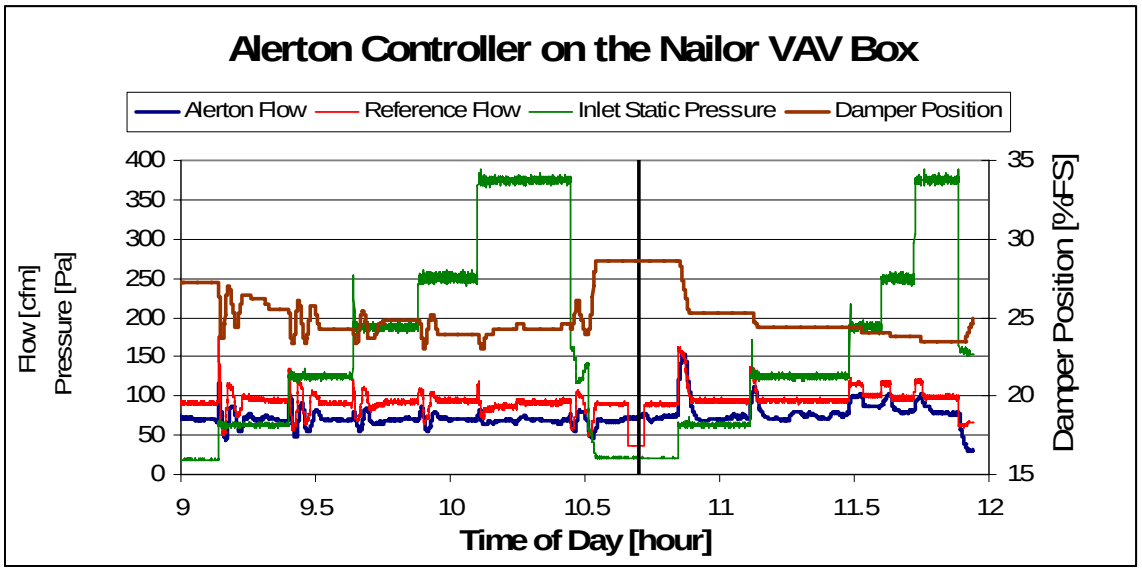


Figure E4e Controller response to inlet static pressure changes at a set point flow of 70 cfm for the Nailor VAV Box.

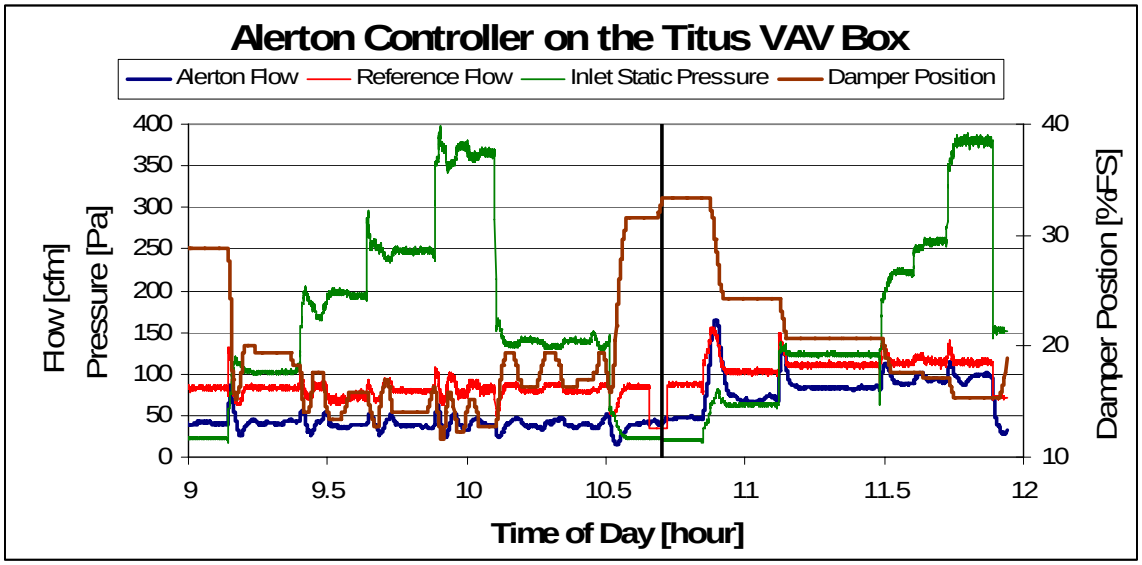


Figure E4f Controller response to inlet static pressure changes at a set point flow of 70 cfm for the Titus VAV Box.

In the test shown in figure E4g the max flow set to 100 cfm, resulting in a dead band of 3 cfm. Figure E4h shows the same test except with the maximum flow at 1000 cfm, resulting in a dead band of 30 cfm. The small dead band results in the controls overshooting and ringing, but this is damped out within ten minutes.

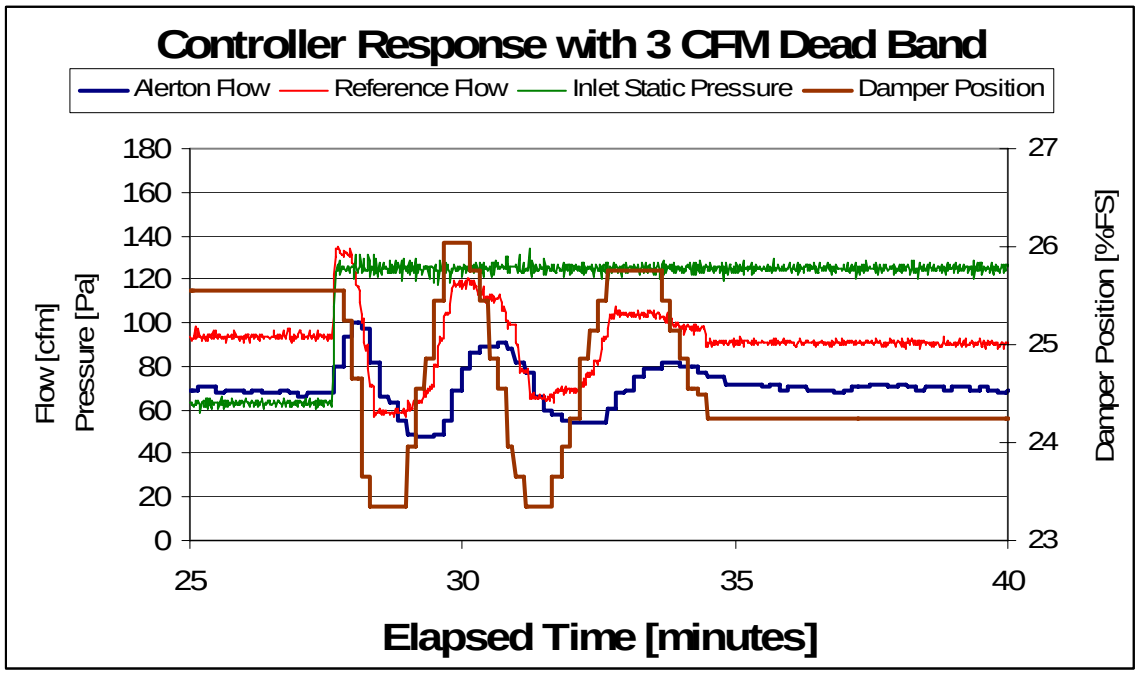


Figure E4g: Controller response to a step change in static pressure at a flow setpoint of 70 cfm. The flow max is 100 cfm with a resulting dead band of 3 cfm.

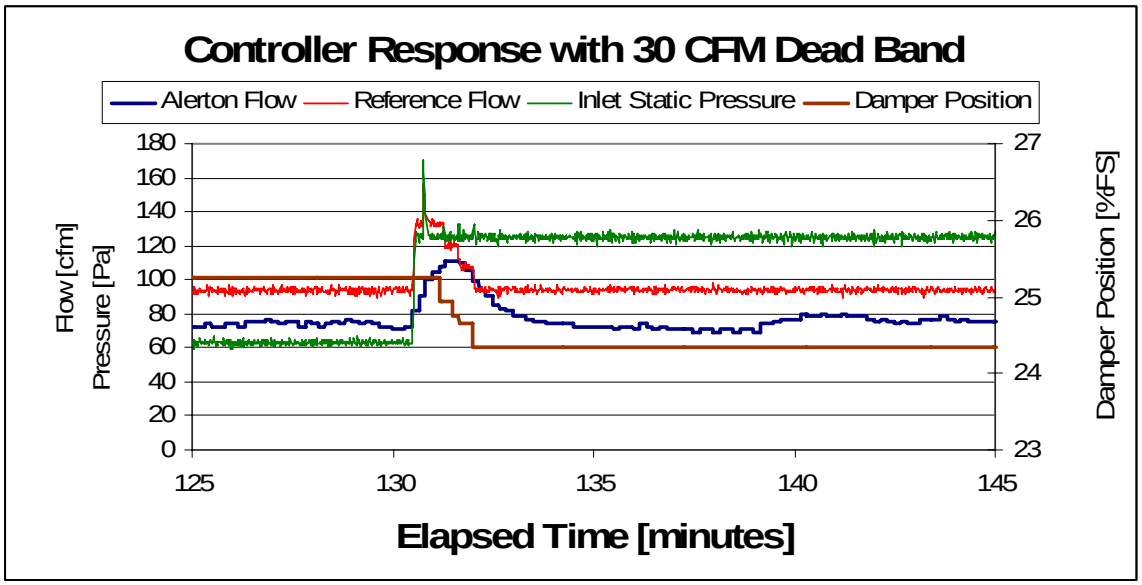


Figure E4h: Controller response to a step change in static pressure at a flow setpoint of 70 cfm. The flow max is 1000 cfm with a resulting dead band of 30 cfm.

At the end of this section damper movements for other flows and pressures and discussed (see tables E1 and E2).

Complete System Test of changes to the flow set point

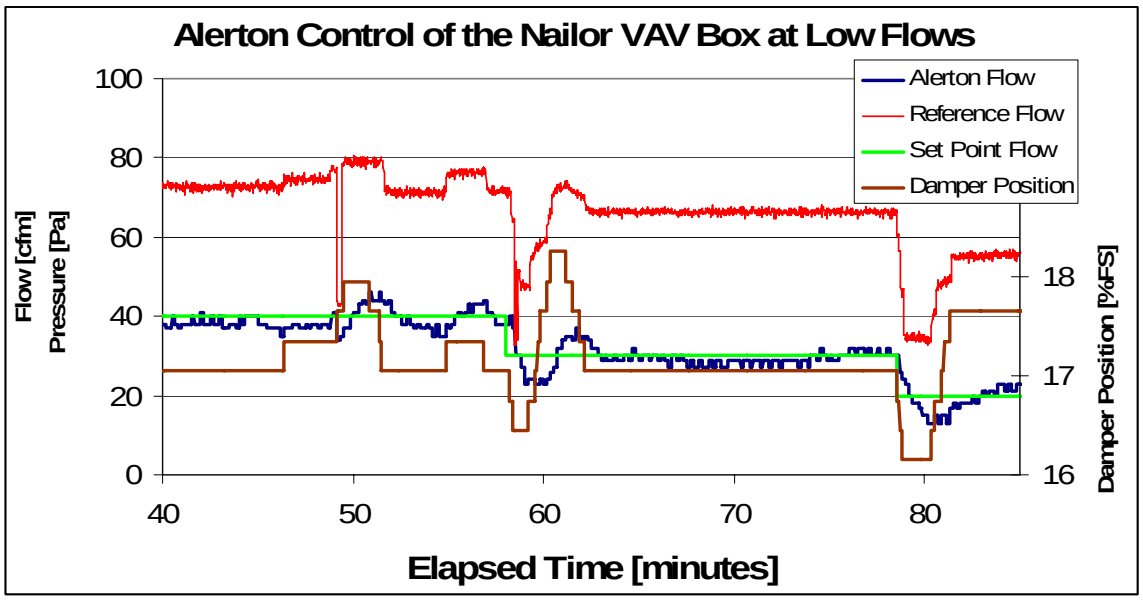


Figure E4i: Controller response to a change in the flow set point for the Nailor VAV Box at an inlet pressure of 1.0 iwc.

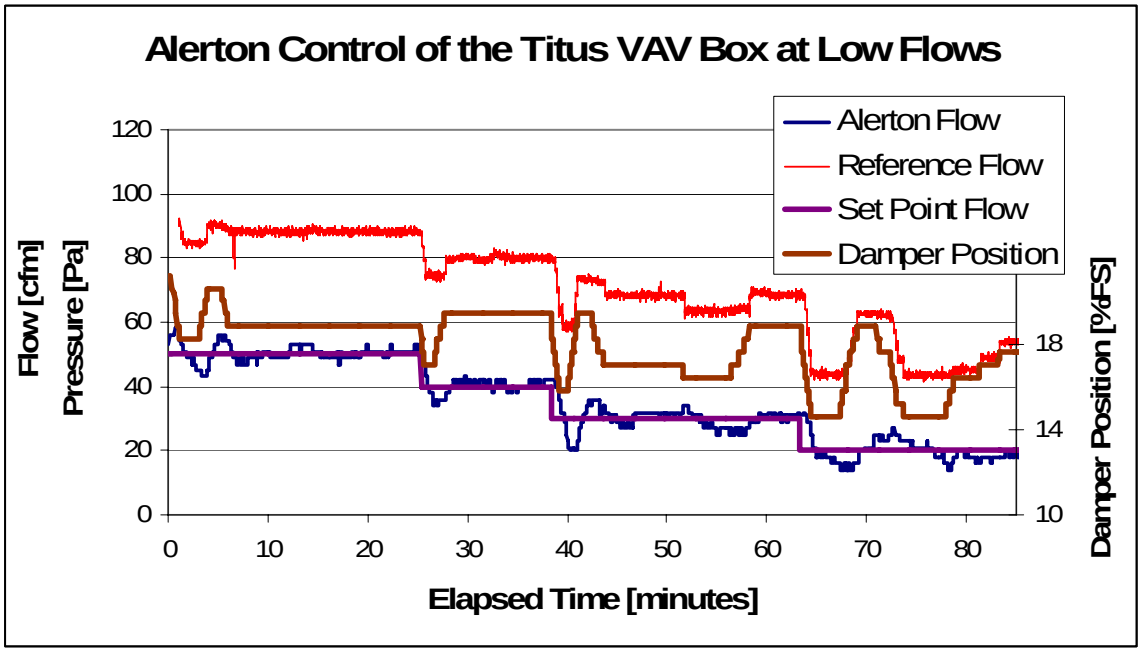


Figure E4j: Controller response to a change in the flow set point for the Titus VAV Box at an inlet pressure of 1.25 iwc.

The Alerton controller was able to achieve and stably hold the set point flows (at the flow the controller believed was correct) from the tested maximum flow rate of 300 cfm down to 20 cfm. It is assumed that these controllers would control larger flows in a similar manner. The normal flow dead band is 25 to 30 cfm and making this smaller requires one to lower the maximum controlled flow limit value. In practice this will limit how low a flow these controllers will successfully operate.

Tables E1 and E2 present a summary of the "complete system tests". The test sections selected for these tables represent times when the reference and Alerton reported flows appear stable. For tests with large flow dead bands, i.e. ±30 cfm, the dampers did not need adjustment, but the flows may settle far from a low flow set point. When the flow dead band is reduced to 3 cfm the data shows that the dampers are occasionally adjusted. It might be possible to make a combination of max, min and flow set points which would result in loosing control but this never occurred during these tests and would be an unlikely setting in normal operation.

Table E1: Alerton Controller Operating the Nailor VAV Box

		Duct Inlet Static pressure [iwc]					
Set Point Flow [cfm]		0.08	0.25	0.50	0.75	1.00	1.50
70 ±3	Alerton Flow [cfm]	72	70	70	71	71	69
	Reference Flow [cfm]	94	94	91	93	94	92
	Damper Position [degrees]	27.3	25.6	24.3	24.8	23.9	24.3
	# of Damper Adjustments	1	1	1	0	0	2
	Damper Adjustments/hour	7.5	15	15	0	0	12
	Minutes of Data	8	4	4	6	7	10
70 ±30	Alerton Flow [cfm]	73	73	75	87	80	77
	Reference Flow [cfm]	89	94	94	101	97	98
	Damper Position [degrees]	28.5	25.2	24.3	24	23.8	23.4
	# of Damper Adjustments	0	0	0	0	0	0
	Damper Adjustments/hour	0	0	0	0	0	0
	Minutes of Data	7	9	15	3	4	4
150 ±30	Alerton Flow [cfm]		132	150	146	153	134
	Reference Flow [cfm]		134	144	143	145	132
	Damper Position [degrees]		27.9	25.6	24.9	24.7	24
	# of Damper Adjustments		0	0	0	0	0
	Damper Adjustments/hour		0	0	0	0	0
	Minutes of Data		8	4	4	4	2
300 ±30	Alerton Flow [cfm]		316	297	322	315	
	Reference Flow [cfm]		250	242	263	260	
	Damper Position [degrees]		31.6	28	27.5	26.9	
	# of Damper Adjustments		0	0	0	0	
	Damper Adjustments/hour		0	0	0	0	
	Minutes of Data		11	3.5	2.5	1	

Table E2: Alerton Controller Operating the Titus VAV Box							
		Duct Inlet Static pressure [iwc]					
Set Point Flow [cfm]		< 0.15	0.25	0.50	0.75	1.00	1.50
40 ±3	Alerton Flow [cfm]			42	37	37	42
	Reference Flow [cfm]			83	71	80	83
	Damper Position [degrees]			19	15	14	13
	# of Damper Adjustments			3	4	0	4
	Damper Adjustments/hour			20	40	0	60
	Minutes of Data			9	6	6	4
70 ±30	Alerton Flow [cfm]	46	69	83	89	93	99
	Reference Flow [cfm]	87	103	111	13	115	114
	Damper Position [degrees]	25	24	21	18	17	15
	# of Damper Adjustments	0	0	0	0	1	0
	Damper Adjustments/hour	0	0	0	0	12	0
	Minutes of Data	7	7	15	3	5	4
150 ±30	Alerton Flow [cfm]		127	144	160	167	
	Reference Flow [cfm]		136	144	146	147	
	Damper Position [degrees]		30.9	24.3	21.3	19.5	
	# of Damper Adjustments		0	0	0	0	
	Damper Adjustments/hour		0	0	0	0	
	Minutes of Data		8	4	4	4	
300 ±30	Alerton Flow [cfm]	245	275	288	305		
	Reference Flow [cfm]	189	207	220	230		
	Damper Position [degrees]	49.7	41.6	33.3	29.7		
	# of Damper Adjustments		2	0	0		
	Damper Adjustments/hour		11	0	0		
	Minutes of Data		11	3.5	2.5		
The Alerton flows are listed for the single point calibration at 150 cfm							

It appears that the dampers need to adjust position more often for the Titus VAV Box than for the Nailor VAV Box, but this is probably an artifact of the testing procedure. In these measurements the inlet static pressure of the Nailor box was controlled by the "Duct Blaster" measurement software, while the control of the Titus box pressure was done manually.

Long Term Control and Pressure Sensor Zero Drift

To evaluate possible zero drift and stability isues an overnight test was performed at a set point flow of 70 cfm. The system HVAC fan was off and the Duct Blaster fans were used to provide flow. The inlet static pressure of the Nailor box was controlled to a constant, very low pressure of 0.02 iwc until the system HVAC fan came on, at about 15 elapsed hours. The inlet static pressure of the Titus box was not controlled and varied from 0.016 to 0.048 iwc.

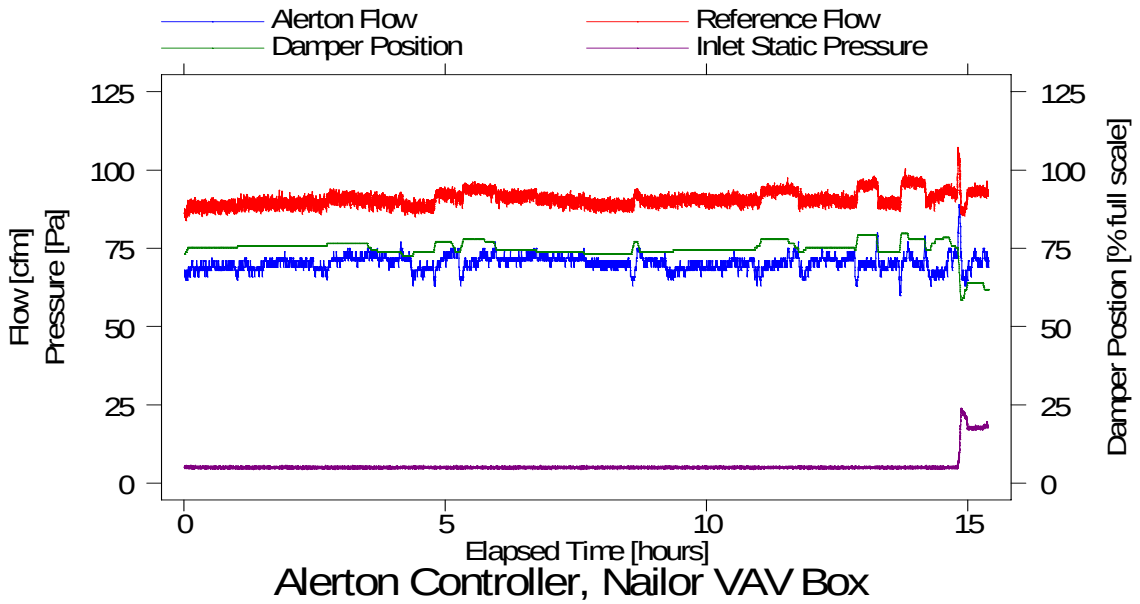


Figure E5a: Stability of the Nailor VAV Box flow for an overnight test.

