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Low Pressure Air-Handling System Leakage in Large Commercial Buildings: Diagnosis, Prevalence, and Energy Impacts

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ABSTRACT

Air-handling system leakage reduces the amount of air delivered to conditioned spaces and in most cases wastes energy and money. Standards exist for where and how to measure system airtightness, but they tend to focus on new construction, and only on the high-pressure (1500 Pa to 2500 Pa (6 to 10 in. w.c.))/ medium-pressure (500 to 1500 Pa (2 to 6 in. w.c.)) portions of the system. This paper investigates air leakage in the "low-pressure" (≤500 Pa (≤2 in. w.c.)) portions of large commercial-building air-handling systems (i.e., downstream of variable-air-volume box inlet dampers). A simplified diagnostic protocol for measuring low-pressure leakage that can be used during normal system operation in an existing building is presented and utilized for this investigation. A validation of the protocol using a calibrated leak in a field installation is also presented, as are the results of applying this protocol in nine other buildings around the United States. The validation results indicate that normalized leakage can be measured to within 10 L/s at 25 Pa (20 cfm at 0.1 in w.c.), with and without the existence of significant flow through the minimum opening of the box inlet damper. The field test results indicate that "low-pressure" leakage varied considerably from system to system (standard deviation of 50% of the mean value), and that the average value was approximately 10% of the flow entering the low-pressure system sections. The variability of the measured results, combined with a simplified analysis of the impacts of this leakage, suggest that testing of low-pressure system leakage in commercial buildings should be economically justifiable.

INTRODUCTION

Air-handling systems in large commercial buildings typically have thousands of field-assembled joints between duct sections and duct-mounted equipment and accessories (e.g., VAV boxes, reheat coils, access doors, fire/smoke dampers, grilles), which create numerous opportunities for leakage, in addition to the leakage of the equipment and accessories themselves. Leakage reduces the amount of air delivered to conditioned spaces and in most cases wastes energy and money. Unlike water or other fluid leaks, air leakage cannot be detected by simple observation.

The heating, ventilating, and air-conditioning (HVAC) industry has developed various standards and guidelines for constructing air-handling systems in large commercial buildings to minimize leakage, and also for measuring it. However, the issue of where and how to measure leakage remains controversial. In particular, one standard (SMACNA 2012) suggests that it is not cost-effective to measure leakage from "low-pressure" system sections, while other industry handbooks indicate otherwise (ASHRAE 2012, 2013). Historically, most leakage testing has been performed in new construction, and has focused on the "main" ducts (i.e., the high-pressure/medium-pressure sections upstream of variable-air-volume (VAV) boxes that are used to control the flow and temperature of air being delivered to conditioned spaces in a building). There are, however, many system sections that operate at lower pressures, such as the ones downstream of VAV box inlet dampers, as well as toilet exhaust systems and kitchen exhaust systems. For example, some VAV systems have as much as 50 to 75% of the system (based on surface area) located downstream of the inlet dampers (Fisk et al. 1999).

This paper investigates air-handling system leakage in the "low-pressure" portions of large commercial building supply air systems that are located downstream of VAV box inlet dampers. A diagnostic protocol that can be used during normal system operation in an existing building to determine this leakage is described and utilized for this investigation. A validation of the protocol based upon adding a calibrated leak in a field installation is presented, as are the results of applying this protocol in various buildings around the United States. Energy impacts of low-pressure leakage are also briefly discussed.

Air Handling System Leakage

Air-handling system leakage is generally expressed in one of two ways: a) as the flow through all of the leaks in a section of interest at some specified pressure difference (often normalized by the surface area of that section), or b) as the leakage flow as a fraction (or percentage) of the flow entering the section during "normal" operation.

The first metric actually represents the aggregate effective flow resistance of all the leaks in the section of interest (e.g., effective leakage area). The advantage of this metric is that it represents a physical characteristic that does not depend on how the system is operated. However, because system pressures vary during normal operation, it is difficult to translate leakage area into leakage flow during normal operation, which is what is needed to calculate the energy impacts or comfort/performance implications of those leaks.

To use the second metric (percentage leakage), one needs to understand how system pressures and flows vary during normal operation, particularly for variable-flow systems, and a "normal" operating state must be defined for benchmarking purposes. For example, in a typical VAV system, each VAV box has one or more dampers that change their position over time based upon the thermostat call for cooling or heating in the zone being served, thereby changing the flow entering the system section controlled by the box, and also the pressure differences across the box walls and across the downstream ductwork.