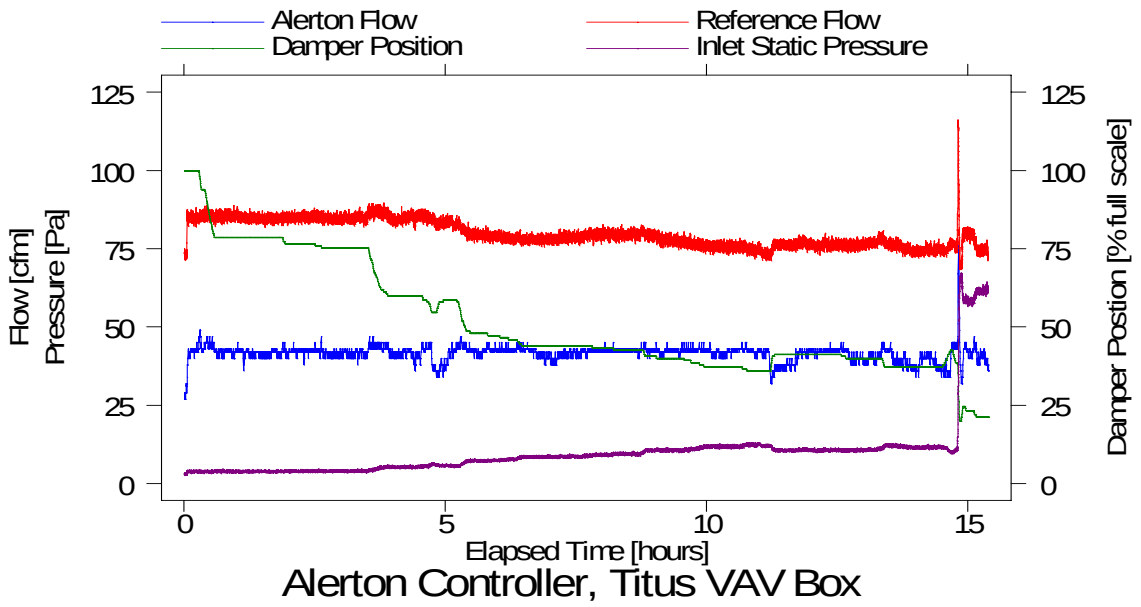


Figure E5b: Stability of the Titus VAV Box flow for an overnight test.

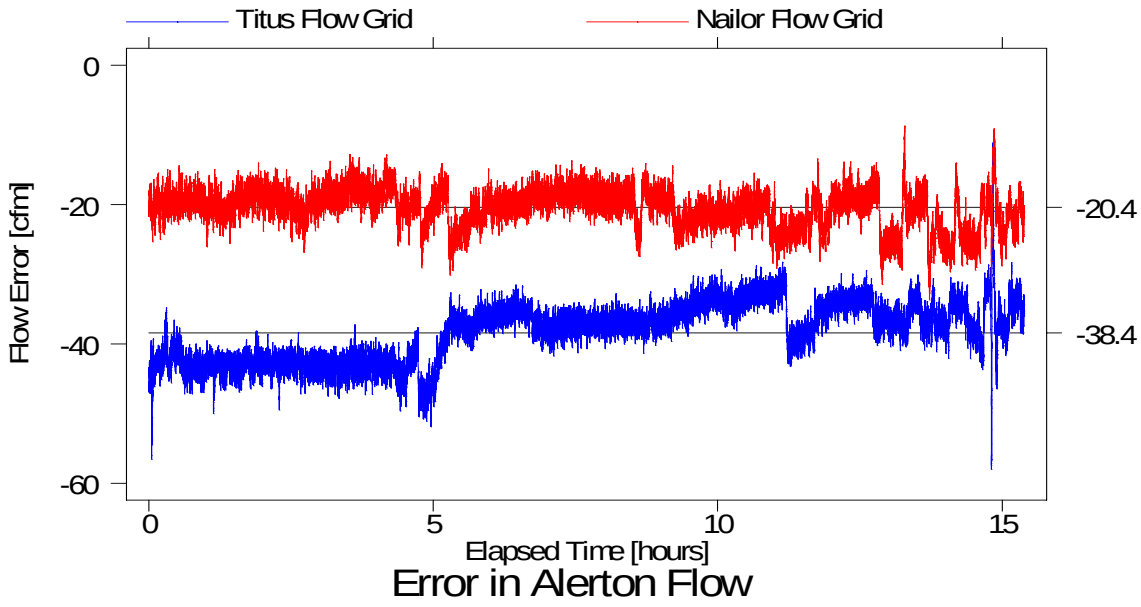


Figure E5c: Flow error and stability for an overnight test.

Figure E5c shows the difference between the flow reported by the Alerton controllers and the reference flow meters for the overnight test. The controllers maintained their set point flow within 5 to 10 cfm, showing no correlation with changes in temperature. The error in the flow (the difference between the controller value and the reference flow meter value) is -20.4 cfm for the Nailor VAV controller and -38.4 cfm for the Titus VAV controller. These "errors" are consistent with the errors found in the calibration as shown if figures E1. The spike just before 15 hours is due to an increase in the inlet static pressure when the HVAC fan turned on for normal operation.

Alerton Hot Wire Control

The signal sent to control the hot wire was also logged and an example can be seen in Figure E6 below. In this flow range the flow resolution is better than 0.1 cfm. A small phase shift between the hot wire control signal and the reported flow rate is also apparent. This indicates that, as expected, there is some averaging or filtering of the flow signal.

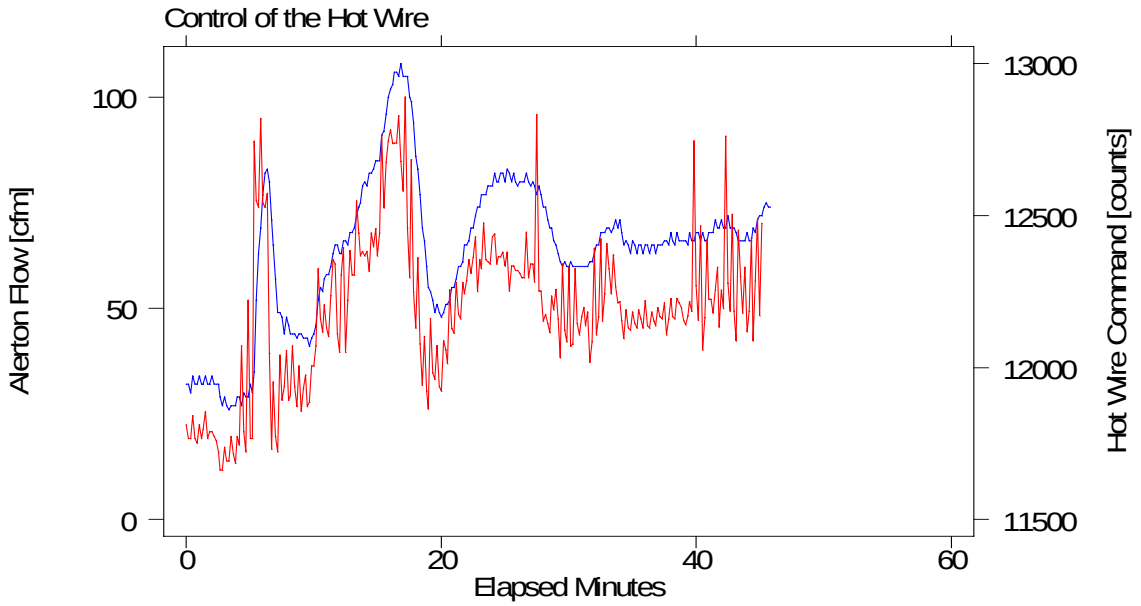


Figure E6: Example of the hot wire control signal at low flows.

Appendix F: Johnson Controls (JC)

Summary

The Johnson Controls controller uses a pressure sensor to measure the pressures generated by the flow grid in the VAV box. The two point calibration of this sensor, at 0 and 500 cfm, resulted in a calibration error of less than 5 cfm from the maximum flow tested, about 650 cfm, to 50 cfm (0.003"). The calibration errors below 50 cfm were larger, apparently due to a slight drift of the pressure zero. Unlike the Siemens controller, where zero drift appeared to be highly correlated to ambient temperature, no correlation between drift and temperature was observed over a 70 to 80 degree range of ambient temperatures. A small amount of sensor drift was observed (~0.001"), which is smaller than the Siemens drift (~0.003"). The source of the Johnson sensor drift is unclear. By default the pressure zero is measured once every two weeks. At the conclusion of the other tests, measurements at low flow were taken for a 30 day interval during which both sensors zeroed twice. On both occasions both sensors did not correctly zero. After performing the "autozero" sequence one sensor reported a flow of 0 when the reference flowmeter measured 50 cfm (during operating hours). The other sensor, after performing the "autozero", reported a flow of about 75 cfm when the reference flowmeter indicates about 50 cfm during operating hours, and reported a flow of about 50 cfm during periods of no flow. It is not clear what caused the error in the zeros. The optional "BO" autozero solenoid was not evaluated. The error caused by this autozero process appears to be in the range of 0.003". At an offset error of 0.003" the minimum flow setpoint would have to be at least ??? to achieve a accuracy within 20% of reading.

The measurements of resolution and stability show that the correctly zeroed controller can accurately track the reference flow in all evaluated conditions at or above 50 cfm. The flow calculated from the pressure signal is significantly damped, so that in tests with rapid cycling of the flow grid pressure, the average is correctly reported.

When allowed to operate the dampers these controllers were able to achieve and stably hold the set point flows from the tested maximum flow rate of about 630 cfm down to 25 cfm at all input static pressures. These controllers also use a flow deadband to prevent excessive damper adjustments. The size of the deadband is dynamic, and responds to the amount of noise in the pressure signal. In these tests this deadband was smaller that the one used by Alerton and would not appear to significantly impact using these controllers at low flow settings.

Description of the Johnson Controls Controller

The controller evaluated is a Metasys AP-VMA1410-0. This has an internal pressure sensor with a maximum pressure rating of 1.5 inches of water. Information about the controller can be obtained from "VMA 1400 Series Variable Air Volume Modular

Assembly" at http://cgproducts.johnsoncontrols.com/CAT_PDF/1928275.pdf. Another useful document is: "Variable Air Volume Modular Assembly (VMA) 1400 Series Application Note", at http://cgproducts.johnsoncontrols.com/MET_PDF/6375125.pdf, LIT-6375125, in which equipment options and control algorithms are described.

No specific information about the accuracy, resolution or stability of the pressure sensor is given in this literature. The controller has an integral damper actuator and its position is measured with an internal sensor. No information about its resolution is given.

The controller was connected to a model NAE 55 communications box. Flow and damper set points and trending intervals were adjusted with a web browser based interface. This was not a full feature implementation of Metasys and it did not create a data base of trended values. The "Metasys Export Utility" was used to down load the data from the controllers into a file at scheduled times. Because of limited memory in these units the down load was necessary every 15 minutes when recording data at a one second interval. It was necessary to make 60 schedules to capture both short term and overnight data. The flows were recorded with a resolution of 1 cfm, and the damper positions were recorded with a resolution of 0.1 % of full scale. The controller resolution of the Nailor VAV box was inadvertently set to 1% of full scale so the damper position sensor described in Appendix A, measured by the "APT" data logging equipment, was used for these measurements.

The flow zero was measured during installation and, according to the installing Johnson Controls technician, it is not measured again by the software during normal operation. In fact, by default, the zero is measured every two weeks. It is possible that the technician did not anticipate that the testing period would extend beyond two weeks and meant that it would not rezero during the anticipated testing period. As investigated at the end of this appendix, neither of the two units studied correctly rezeroed. Starting with version 8.05 of "HVAC Pro" an optional "BO" zeroing solenoid is offered in the literature but was unfamiliar to the installer and was not evaluated.

In-situ calibration of the flow was made during the initial set-up at flow zero and at about 500 cfm. A multi point calibration was not available. As with all the controllers studied, the zero flow point is forced to be one of the calibration points. Zero flow is accomplished by disconnecting the tubing and shorting them, thus insuring a true zero, it is the most accurately determined flow point used.

Johnson Controls also uses a deadband to prevent excessive damper adjustments. The size of the deadband is continually adjusted as described in "Variable Air Volume Modular Assembly (VMA) 1400 Series Application Note":

> "The P-Adaptive flow control algorithm uses a patented fixed gain, proportional control loop with a self-adjusting deadband whose value is related to an estimate of the noise variance. The P-Adaptive control strategy is used in the secondary flow control loop for pressure independent applications. P-Adaptive control has the advantage of much tighter flow control without oscillation, because it dynamically adjusts the flow deadband, based on the turbulence (noise) measured on the pressure sensor. P-Adaptive does not require any tuning."

In the measurements presented here the deadband was about 15cfm, smaller than the one used by Alerton (3% of range, normally 30 cfm), but in situations with higher noise it might be larger.

Accuracy:

Two setups were used to measure the accuracy of the controllers and to evaluate their resolution and stability. Initially a system of closed tubes with a manually operated piston or clamp was used as was done for some of the Siemens tests. Other measurements were made by controlling the "Duct Blaster" fan to produce real flows as was necessary in the Alerton tests. In general the system of closed tubes produces a low noise signal but one with significant drift whereas the "Duct Blaster" methods are easier to control but are somewhat noiser and require slightly longer measurement times to obtain the same degree of accuracy. In practice, use of the "Duct Blaster" was easier and easily offset the extra data collection time.

Figure F1 shows the results of the accuracy measurements. The difference between the flow as reported by Johnson Controls and the reference flow meter were generally less than 5 cfm for flows from 50 to 630 cfm. The accuracy of lower flows decreased as small pressure zero offsets begin to dominate.

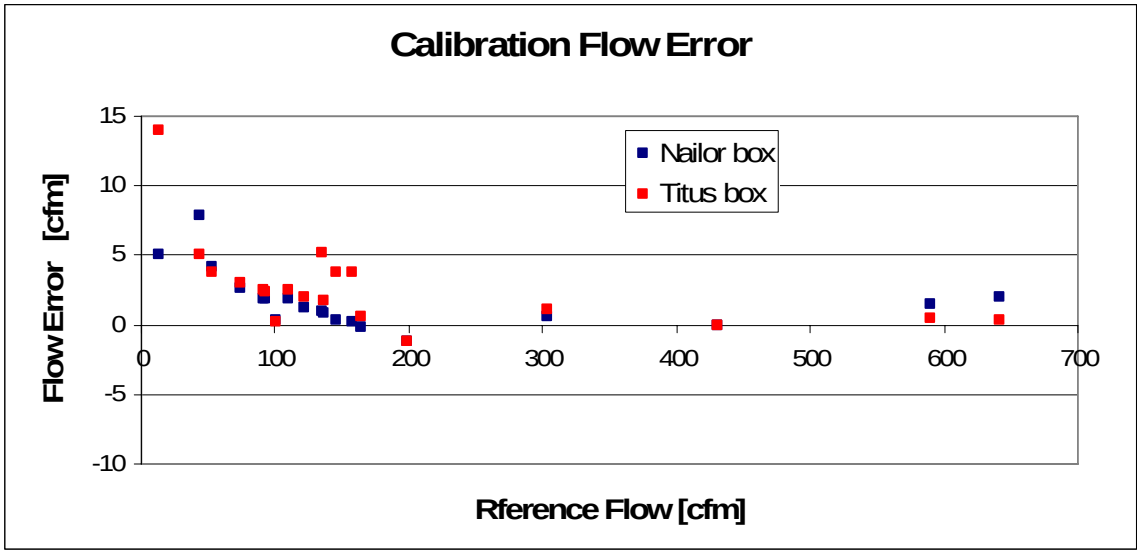


Figure F1: Errors in the calibration of the JC controllers.

The largest error is less than 15 cfm and is almost certainly a result of the drift in the value of the pressure signal zero (a drift of 0.001 iwc would account for the trend seen in the Nailor VAV box).

Resolution & Stability

The response of the reported controller flow during periods of cycling flows was investigated at several flow ranges for several frequencies and amplitudes. Tests were made on both VAV boxes with nearly identical results. Most of the following figures show data for one VAV box only. Figure F2a to F2c illustrate data taken with the system of closed tubes configuration, while the data shown in Figure F2d used the "Duct Blaster" fan to generate real flows.

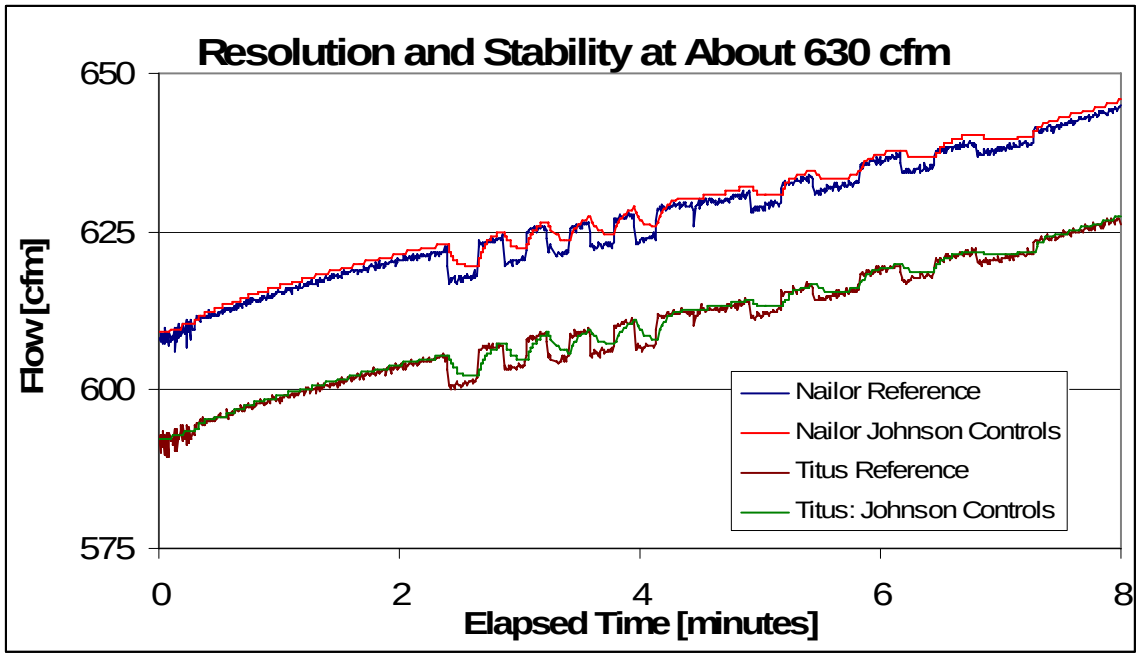


Figure F2a: Controller response to small changes of the flow pressure input at an equivalent flow of about 630 cfm.

The overall increase in pressure (flow) seen in Figure F2a is due to a slight increase in temperature of the closed system of tubes calibration setup during the test. The controller reported flows exhibit a dampened response as evidenced by the rounded response to a square wave change in the flow pressure.

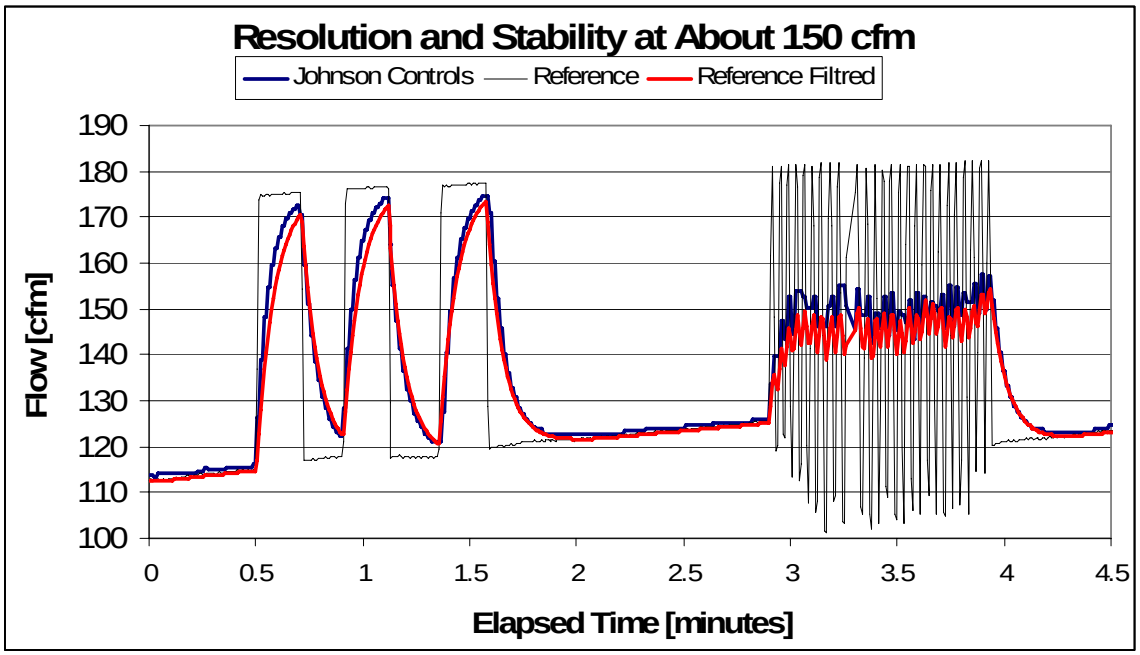


Figure F2b: Controller response to slow then rapid cycling of the flow pressure input at an equivalent flow of about 150 cfm for the Nailor VAV Box.

In figure F2b a filtered value of the reference signal has been calculated to demonstrate that the controller reported flow is accurately tracking the average value of the pressure (flow) signal during periods of rapid change. During the initial two minutes the average of the reference and filtered flow was 140 cfm and the controller reported 141 cfm. During the rapid cycling at the end of the test the reference and filtered values are 142 cfm and the controller reported flow is 146 cfm.

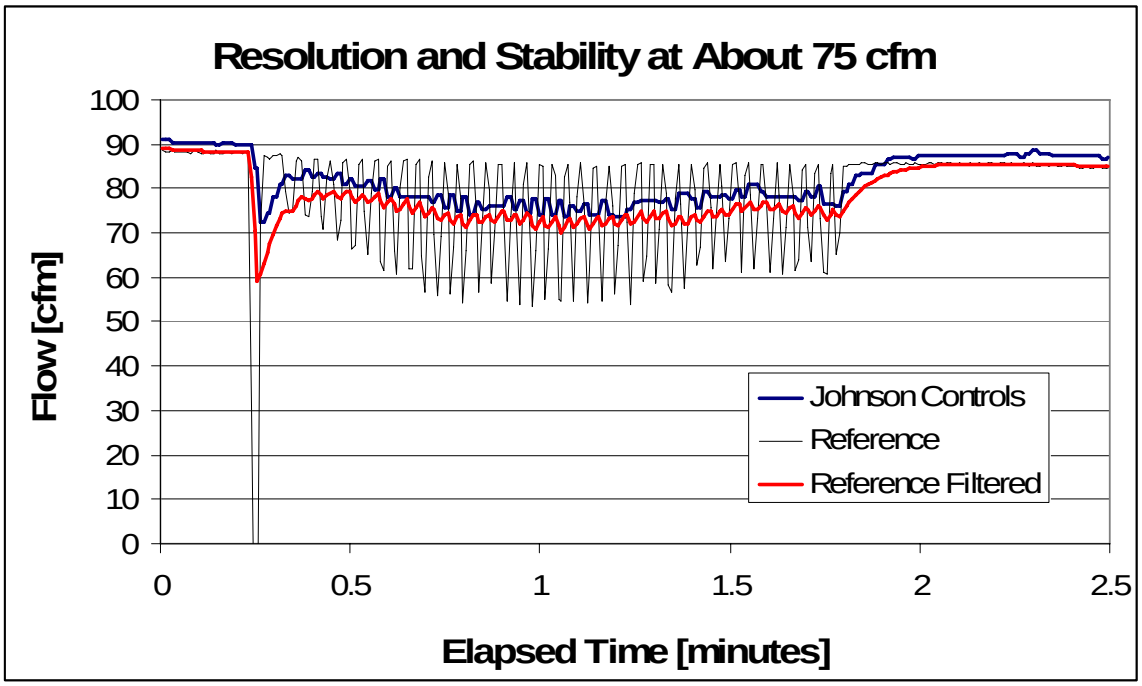


Figure F2c: Controller response to rapid cycling of the flow pressure input at an equivalent flow of about 75 cfm for the Nailor VAV Box.

Figure F2c shows data similar to that of Figure F2b except at lower pressures (flows). The initial flow was 80 cfm as determined by the reference flow meter while the Johnson Controls reported value was 86 cfm. The calibration data suggest the difference would be about 3 cfm. During the period of rapid cycling of the pressures (flows) the average reference flow was 75 cfm and the Johnson Controls value was 78 cfm. Again, this difference is what is expected based on the calibration data, indicating that the flow signal is correctly averaged by the Johnson Controls controller.

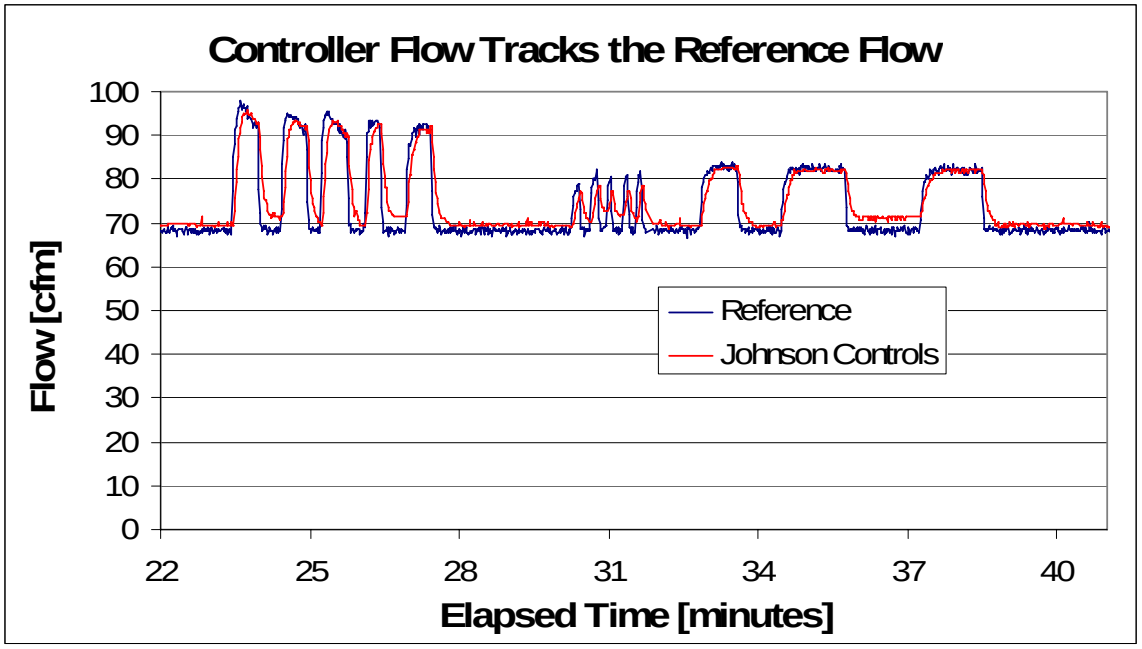


Figure F2d: Controller response to cycling of the flow at about 75 cfm for the Nailor VAV Box using the "Duct Blaster" calibration setup.

Data taken with the "Duct Blaster" calibration set up is illustrated by Figure F2d. In this mode the pressure on the flow grid is produced by real flows. The flow produced by the "Duct Blaster" is controlled by the "Teclog" software in one of several modes: (1) the desired flow can be typed into a control window and the software then adjusts the fan speed to achieve this flow; (2) use the arrow keys to adjust the flow rates up or down as desired or (3) purposely set the Teclog control parameters to unstable values and request a flow as in the first case, this results in fast cycling as the flow first overshoots then undershoots the target value.

Complete System Tests:

The complete system tests are intended to show how the controller operates when the controller is allowed to move the VAV damper to adjust the flow to achieve the flow set point. Tests were made at several different duct inlet static pressures for each of several flow set points. Figures F3a,b, and c illustrate the data for the low flow set points of 100, 50 and 25 cfm. Each static pressure was maintained for several minutes to allow the controller to stabilize the flow at the new static pressure. After stabilization the data was

analyzed to check for flow accuracy and the number of damper position adjustments during periods of stable inlet static pressure. Tables F1 and F2 summarize these results. The flow usually stabilized within 10 cfm of the flow set point. After the initial adjustment to a new inlet pressure or flow set point, the damper position was adjusted in only one instance, the Nailor VAV Box at 600 cfm,

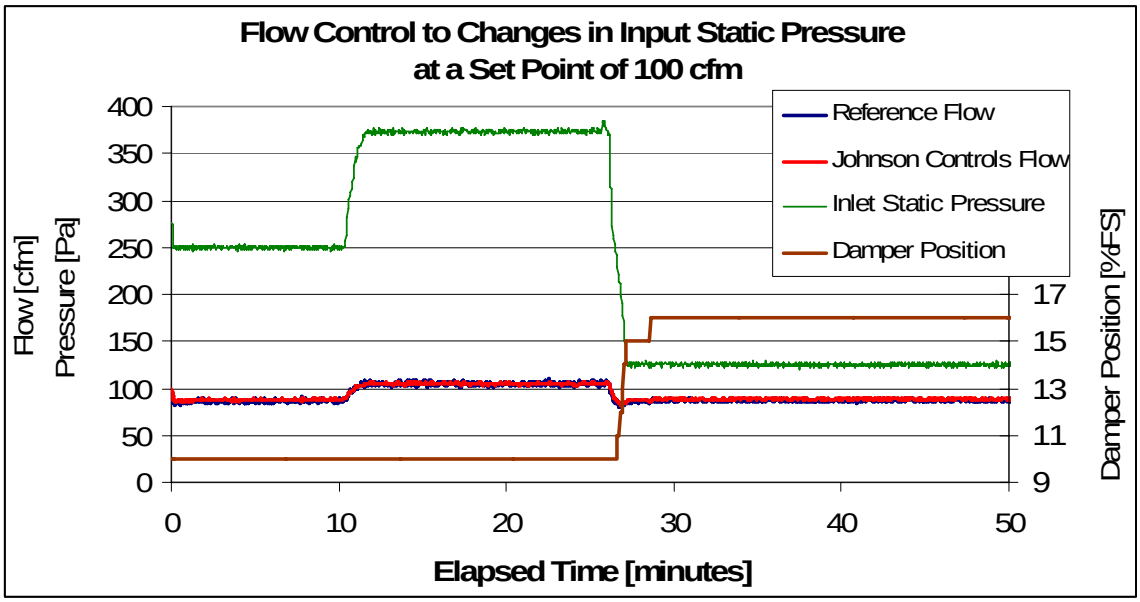


Figure F3a: Controller response to changes in input static pressure at a flow set point of 100 cfm.

Evidence of the size of the flow dead band can be seen in Figure F3a when the input static pressure increased from 1.0 iwc (250 Pa) to 1.5 iwc (375 Pa) and the damper position was not changed. Dropping the inlet static pressure to 0.5 iwc (125 Pa) lowered the flow enough to trigger a response of opening the damper until the flow again was in the dead band. It appears that the dead band is about ±15 cfm (0.0003") under these conditions.

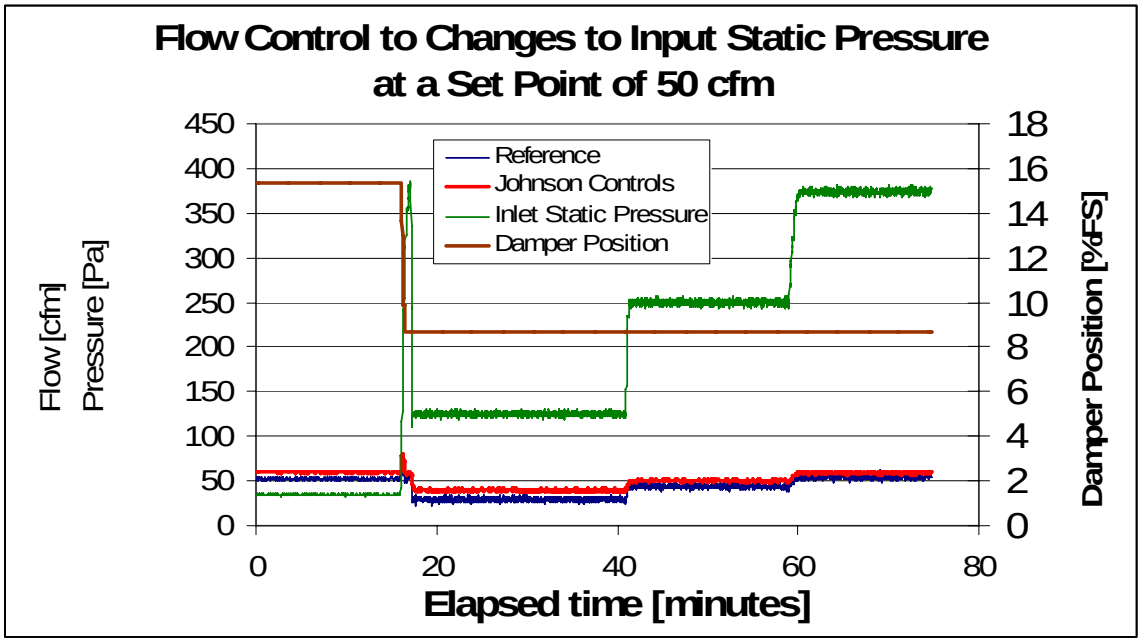


Figure F3b: Controller response to changes in input static pressure at a flow set point of 50 cfm.

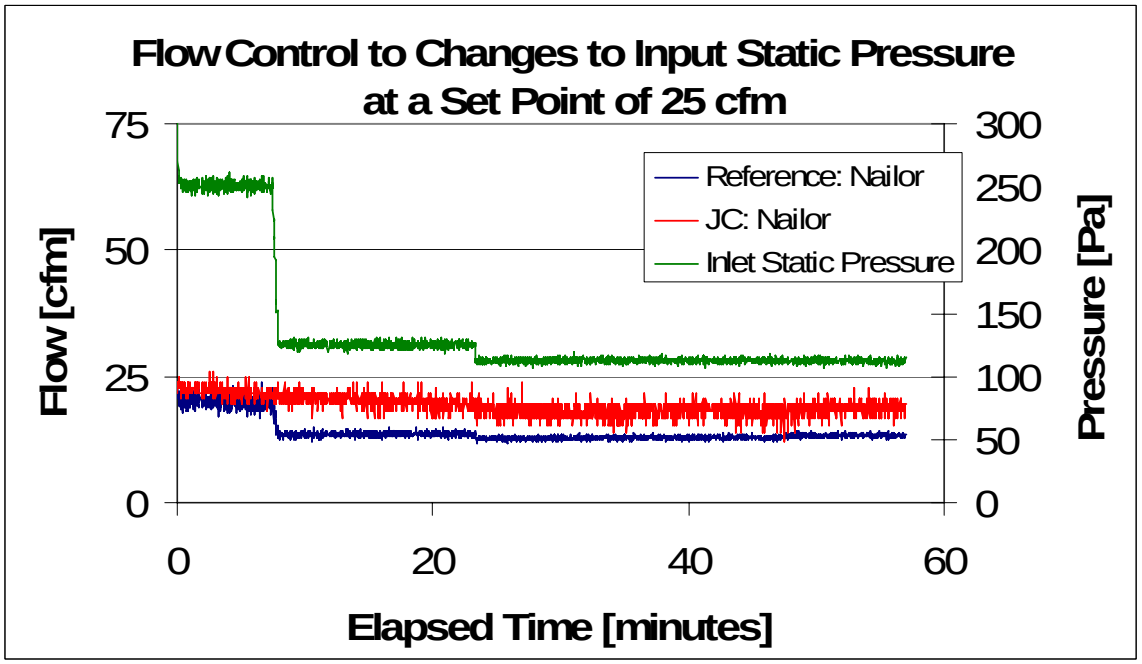


Figure F3c: Controller response to changes in input static pressure at a flow set point of 25 cfm. The damper position was never adjusted by the controller, staying at 5.5% of full scale.

The controller flow does not accurately track step changes in the reference flow at these low flow conditions. It is possible that the controller uses a filter with a long time constant when the "noise" to signal ratio increases.

Table F1: Johnson Controls Controller Operating the Nailor VAV Box					
Set Point Flow	Units: Flows in cfm, Damper Position in % Full Scale	Duct Inlet Static pressure [iwc]			
		0.1 to 0.3	0.4 to 0.7	1	1.5
25	Johnson Controls Flow		20	22	
	Reference Flow		14	20	
	Damper Position		1.2	1.3	
	# of Damper Adjustments		0	0	
	Damper Adjustments/hour		0	0	
	Minutes of Data		12.2	6.6	
50	Johnson Controls Flow	60	40	50	59
	Reference Flow	57	35	50	62
	Damper Position	13.5	5.1	5.1	5.1
	# of Damper Adjustments	0	0	0	0
	Damper Adjustments/hour	0	0	0	0
	Minutes of Data	15.6	21.9	16.2	13.7
75	Johnson Controls Flow	82	75	77	
	Reference Flow	82	74	77	
	Damper Position	14.9	10	7.3	
	# of Damper Adjustments	0	0	0	
	Damper Adjustments/hour	0	0	0	
	Minutes of Data	8.7	10.8	8.7	
100	Johnson Controls Flow		88	87	106
	Reference Flow		87	86	105
	Damper Position		10.6	7.8	7.8
	# of Damper Adjustments		0	0	0
	Damper Adjustments/hour		0	0	0
	Minutes of Data		24.6	8.4	12
150	Johnson Controls Flow	157	146		
	Reference Flow	157	146		
	Damper Position	36.9	19.5		
	# of Damper Adjustments	0	0		
	Damper Adjustments/hour	0	0		
	Minutes of Data	55.2	6.2		
300	Johnson Controls Flow		303		
	Reference Flow		303		
	Damper Position		36.2		
	# of Damper Adjustments		0		
	Damper Adjustments/hour		0		
	Minutes of Data		6		
450	Johnson Controls Flow		430		
	Reference Flow		431		
	Damper Position		48.5		
	# of Damper Adjustments		0		
	Damper Adjustments/hour		0		
	Minutes of Data		7.2		
600	Johnson Controls Flow		589		
	Reference Flow		588		
	Damper Position		51		
	# of Damper Adjustments		2		
	Damper Adjustments/hour		17		

	Minutes of Data		7.2		

Table F2: Johnson Controls Controller Operating the Titus VAV Box					
Set Point Flow	Units: Flows in cfm, Damper Position in % Full Scale	Duct Inlet Static Pressure [iwc]			
		0.1 to 0.3	0.4 to 0.7	1	1.5
25	Johnson Controls Flow		29	31	
	Reference Flow		15	25	
	Damper Position		17.2	17.2	
	# of Damper Adjustments		0	0	
	Damper Adjustments/hour		0	0	
	Minutes of Data		32.7	6.6	
50	Johnson Controls Flow	60	57	59	57
	Reference Flow	54	52	56	56
	Damper Position	27.7	22.2	20.6	19.6
	# of Damper Adjustments	0	0	0	0
	Damper Adjustments/hour	0	0	0	0
	Minutes of Data	15.6	21.9	16.2	13.7
75	Johnson Controls Flow	84	82	82	
	Reference Flow	76	75	75	
	Damper Position	31.7	26.9	24.1	
	# of Damper Adjustments	0	0	0	
	Damper Adjustments/hour	0	0	0	
	Minutes of Data	8.7	10.8	8.7	
100	Johnson Controls Flow		95	106	105
	Reference Flow		90	101	98
	Damper Position		29.3	25.7	24
	# of Damper Adjustments		0	0	0
	Damper Adjustments/hour		0	0	0
	Minutes of Data		24.6	8.4	12
150	Johnson Controls Flow	159	159		
	Reference Flow	155	155		
	Damper Position	56.5	37.8		
	# of Damper Adjustments	0	0		
	Damper Adjustments/hour	0	0		
	Minutes of Data	55.2	6.2		
300	Johnson Controls Flow		294		
	Reference Flow		292		
	Damper Position		54.6		
	# of Damper Adjustments		0		
	Damper Adjustments/hour		0		
	Minutes of Data		6		
450	Johnson Controls Flow		450		
	Reference Flow		450		
	Damper Position		64.8		
	# of Damper Adjustments		0		
	Damper Adjustments/hour		0		
	Minutes of Data		7.2		

600	Johnson Controls Flow		585		
	Reference Flow		594		
	Damper Position		70.1		
	# of Damper Adjustments		0		
	Damper Adjustments/hour		0		
	Minutes of Data		9		

Pressure Sensor Zero Drift

Long term measurements were made to assess the magnitude of the zero drift of the pressure sensors. Both VAV boxes had set point flows of 1000 cfm to force the dampers to fully open. The reference flow meter was set up in the low flow range and the manual dampers at the building's supply duct were adjusted so that the flow into each VAV was about 50 cfm when the buildings system was on.

Figures F4a and F4c show the data for the entire 29 days of the test for the two VAV boxes. After the initial four days the building's static pressure, and thus flows, increase during operating hours. In the seventh day (and 21st day) the controller performs a zeroing cycle in which the VAV damper is closed and the pressure on the flow grid is assumed to go to zero. In both instances, for both controllers, this procedure did not work. After the zeroing procedure, the flow reported by the Nailor VAV box is too high while the flow reported by the Titus VAV box is zero at all times. It is clear that the zeroing procedure did not work correctly but the cause of this error is unknown.

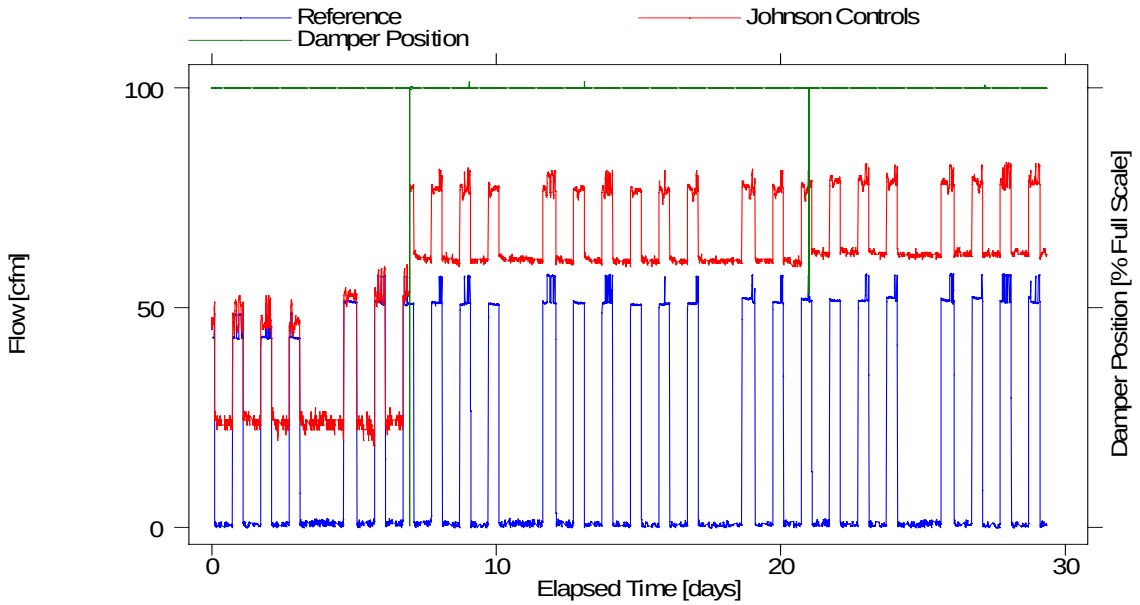


Figure F4a: Long term measurements of the Nailor VAV box with inlet pressures controlled by the buildings HVAC system.