In the "main" duct upstream of the VAV box inlet dampers, system pressure is nominally maintained constant by a supply-fan control loop that senses the pressure at least at one point in that ductwork (typically about two-thirds of the way down the main duct)[1]. This suggests that leakage flow is constant and that percentage leakage will vary as the system flow is varied in response to varying heating and cooling demands. However, leakage flow can change in these sections, because there is flow resistance throughout the "main" duct due to friction and fitting losses, such that the pressure is not constant along the entire duct section upstream of the VAV boxes, particularly at higher airflows. Systems with higher flow resistance will have more variation in pressure throughout the main duct. On average, leaks upstream of the VAV box inlet dampers experience significantly larger pressure differentials as compared to leaks downstream of those dampers, which is why construction standards have focused on the upstream leaks.

Downstream of VAV box inlet dampers, the opposite occurs: leakage fractions are relatively constant and leakage flows vary. One way to understand this behavior is to express both the downstream-section flow and the flow through the leaks as functions of pressure differences downstream of the damper. If one considers all of the resistance of the ductwork to be due to turbulent friction and fitting-related kinetic energy losses, longitudinal pressure loss along the ductwork scales with the square of the velocity, and therefore with the airflow squared. Conversely, the airflow is proportional to the square root of the pressure difference along the length of the downstream ductwork (exponent of 0.5). For the leakage, flows through leaks are commonly assumed to vary with the pressure difference across the section walls raised to the power 0.65 (ASHRAE 2012 page 19.4, Equation 1). The average pressure exponent reported in this paper is 0.62 ± 0.11. Thus, the fractional flow through the leaks downstream of a VAV box inlet damper (leakage-flow/VAV-inlet-flow) scales with the wall pressure difference raised to the power 0.15:

$$\Delta P^{0.65}/\Delta P^{0.5} \Rightarrow \Delta P^{0.15} \tag{1}$$

It should be noted however that, just like upstream of the VAV box inlet damper, the pressure differential across the leaks is not uniform throughout the downstream section, but rather is a function of where the leaks are located relative to the damper.

Not surprisingly, both upstream and downstream of VAV boxes, the leaks closest to the fan experience higher pressures relative to leaks further away from the fan. Therefore, the distribution of leaks can also influence the leakage flow. If the flow through leaks varied linearly with pressure, using the average pressure would address this problem. However, leakage flow

[1] Note that some systems employ a "duct static pressure reset" strategy, which involves changing the pressure set point so that one or more VAV box inlet dampers are as wide open as possible.

generally varies with the pressure differential raised to a power close to 0.65. Thus, the distribution of leak locations influences the fractional leakage flow during normal operation. The impact of this non-linearity is generally higher downstream of VAV box inlet dampers, as the fractional difference in the pressure differential over the length of that system section is larger.

Measuring System Leakage

The standard technique for measuring system leakage during construction is fan pressurization. This technique involves isolating a particular section of the system, attaching a fan with an inline orifice flow meter to it (accuracy typically about 3% or better), and measuring the airflow required to maintain a target pressure in that section. One issue associated with this technique is that the reference pressure (often chosen based upon the ductwork construction “class”) may or may not be representative of the actual pressures across leaks during normal operation, and therefore the measured leakage flow often cannot be used directly for energy or performance analyses. The uncertainty of leakage flows estimated using this method mostly stems from the uncertainty associated with estimating the pressure differences across the leaks at operating conditions. Typically, this technique is only utilized for ductwork upstream of VAV box inlet dampers. In the authors’ experience, a disadvantage of applying fan pressurization downstream of VAV box inlet dampers is that one must physically isolate the downstream section from the upstream sections. This is certainly possible to do, however the authors have found it to be challenging and disruptive in existing buildings.

The second most-common way to determine leakage in systems is as part of test and balancing (TAB) activities, where the flow measured at the supply fan inlet (or entering a particular system section) is compared with the sum of the supply-grille exit flows. The advantage of this method is that it measures operating flows directly, although a particular test condition must be specified for reference. In the authors’ experience, the key downside to using this method for measuring leakage downstream of VAV box inlet dampers is the difficulty associated with accurately measuring the flow through the damper (i.e., the flow entering the system section being tested). Inlet flow grid accuracy is typically not good enough for using subtraction to determine the leakage flow (which is a modest fraction of the total flow entering the test section). Also, two different instruments are usually required to measure the flow entering the test section and all the grille exit flows, such that any bias errors between the two instruments do not necessarily cancel out, and precision errors add in quadrature.