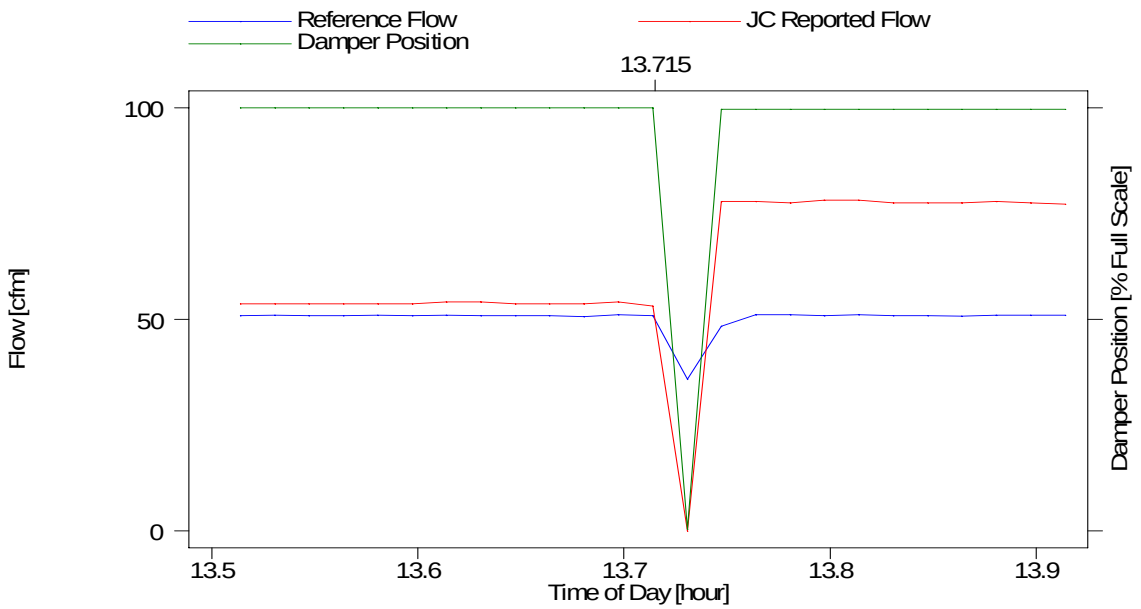


Figure F4b: Close up of the time around the zeroing procedure for the Nailor VAV box.

Figure F4b shows a close up of the time around the first zeroing procedure for the Nailor VAV box. The reference data was recorded in one minute averages and because the zeroing procedure takes less than one minute it does not show the flow going to zero but does indicate that the average flow decreases as expected. The same is true for the damper position (not shown on the plot).

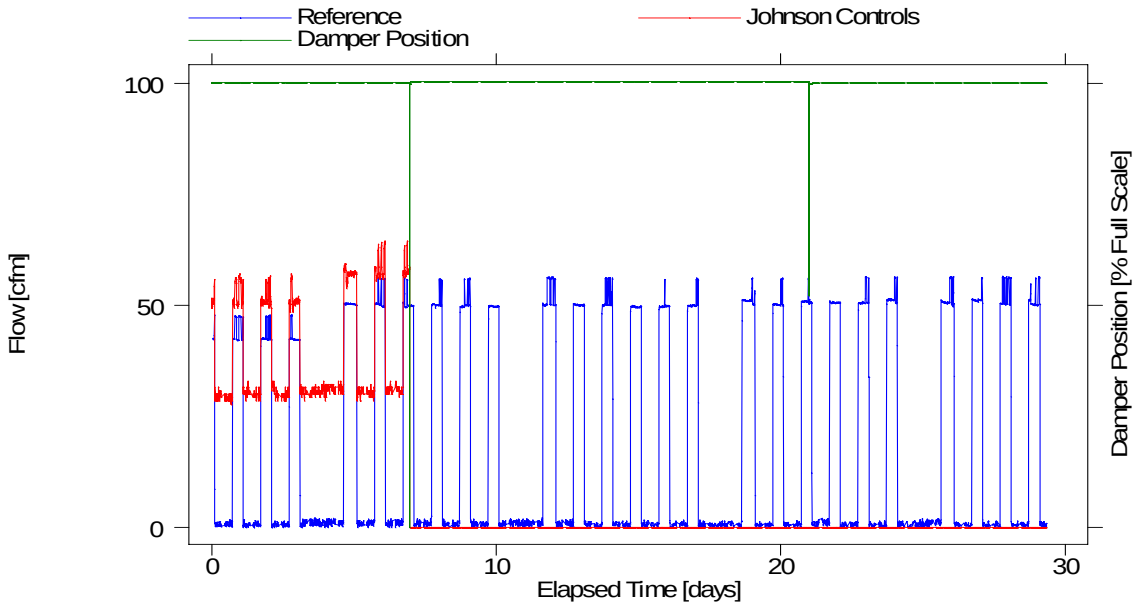


Figure F4c: Long term measurements of the Titus VAV box with inlet pressures controlled by the buildings HVAC system.

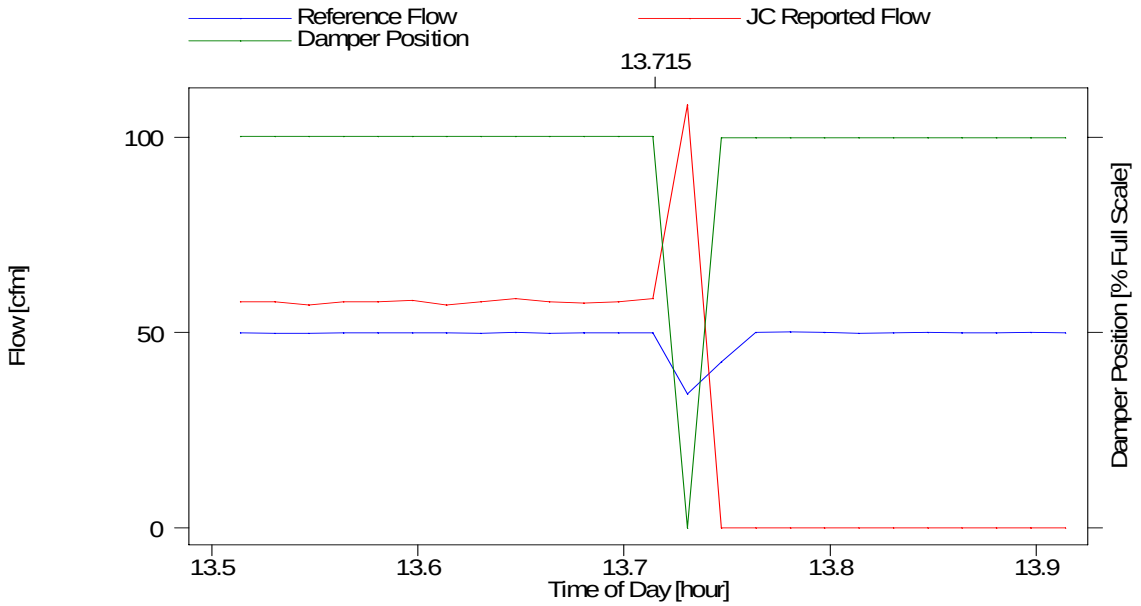


Figure F4b: Close up of the time around the zeroing procedure for the Titus VAV box

The data for the Titus VAV box during the zeroing procedure shows a positive spike in the controller reported flow while the reference flow drops as expected. The meaning of the controller reported flow during the zeroing period is unknown, however the equivalent flow spike from the Nailor VAV box data goes to zero, as if that is the new corrected value. The software does not allow negative flows and sets them to zero, so the value from the Nailor VAV box may be from a negative, "zero corrected", pressure.

Figure F5 shows the calculated value of the pressure sensor zero versus temperature for the non-zero data. The two bands of values for the Nailor VAV box, before and after the auto-zero, are evidence that the zero is misread or misapplied. There is only one band of values for the Titus VAV box because the data after the auto-zero is all equal to zero, and so might come from any "negative" reading. This would violate the assumptions made in making these calculations. There is no correlation to temperature as had been found with the Siemens controllers.

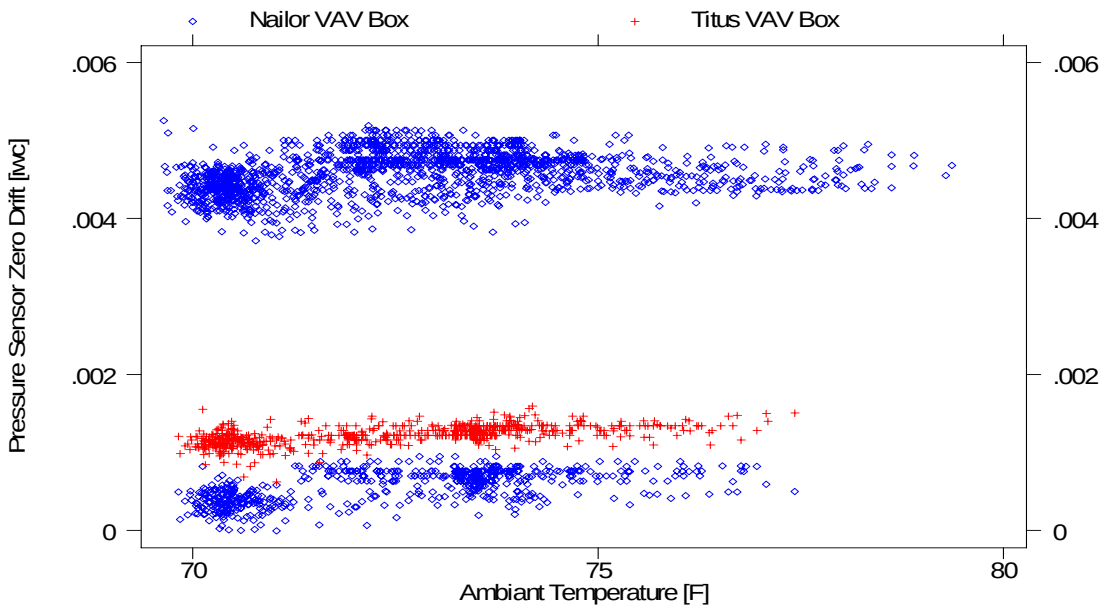


Figure F5: Zero drift of the Johnson Controls controller shows little dependence on temperature.

Appendix G: Automated Logic Corporation (ALC)

Summary

The ALC controller, like the Alerton, uses the pressures generated by the flow grid in the VAV box to induce a small flow across a hot wire type sensor in their controller. This air speed is then appropriately scaled to determine the flow rate of the VAV box. The specifications are given as an equivalent pressure (i.e. 2.00 inches of water full scale).

The difference between the reported flow and the reference flow show that these sensors are not linear at low flows. Because of this nonlinearity, the calibration procedure can play a significant role in determining the overall accuracy. The normal in-situ calibration consists of flows at four points which were 0, about 75, 300 and 600 cfm. The 75 cfm point was selected to represent a possible minimum flow setpoint which is lower than is usually specified. Although more points were used in the ALC calibration procedure than in the Alerton calibration the result is similar in that it under-reports the flow between the low flow calibration point and zero flow, but it only did so by about half the amount. Thus these controllers will always err on the side of supplying a little more than the desired minimum flow. Accurate control of a desired minimum flow setpoint can be achieved if this setpoint is one of the calibration points.

The zero drift seen during a two and a half day period was less than the measurement noise of about 1 cfm. The flow control dead band is about ±5 cfm for flows less than 200 cfm, which ought to be adequate for most applications. The reported flow tracked the reference flow for all changes, big and small, fast and slow when in manual override mode, but appears to have an imposed 10 second “filter” on the flow signal when used in “normal operation” as a controller.

The damper position was not changed when the input static pressure was constant for all flow set points at all input static pressures, i.e. no hunting or instability was observed at any flow setpoint.

Description of the Automated Logic Corporation Controller

The Automated Logic controller tested is a model ZN341V+/zn141V+. This does not use a pressure sensor to measure the flow grid pressure but instead uses this differential pressure to produce a flow past an integral hot wire, or film, anemometer (henceforth referred to as a hot wire). This controller has an integral damper actuator. The two controllers used communicated back to a host PC using a “multi-equipment controllers and routers” model ME-LGR25. Links to literature on both these devices are easily found at:
http://www.automatedlogic.com/alcinternet.nsf/webview/products_control_modules.

No specific information about the accuracy or stability of the hot wire or damper position is given in this literature. The air flow sensor has specifications as if it were a pressure

sensor; listing a range of "*0-2" W.C., sensitive down to ±0.001" W.C.*" This would put the "sensitivity" of the hot wire at about 25 cfm for the 8" VAV boxes studied. In fact the resolution is much better than this and it is unclear what is meant by "sensitive".

The associated software used was WebCTRL. This is a web based tool that is very graphically orientated. The flow and damper position were trended every second; however when the controller was allowed to control the flow (the complete system tests) the trended values were constant for ten second intervals. A continually updated plot of selected data can be made but it lags by about 5 minutes. The temperature and flow set point were trended every minute.

The ALC controller has a flow dead band that corresponds to one second of damper movement. Thus it is dependent on damper position and the inlet static pressure. The dead band is evaluated after every damper movement. In these tests the dead band appears to be about 5 cfm for flows under 200 cfm and up to ±10 cfm for higher flows. The deadband is adjusted at the "minimum occupied flow" point to prevent flows lower than this minimum.

The flow zero was measured during installation and is assumed not to drift; it is not measured again by the software during normal operation. Three other flow points are used in the in-situ calibration. These were made at 77, 292, and 613 cfm for the Nailor VAV box; and 74, 284, and 590 cfm for the Titus VAV box.

Movement of the damper was self timed between mechanical stops during installation. The literature did not indicate if the damper position was determined from mechanical feedback or from software tracking of the timed damper movements.

Accuracy

As with the Alerton controller all these measurements were made using the "Duct Blaster" reference flow meter. Figure G1 shows the results of the accuracy tests. The calibration error is < 3% of flow for flows above 200 cfm, but can get quite large for lower flows.

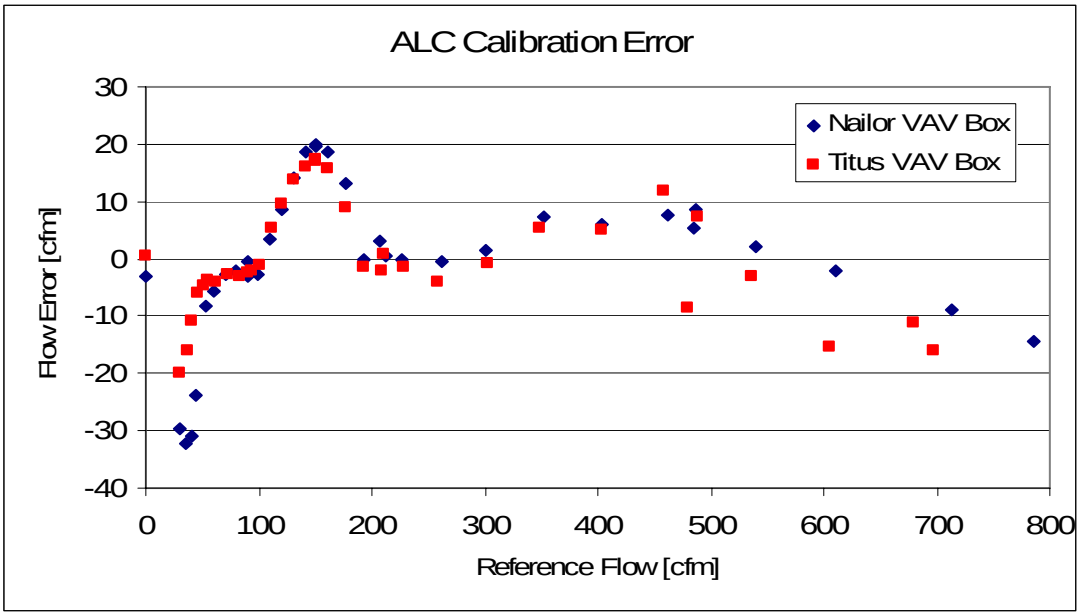


Figure G1: Calibration error of the ALC controller for both VAV box flow grids.

The calibration error approaches zero at the flows where the in-situ calibration points were made (0, and about 75, 300 and 600 cfm), and vary in a smooth curve between these values. The response of these "hot wire" sensors is known to be complex and require multiple calibration points to achieve their ultimate accuracy.[1]

Resolution & Stability

The response of the reported controller flow during periods of cycling flows was investigated at several flow ranges for several frequencies and amplitudes. Tests were made on both VAV boxes with nearly identical results. Most of the following figures show data for one VAV box only. Figures G2a to G2f show that the ALC sensor is at least as fast as the reference flow meter sensor and has the resolution to track any flow change produced. The accuracy, resolution and stability measurements were made with the damper forced to be fully open, but, as will be seen below, when the damper was put back to the normal controlled mode the reported flow data was constant for ten second blocks of time.

[1]Tom, Steve (ALC); "VAV Damper Control", Seminar 37 at the ASHRAE Winter Meting 2003

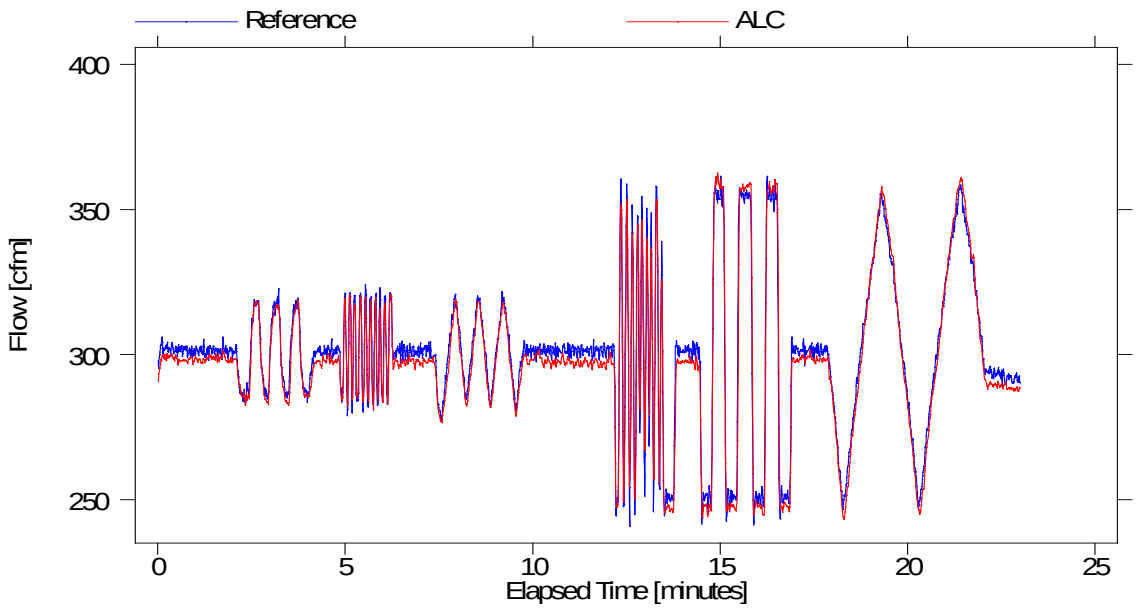


Figure G2a: Controller response to small changes in the flow at a flow of about 300 cfm.

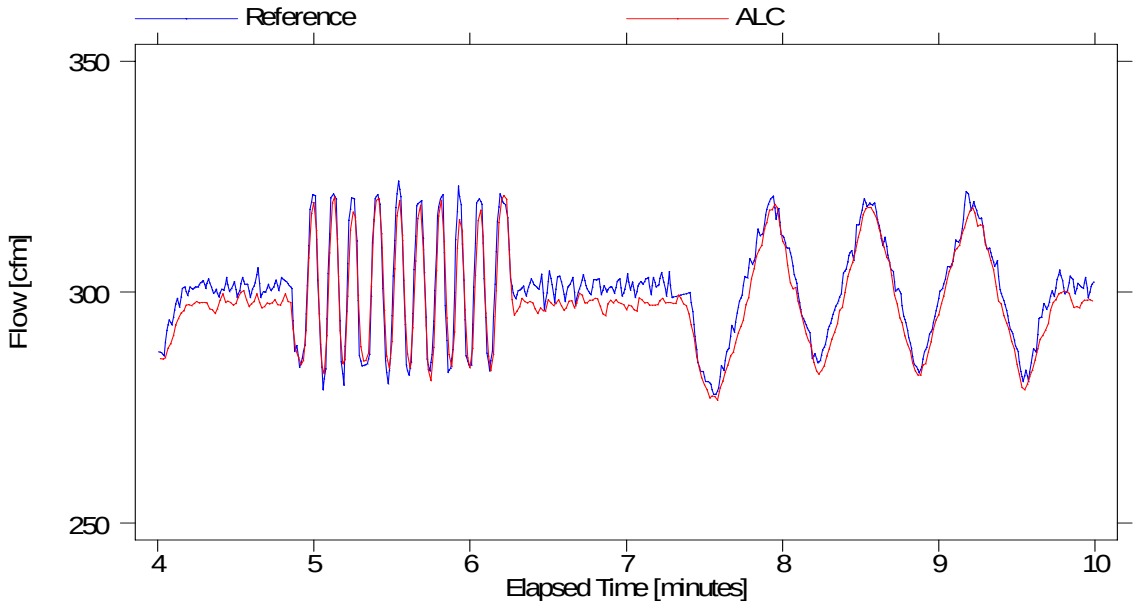


Figure G2b: Close up of Figure G2a, showing the tracking ability of the ALC sensor during periods of rapid flow change.

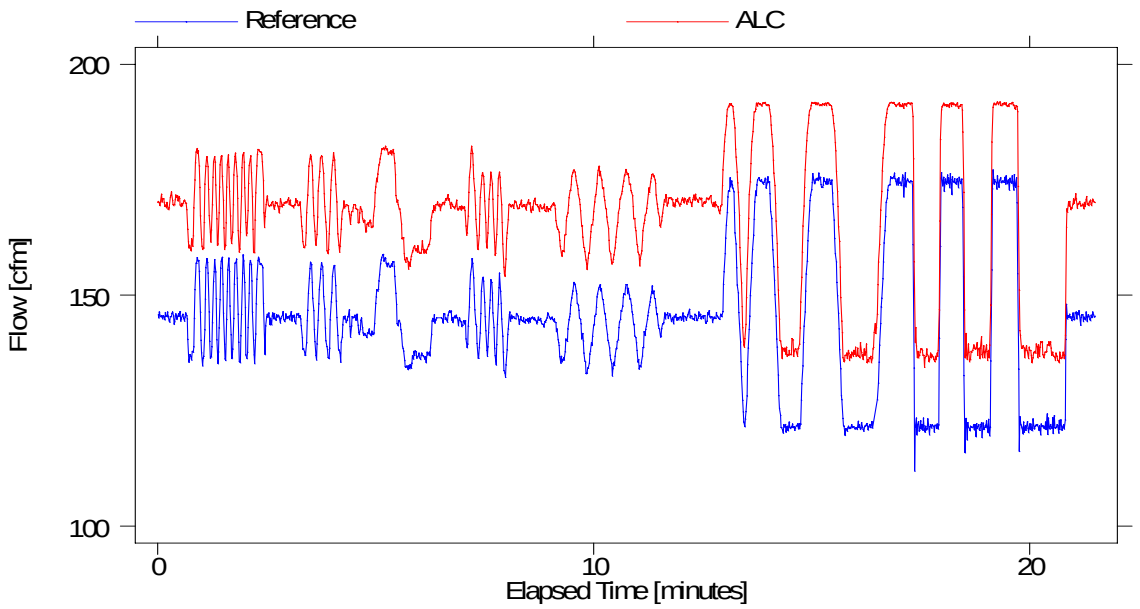


Figure G2c: Controller response to small changes in the flow at a flow of about 150 cfm.

In the flow range around 150 cfm the calibration is off by about 20 cfm.

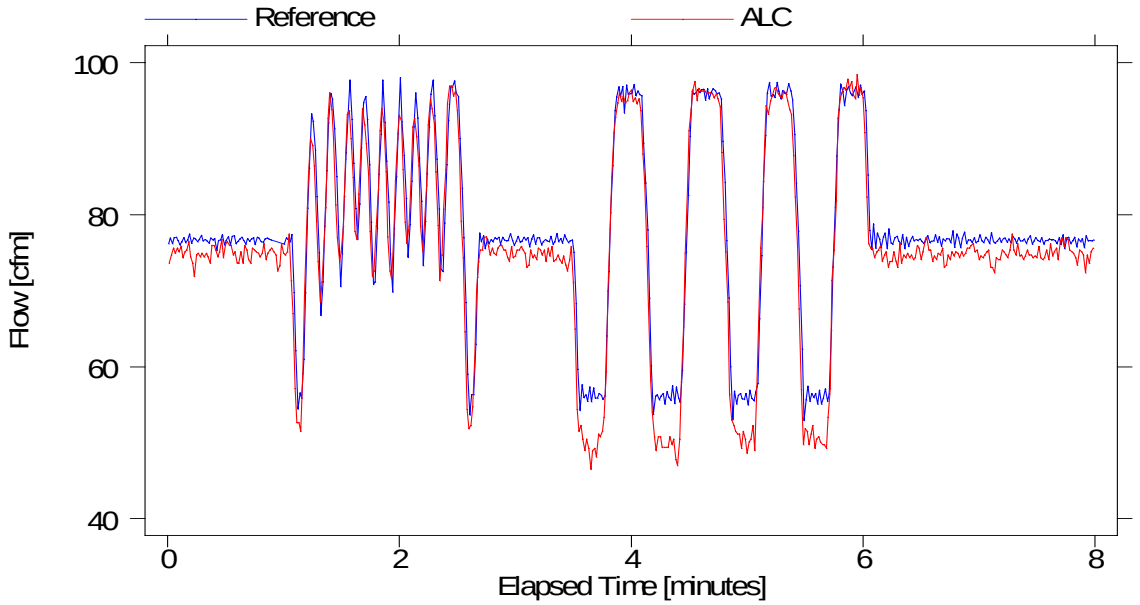


Figure G2d: Controller response to small changes in the flow at a flow of about 75 cfm.

One of the in-situ calibration points was 75 cfm, so the reference and the ALC values agree closely here except at the lowest flows.

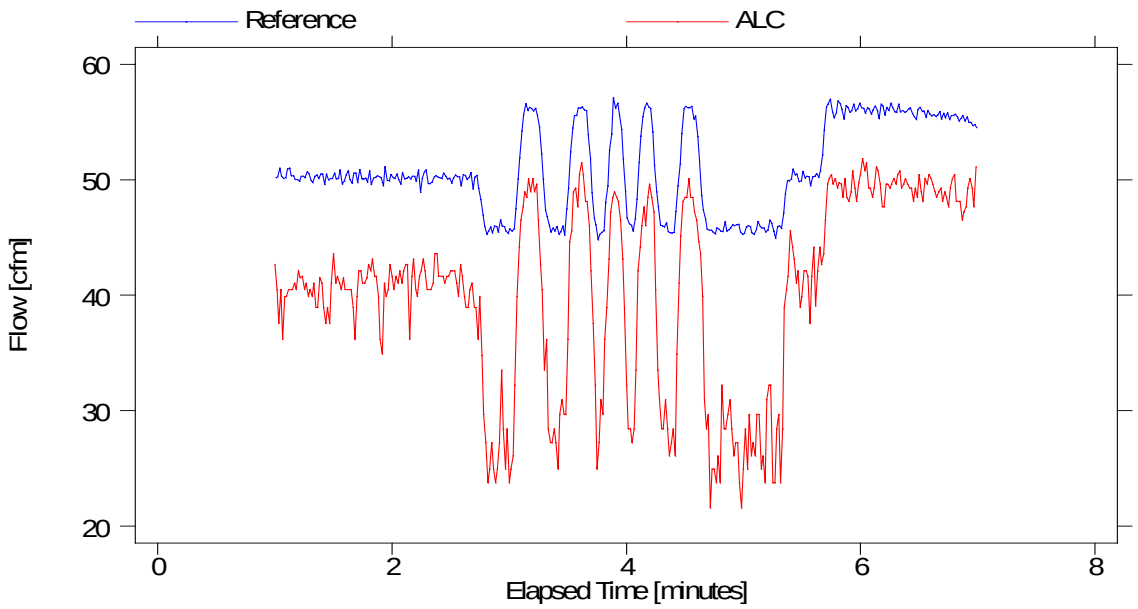


Figure G2e: Controller response to small changes in the flow at a flow of about 50 cfm.

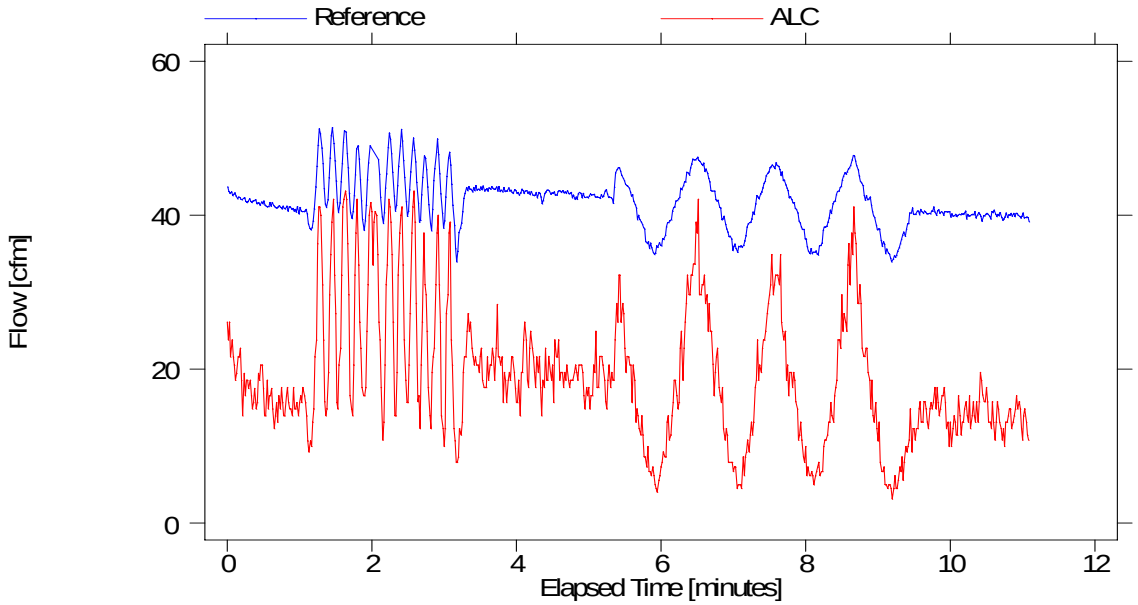


Figure G2f: Controller response to small changes in the flow at a flow of about 40 cfm.

Consistent with the calibration accuracy errors seen in Figure G1b, the reference and ALC flows progressively diverge as the flows decrease.

The observed resolution of the flow signal is better than 1 cfm in all the measurements shown in Figures G2a to G2f. The stated "sensitivity" is 0.001 iwc which would be 28 cfm at 0 and less than 1 cfm at 600 cfm. This is a confusing specification because the hot wire has a much more linear response to flow than a pressure based sensor, yet ALC is trying to specify it as if it were, the more common, pressure sensor.

Complete System Tests:

The complete system tests are intended to show how the controller operates when the controller is allowed to move the VAV damper to adjust the flow to achieve the flow set point. Tests were made at several different duct inlet static pressures for each of several flow set points. A summary of all the test results are presented in Tables G1 and G2.

Figures G3a and G3b show the controller response to changes in the inlet static pressure for a set point flow of 200 cfm for the Nailor and Titus VAV boxes and are typical of the tests at other flow set points. The ALC flow from these "complete system" tests have a much more stepped appearance than in the "resolution and stability" tests because the controller reported values stay constant for 10 seconds when the controller is not in a manual override state. These values appear to be a "snapshot" rather than an averaged or filtered value.

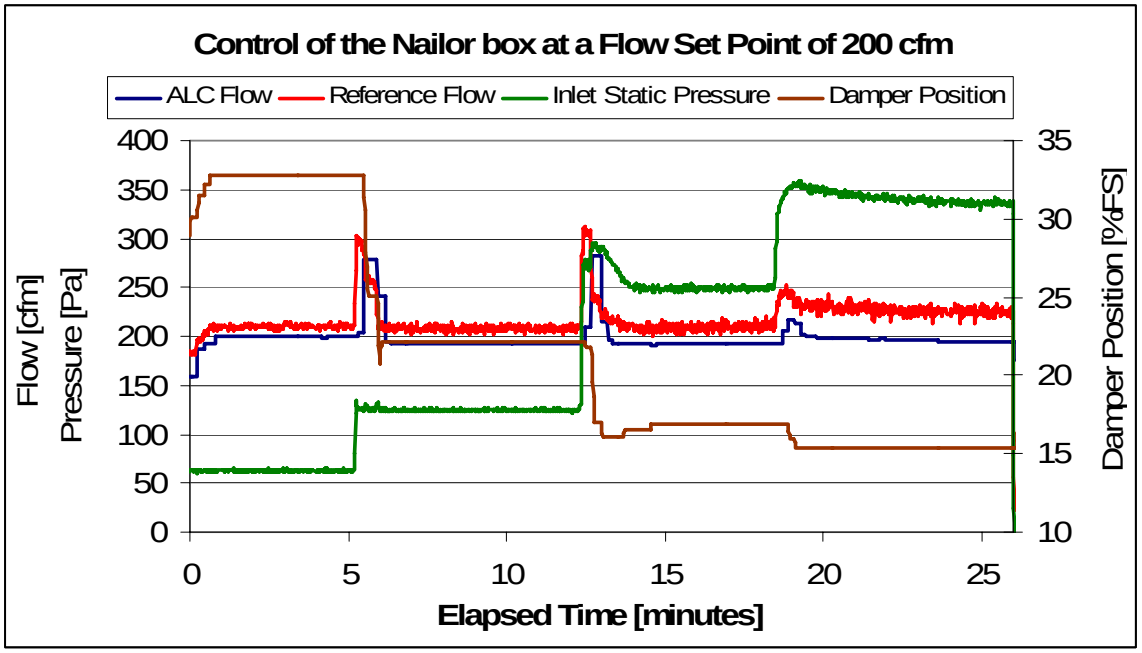


Figure G3a: Controller response to changes in inlet static pressure at a flow set point of 200 cfm for the Nailor VAV box.

After about 20 minutes, in Figure G3a, the inlet static pressure was increased to 350 Pa, but the fan was unable to maintain this pressure and the pressure drifts down which caused a decrease in the flow but not enough to trigger a change in the damper position.

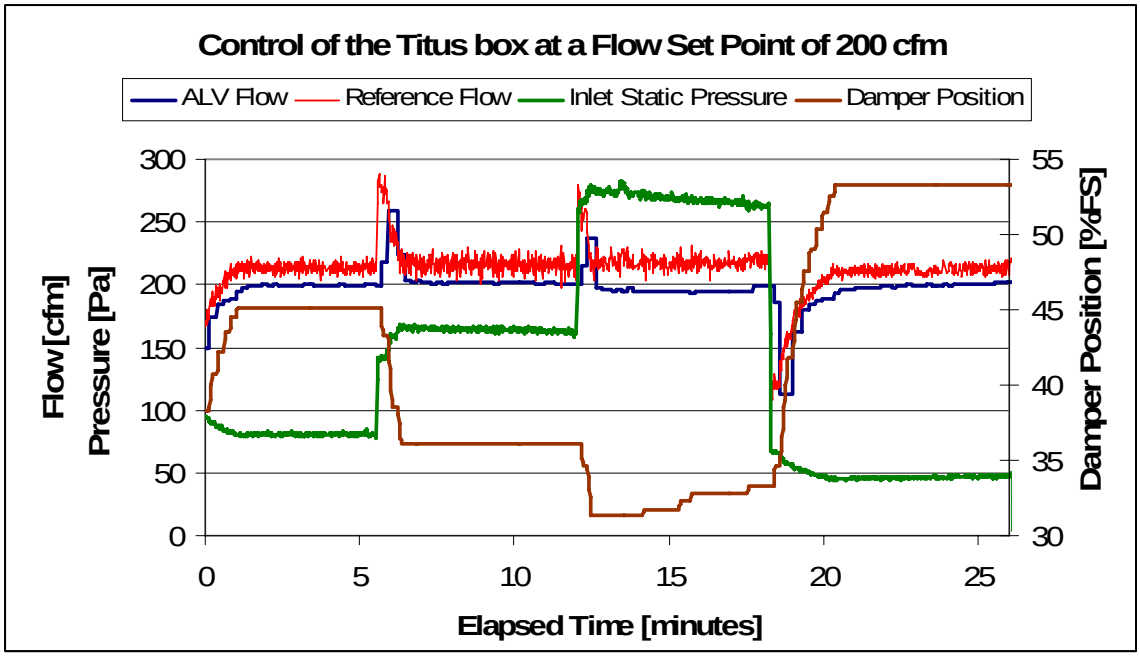


Figure G3b: Controller response to changes in inlet static pressure at a flow set point of 200 cfm for the Titus VAV box.

The damper keeps adjusting when the inlet static pressure was increased to greater than 250 Pa because this pressure was not stable.

Table G1: ALC Controller Operating the Nailor VAV Box					
Set Point Flow	Units: Flows in cfm, Damper Position in % Full Scale	Duct Inlet Static pressure [iwc]			
		<0.3	0.4 to 0.6	0.9 to 1.2	1.3 to 1.5
50	ALC Flow		48	50	51
	Reference Flow		59	62	66
	Damper Position		10	8.3	7.7
	# of Damper Adjustments		0	0	0
	Damper Adjustments/hour		0	0	0
	Minutes of Data		6.5	6.5	7
75	ALC Flow	77	76	77	81
	Reference Flow	81	82	85	94
	Damper Position	18.5	16.6	10	13.5
	# of Damper Adjustments	0	0	0	0
	Damper Adjustments/hour	0	0	0	0
	Minutes of Data	36	5	10.5	2.6
200	ALC Flow	200	193	192	196
	Reference Flow	210	208	209	226
	Damper Position	32.8	22.2	16.9	15.4
	# of Damper Adjustments	0	0	0	0
	Damper Adjustments/hour	0	0	0	0
	Minutes of Data	3.5	5.3	3	5

Table G2: ALC Controller Operating the Titus VAV Box						
		Duct Inlet Static pressure [iwc]				
Set Point Flow	Units: Flows in cfm, Damper Position in % Full Scale	0.2 to 0.3	0.4 to 0.6	0.7 to 0.8	0.9 to 1.1	1.1 to 1.3
50	ALC Flow		54	48		48
	Reference Flow		58	54		58
	Damper Position		21.6	20.7		18.6
	# of Damper Adjustments		0	0		0
	Damper Adjustments/hour		0	0		0
	Minutes of Data		6.5	7		6.5
75	ALC Flow	77	76		74	79
	Reference Flow	82	82		85	90
	Damper Position	27.6	24.5		22.5	22.5
	# of Damper Adjustments	0	0		0	0
	Damper Adjustments/hour	0	0		0	0
	Minutes of Data	36	12		6	12
200	ALC Flow	200		201	195	
	Reference Flow	212		216	218	
	Damper Position	53.3		36.1	32.7	
	# of Damper Adjustments	0		0	8	
	Damper Adjustments/hour	0		0	160	
	Minutes of Data	5		5.3	3	

There is only one instance of the damper needing adjustment during these “complete system” tests and that was when the inlet static pressure was not held constant.

Figure G4 shows the response of the controller to a change in flow set point. The ALC flow generally settles from one set point to the next in less than one minute.

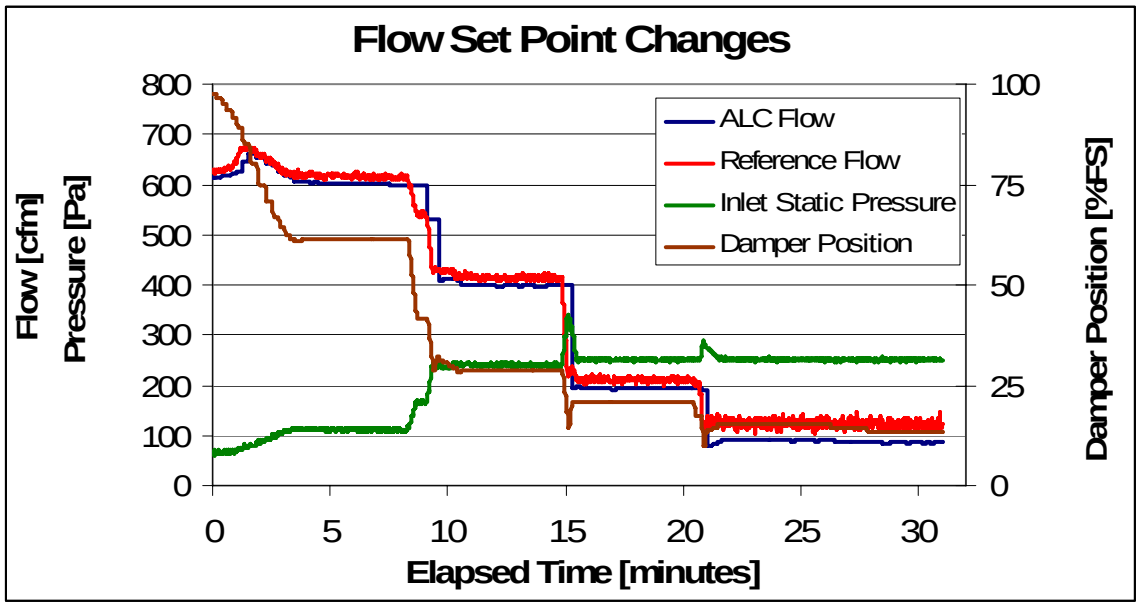


Figure G4: Controller response to a change in flow set point.

Pressure Sensor Zero Drift

Long term measurements were made to assess the magnitude of the zero drift of the pressure sensors. Unlike the other controllers studied, the ALC controllers can calculate an apparent negative flow so it is possible to directly observe the zero drift. The flow inlet was sealed to insure zero flow and data was taken for two and a half days.