MODIFIED FAN PRESSURIZATION PROTOCOL FOR MEASURING LEAKAGE DOWNSTREAM OF VAV BOX INLET DAMPERS

Modera (2007) described a new technique for determining leakage area downstream of VAV box inlet dampers. It is based on a modified fan pressurization protocol that can be performed with the HVAC system operating. This protocol also determines the leakage area of the VAV box inlet damper in the closed (minimum) position (or at the minimum-flow-setting).

The test set-up involves preparing the section to be tested by first turning the VAV box inlet damper to its minimum position (either using the associated control system or manually), and then blocking off all downstream supply-air diffusers (outlets) except for one. The unsealed diffuser is fitted with a collar that connects the diffuser to a portable fan by means of flexible duct. The particular fan utilized for this paper was a commercially-available product that is integrated with a calibrated flow meter that determines flow based upon a measured pressure differential between the fan inlet and a downstream pressure tap located on the fan motor ("flow pressure"). One example application of the process is shown in Figure 1.

Figure 1: Example field application of modified fan pressurization protocol. VAV box inlet damper is being closed manually on the left, and the pressurization fan with integrated flow meter is attached to a supply diffuser two different ways in the two pictures on the right.

The test procedure (described in more detail below) consists of adjusting the speed of the portable fan, while the fan system that supplies air to the VAV box is also operating, so as to produce a series of different pressures inside the duct section being tested ("duct pressure"), and simultaneously measuring the "flow pressure" and "duct pressure". The procedure also includes making several other one-time pressure measurements with the VAV damper fully open: the static pressure in the ductwork upstream of the VAV box inlet damper, the static pressure downstream of the inlet damper, and the static pressure at each diffuser. All of the measured pressure data are currently post-processed using a spreadsheet program that calculates leakage area, which could be replaced in the future with dedicated hardware/firmware, or by a custom software program and Graphical User Interface (GUI) to provide real-time results.

More specifically, the test setup and procedure consists of the following seven steps:

1) With the inlet damper in the fully open position, measure the normal operating static pressure upstream and downstream of the VAV box inlet damper, and at each supply diffuser (outlet).
2) Close the inlet damper to its minimum setting (either by means of the control system or manually).
3) Measure the static pressure upstream of the inlet damper with the damper in its minimum position.
4) Block all downstream diffusers (outlets) for the section of interest, except for one, and connect a pressurization/depressurization fan and flow meter device to the unblocked grille.
5) Measure the flow through the pressurization/depressurization fan and flow meter device and the static pressure in the system section downstream of the VAV box inlet damper at several pressurization and depressurization conditions for this section (at least three to four points, nominally 10 Pa (0.04 in w.c.) apart).
6) Perform an iterative fit to the data, solving simultaneously for the section-leakage flow coefficient, the section-leakage pressure exponent, and the leakage area of the VAV box inlet damper in its minimum position (further details below).
7) Calculate the percentage leakage based upon the average of the measured pressures downstream of the VAV box inlet damper and at the diffusers (outlets) with the VAV damper wide open, the nominal or measured flow through the VAV box, and the calculated section leakage flow coefficient and pressure exponent.

Step 6 of the procedure involves an iterative linear-regression solution to the conservation of mass equation (with density assumed to be constant):

$$Q_{damper} + Q_{fan} + Q_{leaks} = 0 \tag{2}$$

where

Q_{damper} is the flow through the VAV box inlet damper,
Q_{fan} is the flow through the portable fan, and
Q_{leaks} is the flow through the leaks downstream of the box inlet damper.

The analysis requires careful attention to the signs of the various flows under different test conditions. There are three different test modes: 1) when the portable fan is *off*, the pressure in the duct section being tested is larger than the surroundings pressure, so the flow through the inlet damper is split between the flow exiting through leaks downstream of the damper and the flow through the portable fan; 2) when the portable fan is blowing air into the duct section downstream of the damper (*pressurizing* the section being tested), the damper flow is leaving only through the leaks; and 3) when the portable fan is *depressurizing* the section being tested, enough so that the flow through the leaks is reversed and enters from the surroundings, all of the airflow through the VAV damper and through the leaks is going out through the portable fan. Mode 3 is not necessarily required, however it can be used to identify one-way-valve effects in the leaks (e.g. lifting within diffusers).