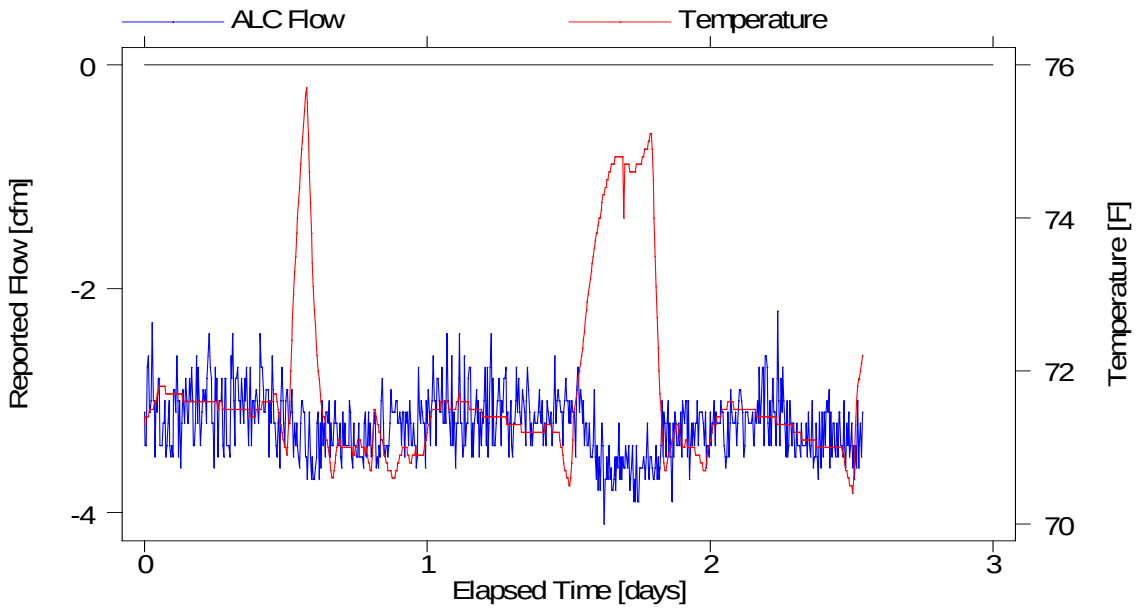


Figure G5: Zero drift of the ALC controller.

The zero drift seen is less than 1 cfm, which is about what the sensor noise was as well. There appears to be some diurnal trend to the zero value, but it is too small to be of concern.

Appendix H: Summary of Zone Controllers

Manufacturer	ALC	Johnson	Siemens	Invensys	Alerton
Model	ZN341v+/ZN141v+	(LON/BACnet controller)	APOGEE Products End Devices and Controllers Terminal Box Controller model #540-100	MNB-V2 (new BACNET model)	VAV-SD
Transducer Make	Honeywell	Setra	Kavlico	Invensys	
Transducer Model	Microbridge				
Transducer Type	Hot wire	Differential pressure	steel diaphragm (differential pressure)	Differential pressure	hot wire
A/D converter resolution	10 bit			10 bit	
Damper Actuator			GDE131.1P		LM24-3-T US
RAM	512k				
Software / Algorithm	Slope-Based Control*				Envision for BACtalk
Auto zero frequency			12 hours		
VP Control Range				0.004 to 1.5 in. W.C.	0–1.25"
Listed Accuracy			The accuracy, assuming an amplification factor of 1, is listed as +-5% from 300 to 4000 fpm, and +-15% from 200 fpm to 300 fpm.	±5% at 1.00 in. of W.C. with laminar flow at 77 °F (25 °C) and suitable flow station.	
Listed Resolution			4 CFM		
Manufacturer Contact	Steve Tom for algorithm		Jim Coogan	John Sullivan	
Controls contractor contact	Tony Skibinski, Air Systems	John Burgess, JCI	Dennis Thompson, Siemens	Shad Buhlig, Automatic Controls	Eddie Olivares, Syserco

Appendix I: Simulated Energy Performance of Three Zone Control Sequences

A. Summary

DOE-2.2 was used to compare the energy performance of three zone control sequences: Single Maximum, Dual Maximum with VAV Heating and Dual Maximum with Constant Volume Heating. These three sequences are described in detail below.

The basecase model is a typical office building in Sacramento with a packaged VAV and hot water reheat system. This model was also run in San Francisco, Los Angeles, Chicago and Atlanta. Numerous parametric analyses were also run to determine the impact of supply air temperature reset, of single maximum sequences with 40% and 50% minimums, of oversized zones, of systems that are left running 24/7 and of very lightly loaded buildings.

In the basecase model the Dual Max-VAV saved 5 cents/ft^2-yr compared to the single maximum but the Dual Max-Constant Volume actually used 2 cents/ft^2-yr more energy than the Single Maximum case even though it has a lower flow in deadband (20% versus 30%). As shown in Figure 1, the Dual Max-VAV savings go down if supply air temperature reset is employed and go up if the zones are oversized, if the fan runs 24/7 or if the minimum flow for the Single Maximum sequence is higher than 30%. It is estimated that the average savings of the Dual Max-VAV sequence for a typical office building would be approximately 10 cents/ft^2-yr (0.5 kWh/ft^2-yr and 0.08 therms/ft^2-yr). The Dual Max-Constant Volume never saves as much as the Dual Max-VAV and in many cases uses more energy than the Single Maximum. On average, the Dual Maximum-CV is no more efficient than the Single Maximum control sequence.

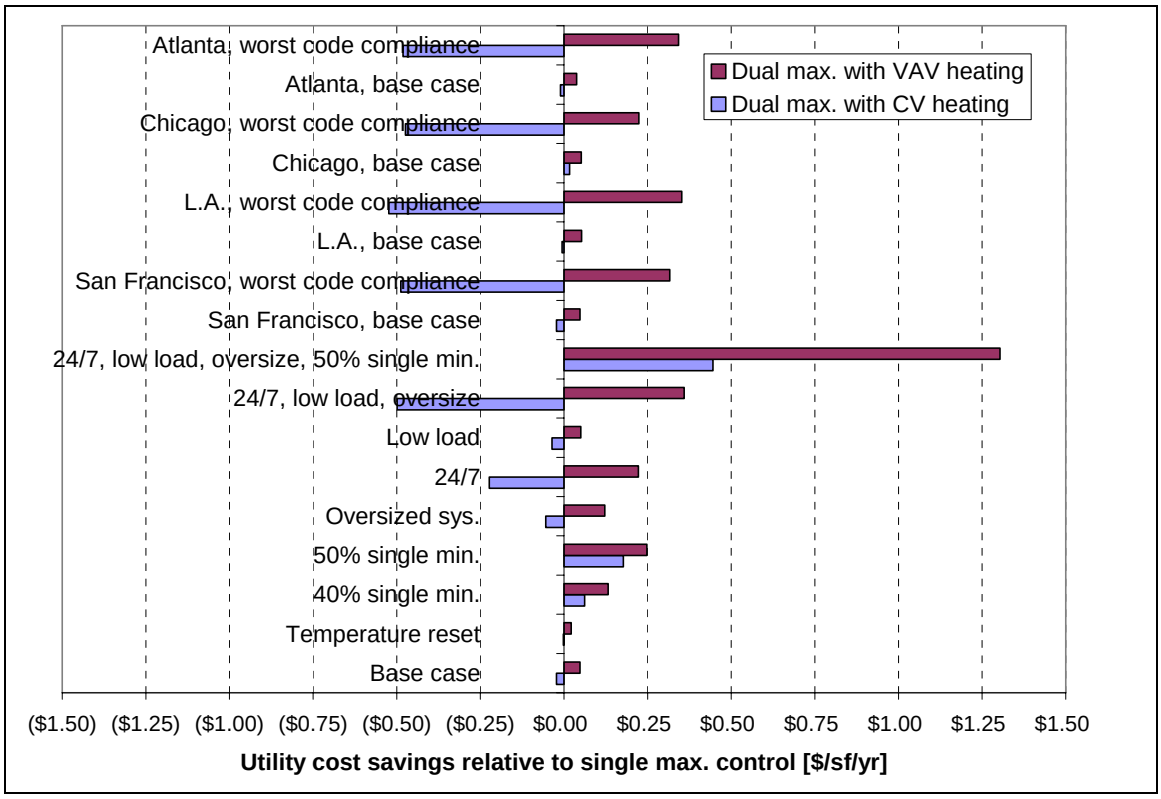


Figure 1 Annual utility cost savings

B. Control Sequences

1. Single Maximum

The sequence of control is show in Figure 2. As cooling load decreases the airflow is reduced from the maximum airflow (on the far right side of the figure) down to the minimum flow. Then as heating is required the reheat valve is modulated to maintain the space temperature at setpoint. With this sequence the minimum flow rate in deadband (between heating and cooling) is also the flow rate in heating mode.

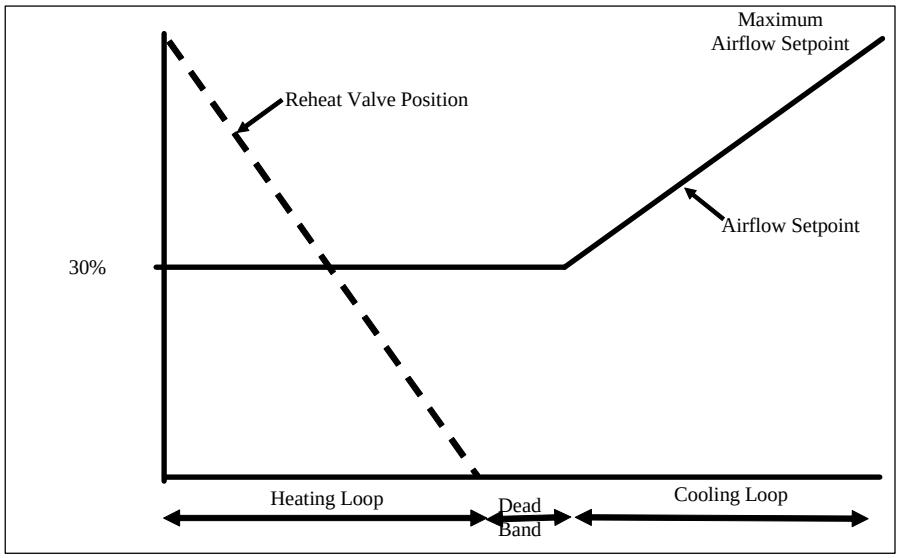


Figure 2. Single Maximum Zone Control Sequence

2. Dual Maximum-VAV Heating

Figure 3 illustrates a dual maximum zone control sequence with variable air volume in heating. Airflow is reduced from cooling maximum airflow to minimum airflow as cooling load reduces; as the zone enters into heating mode the discharge air temperature setpoint is reset from minimum temperature (e.g. 55°F) to maximum temperature allowed by reheat coil's capacity, in this case, 95 °F. If more heating is required then the airflow is reset from the minimum up to the heating maximum. With a dual maximum zone control sequence, the airflow in deadband is lower than the airflow at full heating. The minimum flow needs to only be high enough to satisfy the ventilation requirements, which are can be 10% or less for perimeter zones.

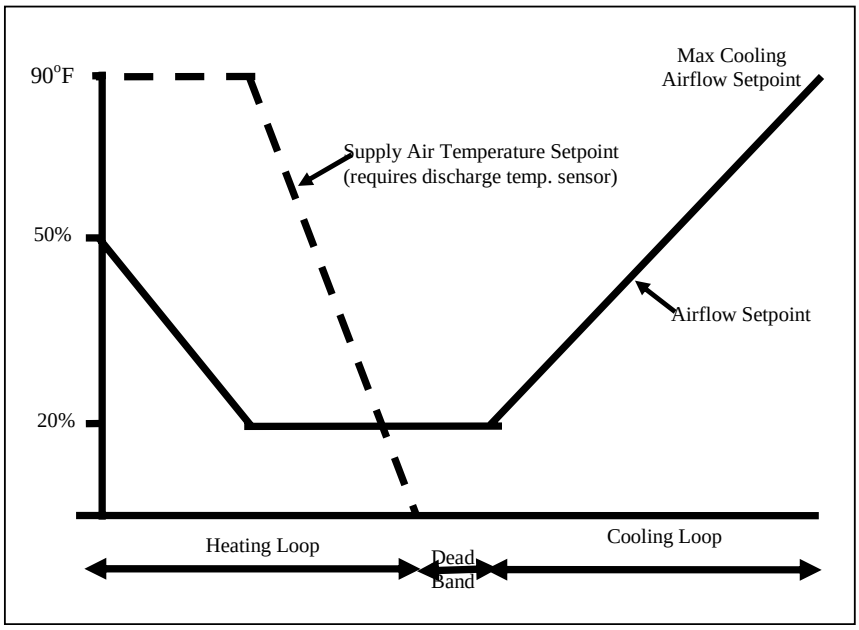


Figure 3. Dual Maximum with VAV Heating

3. Dual Maximum with Constant Volume Heating

Figure 4 shows the control sequence of dual maximum with constant volume heating. As cooling load decreases the airflow is reduced from the maximum airflow (on the far right side of the figure) down to the minimum flow; as heating is required the airflow is reset from the minimum to the heating maximum and the reheat valve is modulated to maintain the space temperature at setpoint.

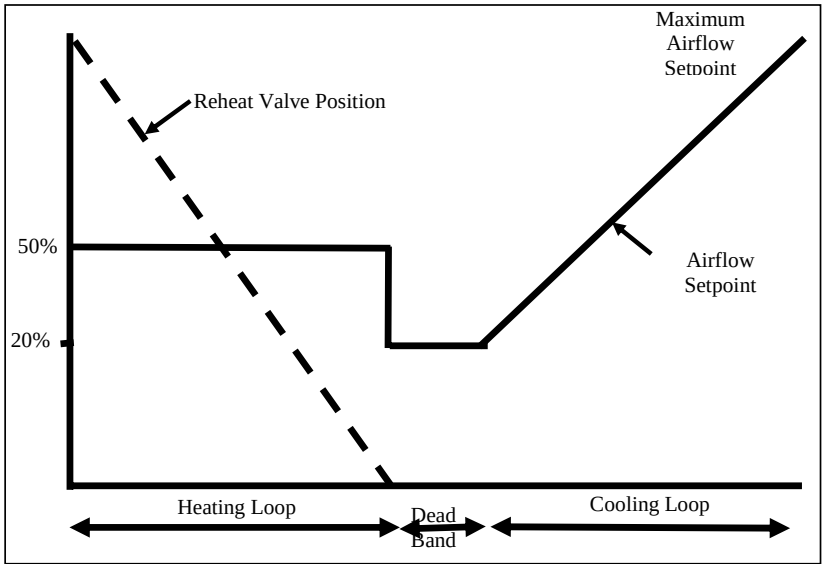


Figure 4. Dual Maximum with CV Heating

C. Model Description

A 10,000 ft^2 five zone office building was modeled in eQuest to evaluate annual energy performance of the three zone control sequences. Figure 5 shows the dimension of the model.

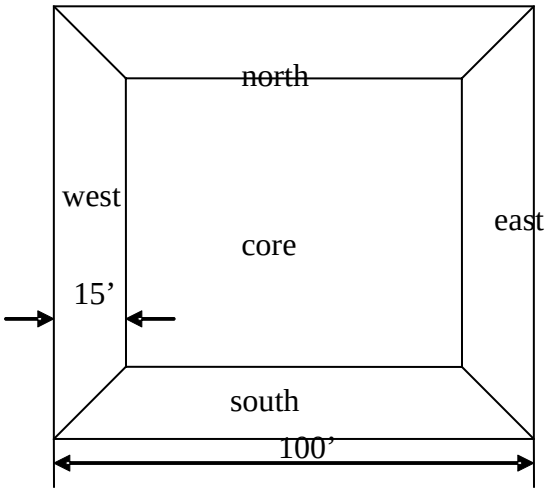


Figure 5 Model dimensions

The building envelope consists of R-19 metal frame roof and R-13 metal frame wall with 40% window wall ratio. All windows use double pane glazing. The U value of the glass is 0.47 and the SHGC value of the glass is 0.31 for non-north facing windows and 0.47 for north facing windows.

The building was modeled to be occupied from 7:00 am to 7:00 pm Monday through Friday and was closed on Saturday, Sunday and holidays. Building internal loads consist of an average 100 sf per person occupancy density, 1.3 w/sf lighting power densities and 1.5 w/sf equipment power density.

In order to simulate "real-life" building operation, five occupancy day schedules were modeled as shown in Figure 6. The simulation models were set up such that on any weekday, each of the five zones uses one of the schedules shown in Figure 6 and no two zones use the same schedule on the same day. From Monday to Friday, each zone uses a different day schedule on a different day. Lighting and equipment schedule are the same as the occupancy schedule. This is a simplifying assumption which assumes that people turn off lights in proportion to occupancy, which might not be the case with multi-occupant spaces. Similarly, equipment loads may not drop off as rapidly as occupancy. On the other hand, these schedules are somewhat arbitrary so having separate schedules for lighting and equipment does not necessarily make sense. Furthermore, parametrics were performed to determine the sensitivity of the results to changes in schedules.

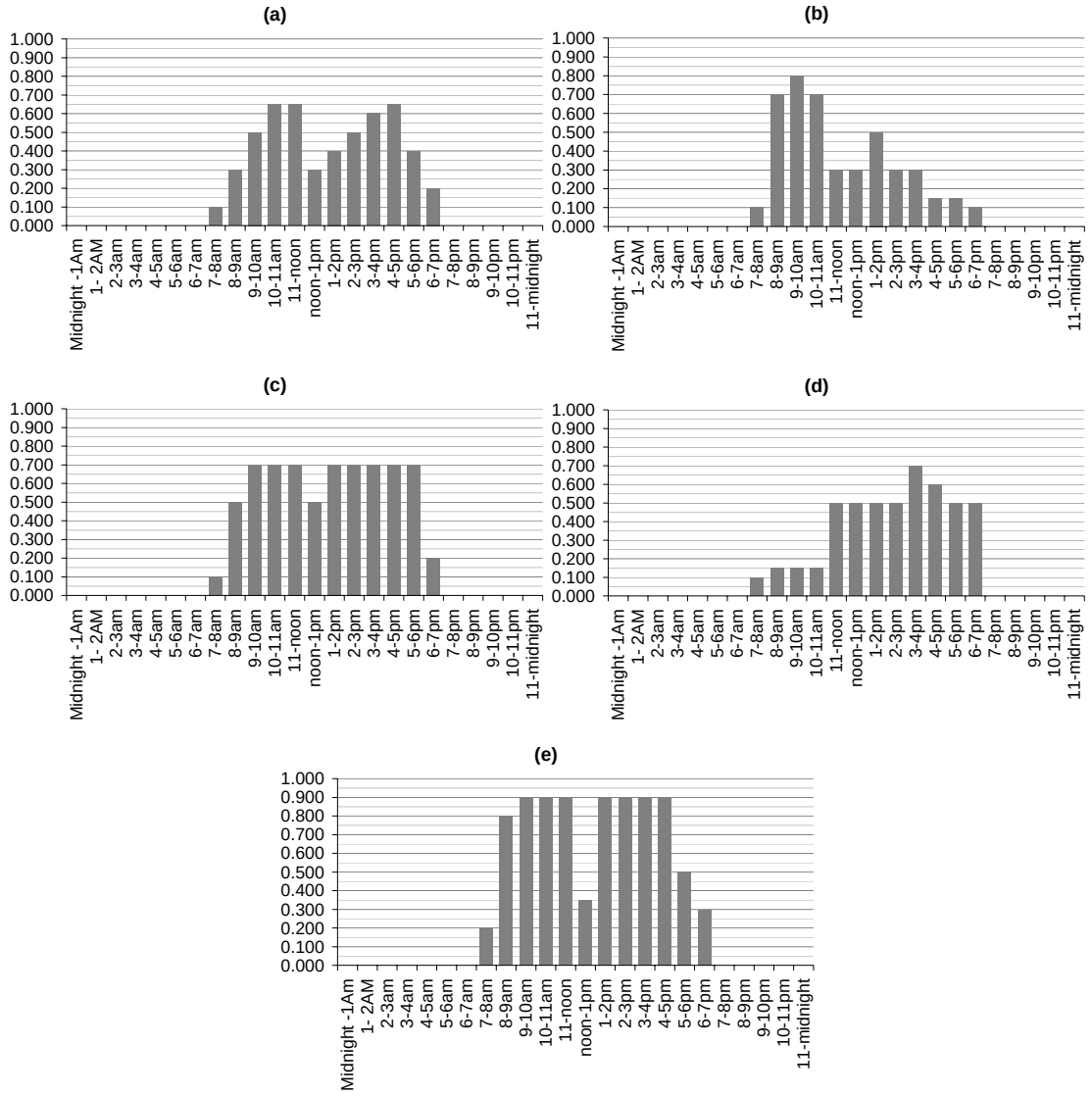


Figure 6 Occupancy schedule

The building is conditioned by a packaged VAV system with hot water reheats at VAV boxes. Room temperature setpoint are 75/82 for cooling and 70/64 for heating during occupied/unoccupied hours. The HVAC system runs from one hour before occupancy to one hour after occupancy. System supply air temperature is fixed at 55°F in the basecase. A DOE-2 fan curve that represents a variable speed drive and demand-based static pressure reset was used for all runs.

The model was run using the weather data representing Sacramento, CA (climate zone 12) which is a relatively hot climate in California.

Three VAV control sequences were investigated, the detailed modeling assumptions for each of the control is are shown in Table 1 and in the screen captures from eQuest below.

Table 1 Basecase modeling assumptions

	Case #	Single max.	Dual max. with CV heating	Dual max. with VAV heating
HVAC System	System Type	PVAVS	-ditto-	-ditto-
	Sizing Ratio	1.25	-ditto-	-ditto-
	Fan Control	VSD	-ditto-	-ditto-
	Air Flow	min Fan ratio = 0.1, max Fan ratio = 1.1	-ditto-	-ditto-
	Fan Eff.	SA Fan 53%, RA Fan 53%	-ditto-	-ditto-
	Fan Performance Curve	VSD fan with SP reset		
	Fan static pressure	3.5"	-ditto-	-ditto-
	OA ratio	Default (calc. from zone OA CFM)	-ditto-	-ditto-
	Economizer	differential drybulb, max temperature limit = 59	-ditto-	-ditto-
	Cooling EIR	0.36 (9.5 EER)	-ditto-	-ditto-
	Min SAT	55. °F	-ditto-	-ditto-
	Max Cooling SAT Reset Temp	59. °F	-ditto-	-ditto-
	Cooling SAT temp control	Constant	-ditto-	-ditto-
	Heating SAT temp control	Constant	-ditto-	-ditto-
	Heating Coil	No coil at packaged unit, only hot water reheating coil at each zone	-ditto-	-ditto-
	RH Coil Valve	3-way valve	-ditto-	-ditto-
	Min Heating Reset Temp	75. °F	-ditto-	-ditto-
Zone (each)	Thermostat	Proportional	Proportional	ReverseAction
	Throttling Range	.1 °F	-ditto-	-ditto-
	Cooling Min Flow Ratio	30%	20%	20%
	Cooling Max Flow Ratio	100%	100%	100%
	Heating Min Flow Ratio	30%	50%	20%
	Heating Max Flow Ratio	30%	50%	100%
	Cooling setpoint	75. °F	-ditto-	-ditto-
	Heating setpoint	70. °F	-ditto-	-ditto-
	Cooling setpoint unoccuppied	82. °F	-ditto-	-ditto-
	Heating setpoint unoccuppied	64. °F	-ditto-	-ditto-
Boiler Plant	Boiler HIR	1.25	-ditto-	-ditto-
	Design HWST	180 °F	-ditto-	-ditto-
	Design HW loop dT	40 °F	-ditto-	-ditto-
	HW loop pump control	one speed pump	-ditto-	-ditto-
Building Envelope	Exterior wall U value	R-13 (code)	-ditto-	-ditto-
	Roof U value	R-19 (code)	-ditto-	-ditto-
	WWR	40%	-ditto-	-ditto-
	Glass Type	U = 0.47, SHGC = 0.31 (nonnorth), 0.47 (north)	-ditto-	-ditto-
	Area	100 ft by 100 ft, 15 ft perimeter zone depth	-ditto-	-ditto-
Building Internal Load	Occupancy	100 sf/person	-ditto-	-ditto-
	Lighting	1.3 w/sf	-ditto-	-ditto-
	Equipment	1.5 w/sf	-ditto-	-ditto-
	Schedule	Occupied 7:00 ~19:00 M-F, Unoccupied other days	-ditto-	-ditto-
Climate Zone		CZ 12 (Sacramento)	-ditto-	-ditto-

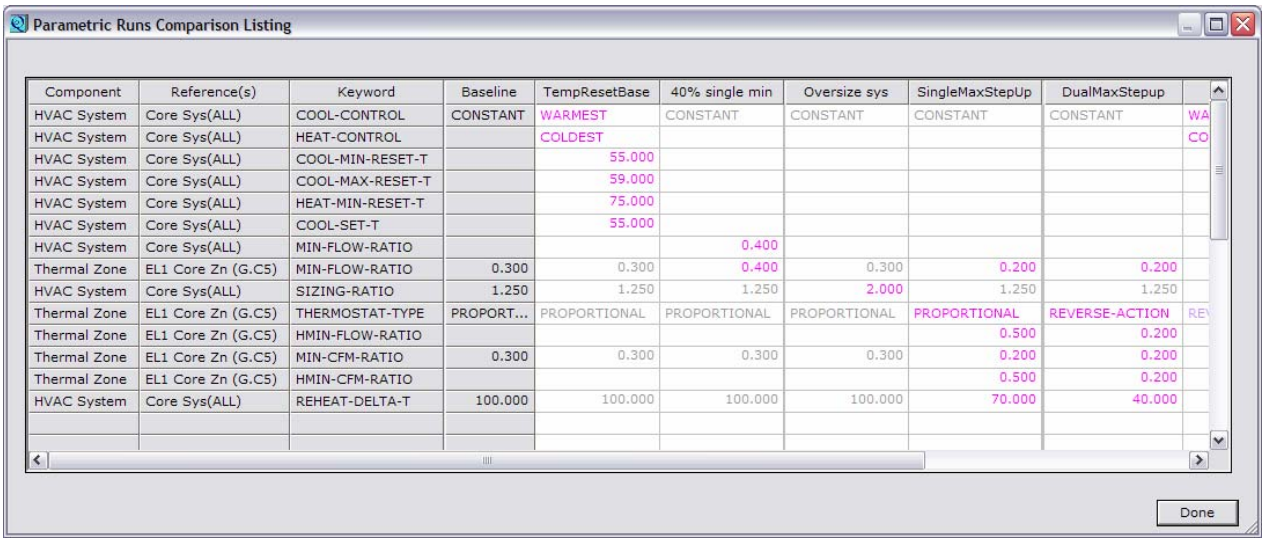


Figure 7 eQuest parametric run inputs

Note that Thermal Zone parametric References in Figure 7 only refer to the Core Zone. This is because the inputs for the perimeter zones are linked to the core zone.

1. Utility Rates

The PG&E Sch-A10a Electricity Rate was used in all runs. The virtual electricity rate (including demand charges) was approximately $0.18/kwh for all runs.

The PG&E GNR-1 Gas Rate was used for all runs. The virtual gas rate was approximately $0.60/therm.

D. Basecase Results

Table 2. Summary of Energy Consumption of Three Control Sequences

	Electricity		Nature Gas		Annual Utility Costs:						
	Annual Energy	Peak Demand	Annual Energy	Peak Demand	ES-E Total Electric	ES-E Total Fuel	ES-E Electric kWh	ES-E Electric kW	**Utility Cost Savings**	**Electrical Energy Savings**	**Natural Gas Savings**
	kWH	kW	Therms	Therms/Hr	$	$	$	$	**$/sf/yr**	**Wh/sf/yr**	**kBtu/sf/yr**
Base case: Sizing Ratio = 1, No Temperature Reset at Cooling coil, Sacramento											
Single Max	68223	55	1710	4.3	12768	1106	9605	2263	-	-	-
Dual Max with CV Heating	68330	56	2092	4.8	12776	1323	9590	2286	**($0.02)**	**(10.69)**	**(3.82)**
Dual Max with VAV Heating	65688	56	1419	5.2	12448	947	9274	2273	**$0.05**	**253.59**	**2.91**

Energy consumption of the three control sequences are listed and compared in Table 2. Results show that with current modeling assumptions, the Dual Maximum-VAV control sequences saves about $0.05 $/sf/yr, 253.6 Wh/sf/yr and 2.9 kBtu/sf/yr compared with Single Maximum control. The Dual Maximum with CV Heating control used more energy than the Single Maximum control. This is due to higher space heating, fan and pumping energy from the higher flow rate in heating mode (50% vs. 30%).

Figure 8 through Figure 11 show the zone VAV damper control, reheating coil output and fan flow histograms for the basecase scenario. It can be seen that the control sequence is well simulated. From the histogram it is clear that there are very few hours where the airflow is above the desired 50% in heating for the Dual Maximum-VAV

sequence. Thus it accurately represents the desired sequence (even though DOE-2 does not have a keyword for heating maximum)

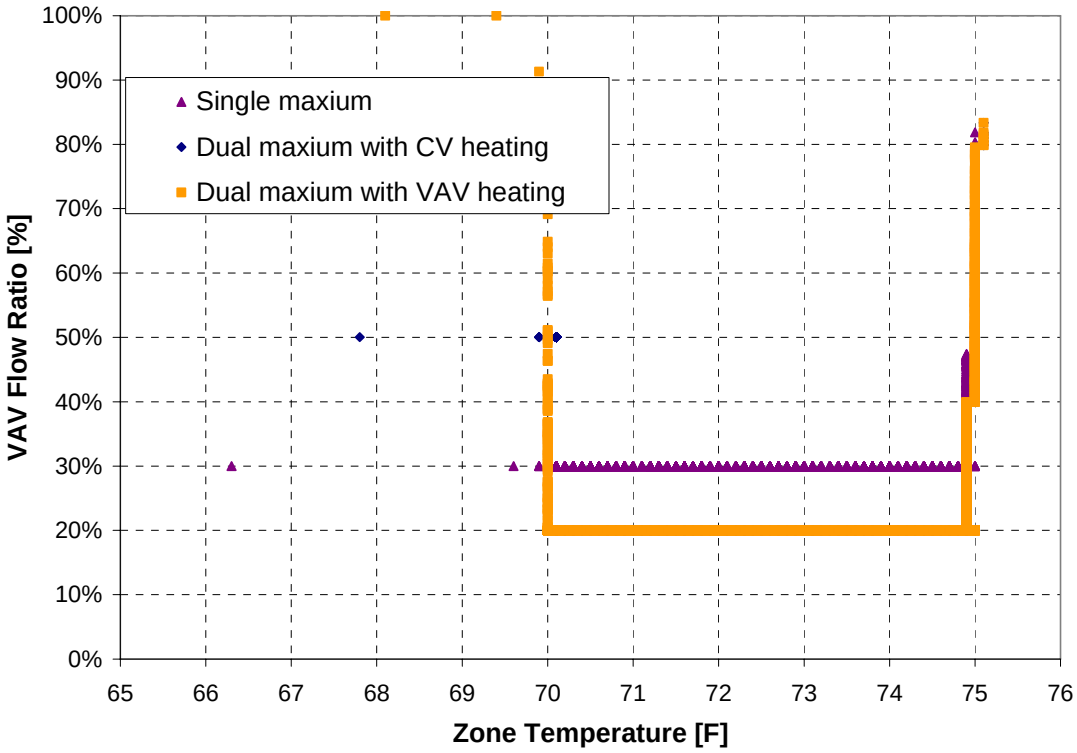


Figure 8 VAV flow ratio and heating for one zone

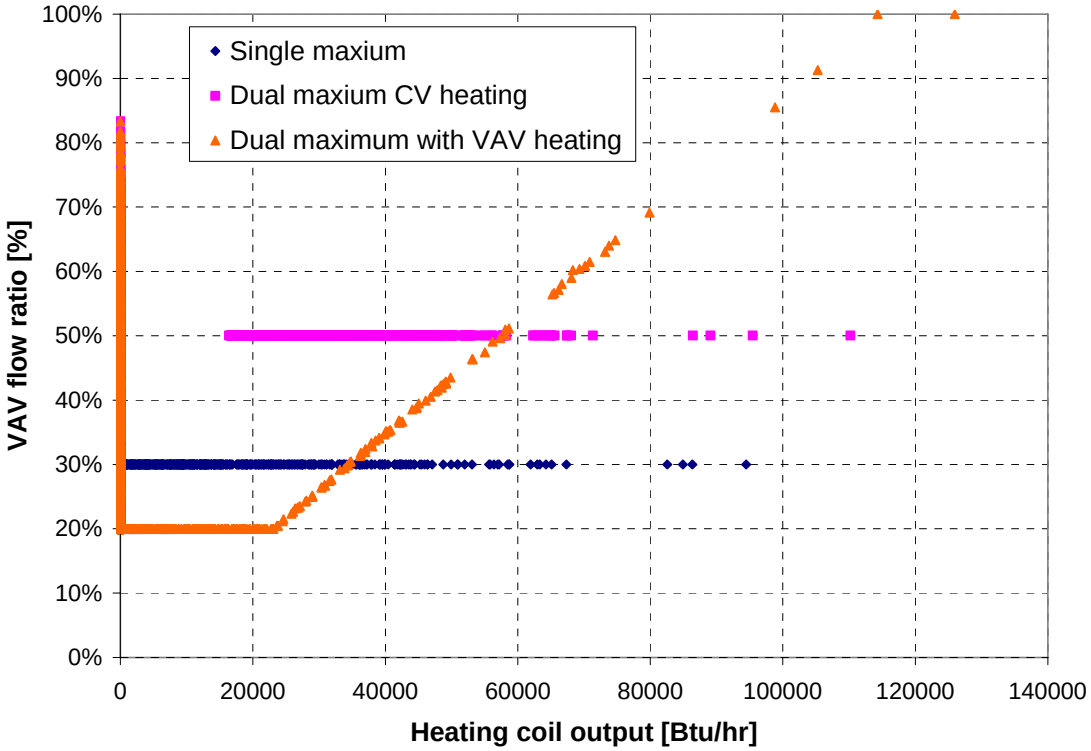


Figure 9 VAV flow control for one zone

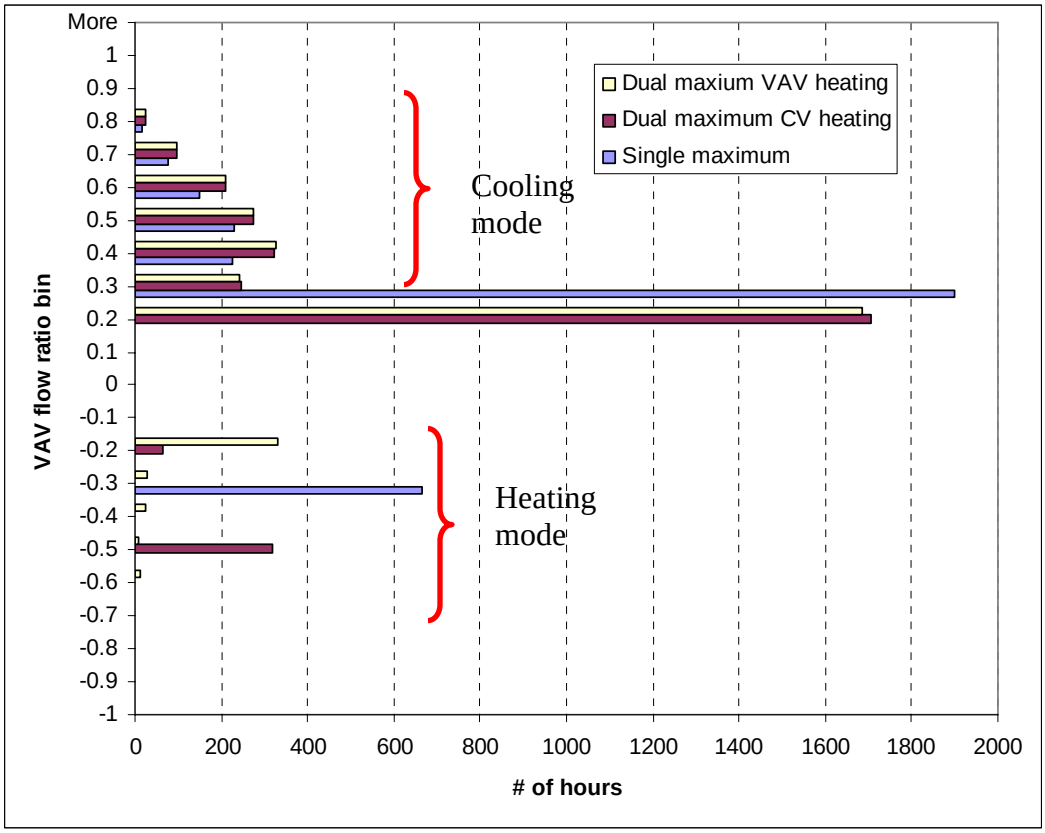


Figure 10 VAV flowrate histogram for one zone

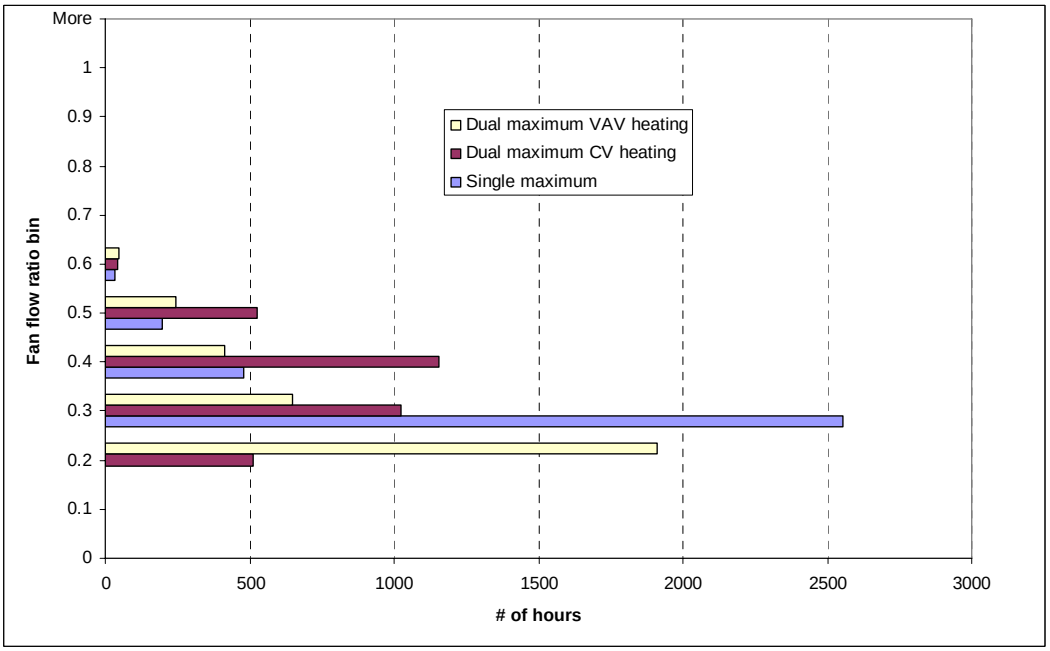
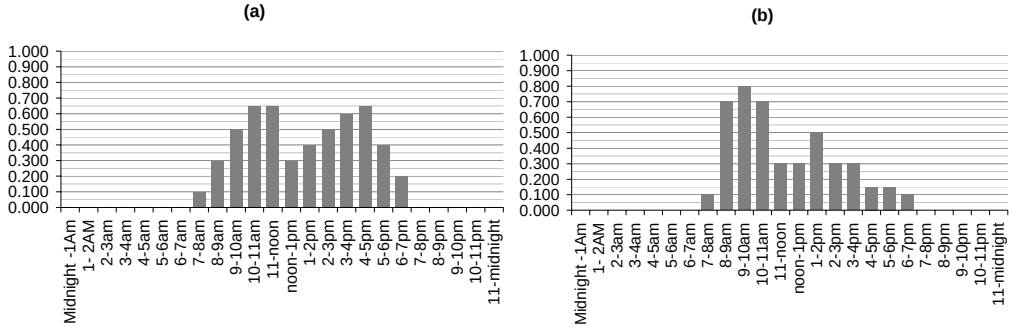


Figure 11 Fan flow histogram

E. Parametric Analysis

Parametric runs were carried out to evaluate the energy saving potential of different VAV control sequences under different scenarios. Descriptions of the parametric runs are as following.

1. **Basecase**. Base case runs are described above in section I.C;
2. **SAT Reset**. Supply air temperature reset control was enabled for all three VAV control sequences. Cooling supply air temperature resets based on the "warmest" zones and can be reset from 55°F up to 59°F. In the basecase it is fixed at 55°F. The maximum supply air temperature is limited to 59°F (as opposed to say 65°F) because DOE-2 seems to almost always peg the supply air temperature at the maximum allowed which is not necessarily realistic.
3. **40% Single Minimum**. The minimum VAV flow ratio for single maximum control sequence is 40%. While this does not generally comply with code it is often done in practice in order to meet design heating loads at reasonable supply air temperatures which limit stratification.
4. **50% Single Minimum**. The minimum VAV flow ratio for single maximum control sequence is 50%. Similarly, this is commonly seen in the field.
5. **Oversized Zones**. The DOE-2.2 sizing ratio was set to 200% for all three control sequences compared to 125% in the basecase.
6. **24/7 Operation**. The HVAC system runs 24-hours a day and 7 days a week. The cooling temperature setpoint is kept at 75°F, heating temperature setpoint is kept at 70°F 24 hours a day and 7 days a week;
7. **Low Load Profile**. The daily occupancy/lighting/equipment schedules have a reduced profile as shown in Figure 12.



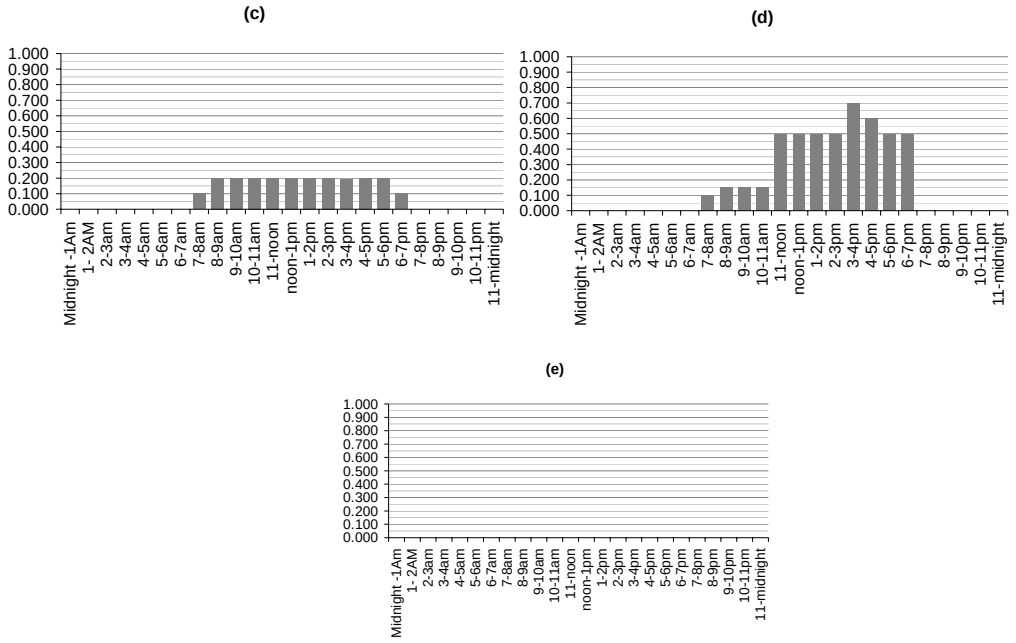


Figure 12 Low-load day schedules

8. **Worst Case Code Compliant.** This is the combination of 5, 6 and 7 above. For all three control sequences, the HVAC system is sized to be 200% of the load, systems runs 24/7, and uses low-load profile.
9. **Worst Case**. This is the same as 8 above except that 50% minimum flow rate ratio is used in the single maximum control.
10. **Different locations**. Four different locations are analyzed besides Sacramento in the basecase. These are: San Francisco, Los Angeles, Chicago and Atlanta.

Table 3 shows how the parametric runs are organized. Note that X means run all three control sequences. The cells highlighted in yellow are illustrated in more detail below.

Table 3 Parametric runs

	Sacramento	San Francisco	Los Angeles	Chicago	Atlanta
Basecase	X	X	X	X	X
SAT reset	X				
40% single min	X				
50% single min	X				
Oversize HVAC	X				
24/7	X				
low load profile	X				
worst case code compliant: oversize HVAC, 24/7, low load	X	X	X	X	X
worst case: 50% single min, oversize HVAC, 24/7, low load	X				

The results of the parametric runs are shown in Table 4.

Table 4 Summary of parametric runs

	Electricity		Natural Gas		Annual Utility Costs:						
	Annual Energy	Peak Demand	Annual Energy	Peak Demand	ES-E Total Electric	ES-E Total Fuel	ES-E Electric kWh	ES-E Electric kW	**Utility Cost Savings**	**Electrical Energy Savings**	**Natural Gas Savings**
	kWH	kW	Therms	Therms/Hr	$	$	$	$	**$/sf/yr**	**Wh/sf/yr**	**kBtu/sf/yr**
Base case(Sizing Ratio = 1, No SAT Reset, Sacramento)											
Single Max	68223	55	1710	4.3	12768	1106	9605	2263	-	-	-
Dual Max with CV Heating	68330	56	2092	4.8	12776	1323	9590	2286	($0.02)	(10.69)	(3.82)
Dual Max with VAV Heating	65688	56	1419	5.2	12448	947	9274	2273	$0.05	253.59	2.91
Supply Air Temperature Reset at Cooling Coil allow 55 ~ 59 °F SAT											
Single Max	65560	56	1215	4.3	12457	834	9281	2276	-	-	-
Dual Max with CV Heating	65240	56	1332	4.8	12407	902	9224	2283	($0.00)	32.08	(1.17)
Dual Max with VAV Heating	64251	56	1121	5.2	12291	783	9110	2281	$0.02	130.90	0.94
40% single min											
Single Max	71941	55	2351	5.1	13264	1453	10108	2257	-	-	-
Dual Max with CV Heating	68330	56	2092	4.8	12776	1323	9590	2286	$0.06	361.02	2.60
Dual Max with VAV Heating	65688	56	1419	5.2	12448	947	9274	2273	$0.13	625.30	9.33
50% single min											
Single Max	77225	54	3114	5.5	14010	1864	10850	2260	-	-	-
Dual Max with CV Heating	68330	56	2092	4.8	12776	1323	9590	2286	$0.18	889.49	10.23
Dual Max with VAV Heating	65688	56	1419	5.2	12448	947	9274	2273	$0.25	1153.77	16.96
Oversize sys											
Single Max	76109	52	3012	5.5	13811	1808	10710	2201	-	-	-
Dual Max with CV Heating	77086	53	3825	6.1	13890	2270	10741	2249	($0.05)	(97.68)	(8.13)
Dual Max with VAV Heating	70284	53	2226	6.2	13010	1386	9902	2207	$0.12	582.50	7.85
24/7											
Single Max	89199	51	6486	2.0	15712	3669	12684	2128	-	-	-
Dual Max with CV Heating	95991	52	8907	2.6	16623	4987	13573	2150	($0.22)	(679.15)	(24.21)
Dual Max with VAV Heating	79400	52	4904	1.9	14330	2828	11284	2146	$0.22	979.94	15.82
Low load											
Single Max	45006	42	1981	3.9	8934	1255	6379	1655	-	-	-
Dual Max with CV Heating	45787	42	2457	4.4	9022	1524	6434	1688	($0.04)	(78.10)	(4.76)
Dual Max with VAV Heating	42257	42	1709	4.7	8577	1109	6008	1669	$0.05	274.83	2.73
24/7, LOW-LOAD, OVERSIZE											
Single Max	83426	36	10380	2.5	14482	5742	12011	1570	-	-	-
Dual Max with CV Heating	101622	34	15135	3.5	16904	8306	14481	1523	($0.50)	(1819.59)	(47.55)
Dual Max with VAV Heating	69034	34	7723	2.3	12303	4328	9939	1464	$0.36	1439.26	26.58
24/7, LOW-LOAD, OVERSIZE,50% single min											
Single Max	119672	44	17648	3.8	20058	9609	17189	1969	-	-	-
Dual Max with CV Heating	101622	34	15135	3.5	16904	8306	14481	1523	$0.45	1805.00	25.13
Dual Max with VAV Heating	69034	34	7723	2.3	12303	4328	9939	1464	$1.30	5063.85	99.25
San Francisco Base											
Single Max	68223	55	1710	4.3	12768	1106	9605	2263	-	-	-
Dual Max with CV Heating	68330	56	2092	4.8	12776	1323	9590	2286	($0.02)	(10.69)	(3.82)
Dual Max with VAV Heating	65688	56	1419	5.2	12448	947	9274	2273	$0.05	253.59	2.91
San Francisco-low load, 24/7, oversize											
Single Max	71824	29	9404	2.1	12417	5190	10153	1364	-	-	-
Dual Max with CV Heating	90422	30	13743	3.0	14974	7513	12711	1363	($0.49)	(1859.87)	(43.39)
Dual Max with VAV Heating	58834	27	7121	2.0	10466	3978	8317	1249	$0.32	1298.95	22.82
Los Angeles Base											
Single Max	59790	40	1478	3.5	11019	967	8283	1836	-	-	-
Dual Max with CV Heating	59185	41	1703	3.9	10943	1095	8177	1866	($0.01)	60.47	(2.25)
Dual Max with VAV Heating	56840	41	1199	4.3	10645	817	7882	1862	$0.05	294.92	2.79
Los Angeles-low load, 24/7, oversize											
Single Max	88824	32	8792	2.1	14853	4859	12515	1438	-	-	-
Dual Max with CV Heating	111111	30	12880	3.0	17902	7046	15584	1419	($0.52)	(2228.72)	(40.87)
Dual Max with VAV Heating	73374	29	6470	1.9	12567	3624	10363	1304	$0.35	1544.96	23.22
Chicago Base											
Single Max	65378	41	1056	3.1	11843	736	9050	1893	-	-	-
Dual Max with CV Heating	63881	42	1083	3.6	11655	754	8837	1918	$0.02	149.67	(0.27)
Dual Max with VAV Heating	62312	42	794	3.8	11465	594	8647	1918	$0.05	306.55	2.62
Chicago-low load, 24/7, oversize											
Single Max	71206	35	11610	3.3	12817	6436	10489	1428	-	-	-
Dual Max with CV Heating	87833	32	16076	3.5	15161	8828	12839	1422	($0.47)	(1662.69)	(44.66)
Dual Max with VAV Heating	61692	32	10262	7.3	11251	5758	9032	1319	$0.22	951.39	13.47
Atlanta Base											
Single Max	71454	51	1690	5.5	13201	1100	10117	2184	-	-	-
Dual Max with CV Heating	71349	51	1899	5.3	13183	1221	10082	2201	($0.01)	10.58	(2.08)
Dual Max with VAV Heating	69506	51	1431	6.1	12960	959	9869	2192	$0.04	194.80	2.59
Atlanta-low load, 24/7, oversize											
Single Max	98589	38	9284	2.5	16825	5169	14325	1600	-	-	-
Dual Max with CV Heating	117814	33	13401	3.3	19414	7392	16965	1549	($0.48)	(1922.46)	(41.16)
Dual Max with VAV Heating	83880	33	7059	3.5	14576	3992	12214	1462	$0.34	1470.95	22.25

Figure 13 through Figure 16 shows the VAV flow control, reheating coil output and fan flow histogram for the worst code compliance case in L.A (highlighted in yellow in Table 3).

Since the system is largely oversized, almost all the hours the airflow stays at minimum flow ratio.

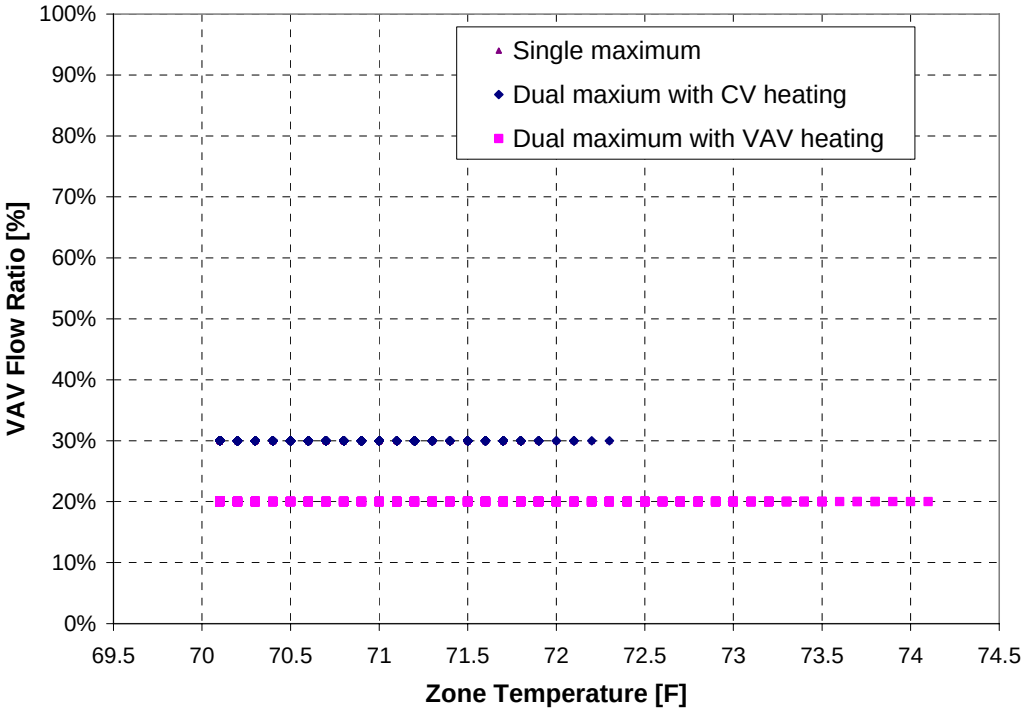


Figure 13 Zone air flow control for worst code compliance case in L.A.

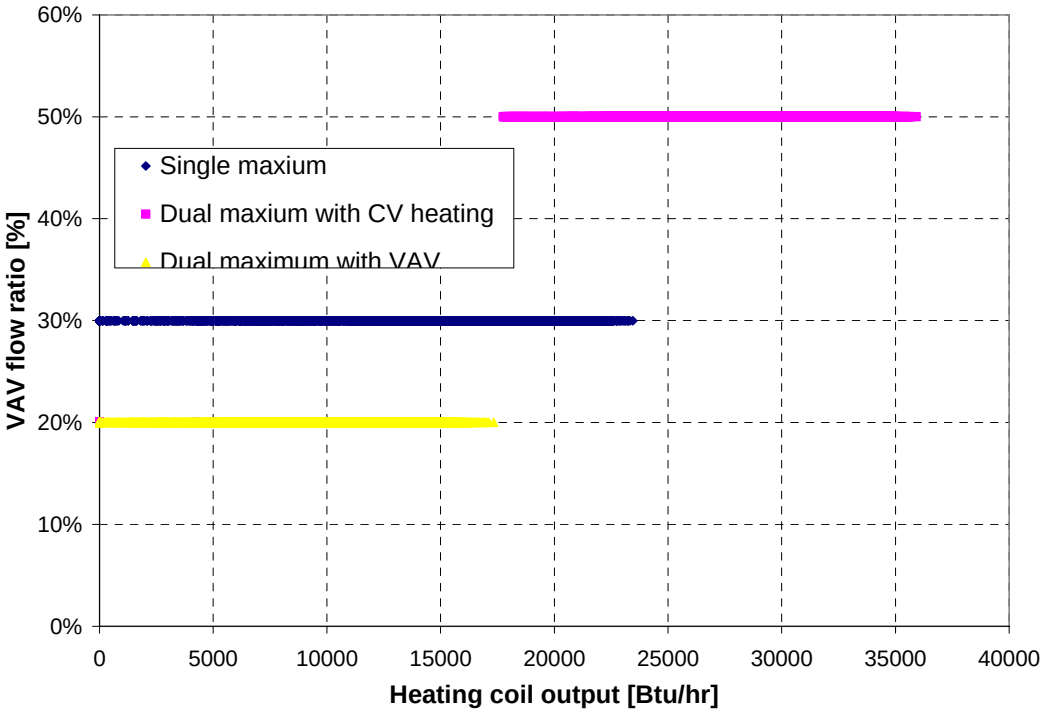


Figure 14 Zone air flow rate and heating coil output for worst code compliance case in L.A.

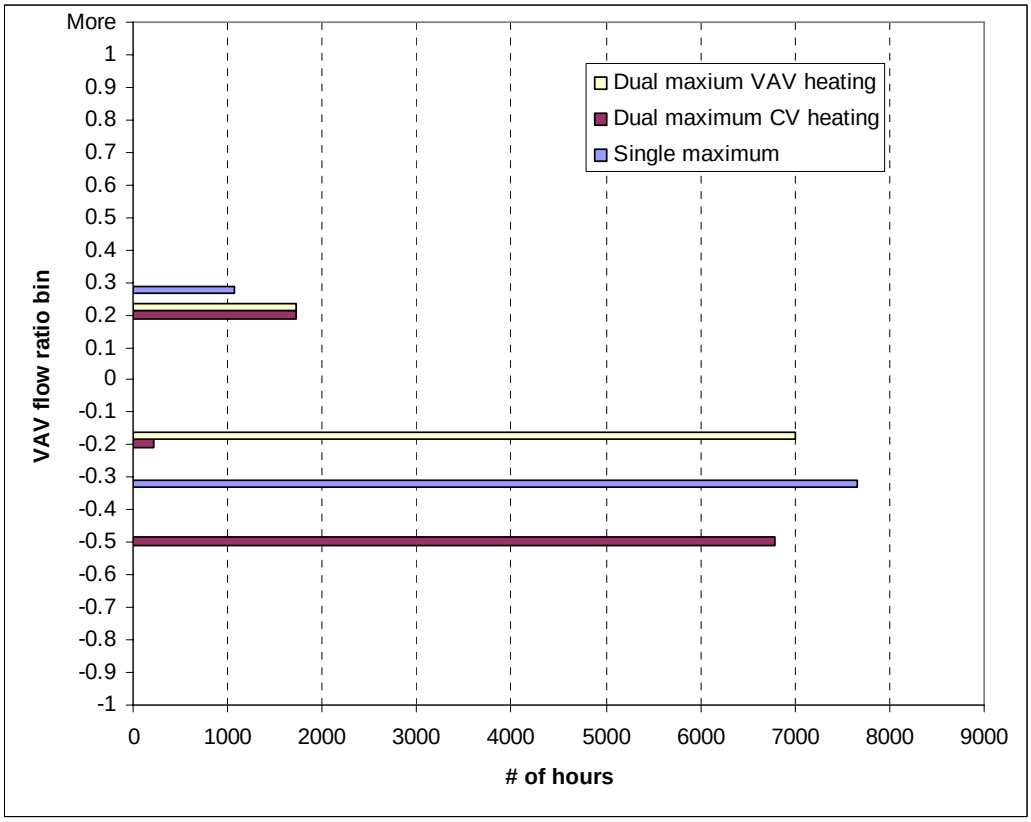


Figure 15 Zone air flow histogram

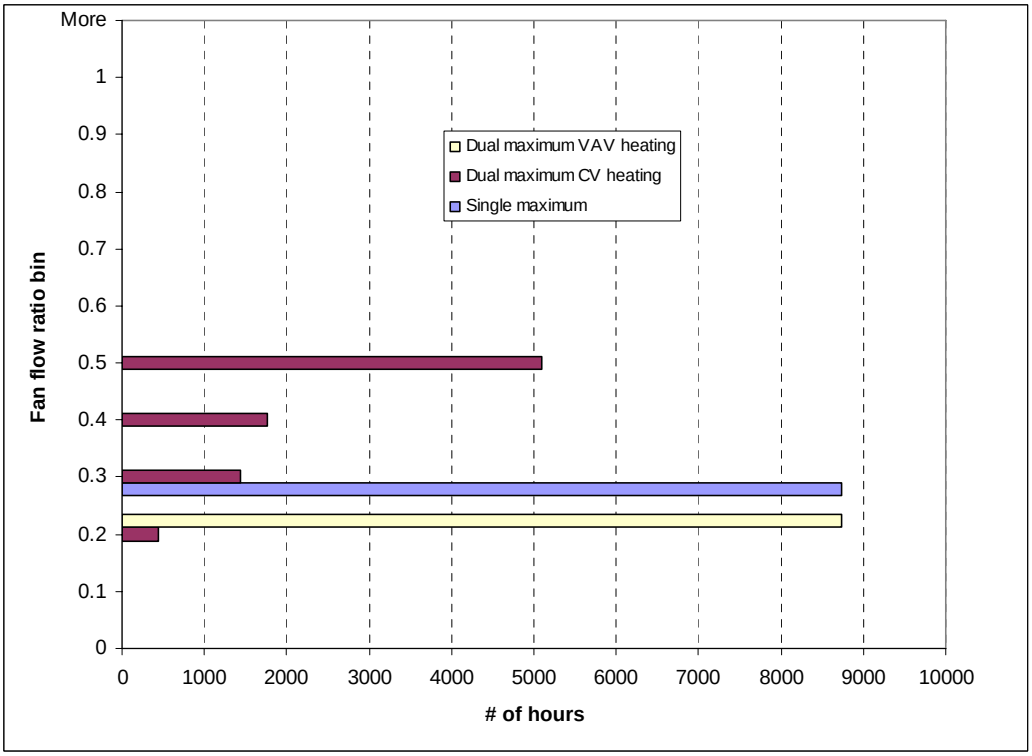


Figure 16 Fan flow histogram

F. Potential Statewide Savings

According to the California Energy Commission (http://www.energy.ca.gov/reports/2000-07-14_200-00-002.PDF) there are approximately 6 billion square feet of existing commercial buildings in California. Of this area, about 2 billion square feet is office and university/college. The office + univ/college sector is expected to add about 50 million square feet every year through the end of the decade. If we assume that half of existing and new buildings in these sectors are VAV systems and that 0.5% of existing VAV systems will be retrofit annually with new lower minimum setpoints and that 20% of new VAV systems will be installed with lower minimum setpoints then the penetration will be about 10 million square feet per year. At an estimated savings of $0.10/ft^2 this comes out to $1 million in energy savings the first year, $2 million the second year, and $10 million per year in year 10. Energy savings are also estimated at 0.5 kWh/ft^2-yr and 0.08 therms/ft^2-yr, which comes out to 5,000,000 kWh and 800,000 therms of energy savings the first year and 50,000,000 kWh and 8,000,000 therms of energy savings in year 10.

	YEAR	2001	2002	2003	2004	2005	2006	2007	2008	2009	2010
	STOCK	369	376	384	391	398	405	411	418	424	431
SML-OFF.	ADDITIONS	11	11	11	11	11	11	11	11	11	11
	STOCK	148	150	152	154	156	158	160	162	164	166
RESTAUR.	ADDITIONS	4	4	4	4	4	4	4	4	4	4
	STOCK AD	896	911	925	939	953	966	978	990	1,003	1,016
RETAIL	DITIONS	23	24	24	24	24	24	23	23	24	25
	STOCK	234	238	242	246	249	253	256	260	263	267
FOODSTR.	ADDITIONS	7	6	7	7	7	7	6	6	7	7
	STOCK	762	779	795	811	826	841	855	870	885	900
NRFGWHSE	ADDITIONS	26	26	25	25	24	24	23	23	24	24
	STOCK	44	45	45	46	47	48	48	49	50	51
REFGWHSE	ADDITIO	1	1	1	1	1	1	1	1	1	1
	STOCK	466	474	482	489	496	503	509	514	519	524
ELEM SCH	ADDITIONS	10	10	10	10	10	10	10	9	9	8
	STOCK	273	277	280	283	286	289	292	295	298	301
UNIV/COL	ADDITIONS	4	5	5	5	5	5	5	5	5	5
	STOCK AD	283	288	293	298	304	309	314	320	325	334
HOSPITAL	DITIONS	5	6	7	6	6	7	6	7	8	10
	STOCK	276	281	286	291	295	300	304	308	313	317
HTL/MTL	ADDITIONS	8	8	8	8	8	8	7	7	8	8
	STOCK	1,008	1,023	1,038	1,053	1,067	1,082	1,095	1,108	1,121	1,134
MISCELL.	ADDITIONS	24	24	25	25	26	26	25	25	26	26
	STOCK	1,047	1,070	1,092	1,113	1,133	1,152	1,171	1,191	1,211	1,231
LRG-OFF.	ADDITIO	32	33	33	32	32	32	32	33	34	35
	STOCK	5,805	5,910	6,013	6,113	6,212	6,306	6,395	6,484	6,576	6,670
TOTAL	ADDITIONS	157	159	159	159	159	157	154	156	160	164

Figure 17. SUMMARY OF CALIFORNIA FLOOR SPACE STOCK PROJECTIONS BY BUILDING TYPE (106 SQ FT). Source: http://www.energy.ca.gov/reports/2000-07-14_200-00-002.PDF**