In Mode 1 when the portable fan is *off*, substituting a power-law model (based on a flow coefficient and pressure exponent) for leaks downstream of the box inlet damper into Equation 2, rearranging, and assuming one knows the flow through the inlet damper, Equation 3 can be used to solve for the test section leakage flow:

$$Q_{leaks} = K_{leaks}(P_{duct})^n = Q_{damper} - Q_{fan,off,out} \quad (3)$$

where

K_{leaks} is the flow coefficient for all of the leaks combined downstream of the damper,
n is the pressure coefficient for these combined leaks,
P_{duct} is the average static pressure difference across these combined leaks, and
$Q_{fan,off,out}$ is the flow *exiting* the duct section through the portable fan when it is *off*.

Also, assuming that the damper can be modelled as an orifice:

$$Q_{damper} = A_{damper}\sqrt{\frac{2\left(P_{upstream} - P_{duct}\right)}{\rho}} \quad (4)$$

where

A_{damper} is the leakage area of the inlet damper opening,
$P_{upstream}$ is the static pressure inside the ductwork upstream of the inlet damper, and
ρ is the density of the air entering through the damper opening.

So, substituting Equation 4 into Equation 3:

$$K_{leaks}\left(P_{duct}\right)^{n}=A_{damper}\sqrt{\frac{2\left(P_{upstream}-P_{duct}\right)}{\rho}}-Q_{fan,off,out} \tag{5}$$

In <u>Mode 2,</u> when the portable fan is *pressurizing* the test section:

$$K_{leaks}\left(P_{duct}\right)^{n}=A_{damper}\sqrt{\frac{2\left(P_{upstream}-P_{duct}\right)}{\rho}}+Q_{fan,on,in} \tag{6}$$

where

$Q_{fan,on,in}$ is the flow *entering* the duct section through the portable fan.

Taking the natural logarithm of both sides of Equation 6:

$$\ln\left(K_{leaks}\right)+n\ln\left(P_{duct}\right)=\ln\left(A_{damper}\sqrt{\frac{2\left(P_{upstream}-P_{duct}\right)}{\rho}}+Q_{fan,on,in}\right) \tag{7}$$

In <u>Mode 3</u> when the portable fan is *depressurizing* the test section:

$$K_{leaks}\left(P_{duct}\right)^{n}=Q_{fan,on,out}-A_{damper}\sqrt{\frac{2\left(P_{upstream}-P_{duct}\right)}{\rho}} \tag{8}$$

where

$Q_{fan,on,out}$ is the flow *exiting* the duct section through the portable fan.

Taking the natural logarithm of both sides of Equation 8:

$$\ln\left(K_{leaks}\right)+n\ln\left(P_{duct}\right)=\ln\left(Q_{fan,on,out}-A_{damper}\sqrt{\frac{2\left(P_{upstream}-P_{duct}\right)}{\rho}}\right) \tag{9}$$

The final solution is obtained by iterating using guesses for A_{damper} until the leakage flow parameters K_{leaks} and *n* satisfy both Equation 5, and linear regression fits to Equations 7 or 9. Note that the linear regressions determine K_{leaks} and *n* for an assumed value of A_{damper}, based upon the series of measurements in Modes 2 or 3 at different duct pressures.

RESULTS: FIELD VALIDATION OF MODIFIED FAN PRESSURIZATION PROTOCOL

To test the performance of the new protocol for measuring leakage downstream of VAV box inlet dampers, a field test was performed on one branch of a single-duct VAV system in a large office building in Sacramento, California. The section tested is illustrated in Figure 2. It had five supply grilles downstream of the VAV box, with a design cooling maximum flow of 684 L/s (1,450 cfm) equally distributed to each grille. One of the grilles was fitted with a calibrated fan pressurization device that has a rated accuracy of 3% or 0.5 L/s (1 cfm), whichever is greater. The other four grilles were sealed with tape from inside the room. The box's circular inlet damper was set to two different positions for test purposes: a) its normal minimum position, and b) as tight as it could be turned manually. The protocol was applied to measure the leakage of the section downstream of the inlet damper in both positions, and using pressurization and depressurization. After adding a known leak to the test section (at the end of the plenum downstream of the box), the procedure was repeated, again using the protocol to measure the leakage at both inlet damper positions, with both pressurization and depressurization. Pressures were measured using an auto-zeroing two-channel digital manometer with a rated accuracy of 1% or 0.15 Pa (0.0006 in.w.c.), whichever is greater.

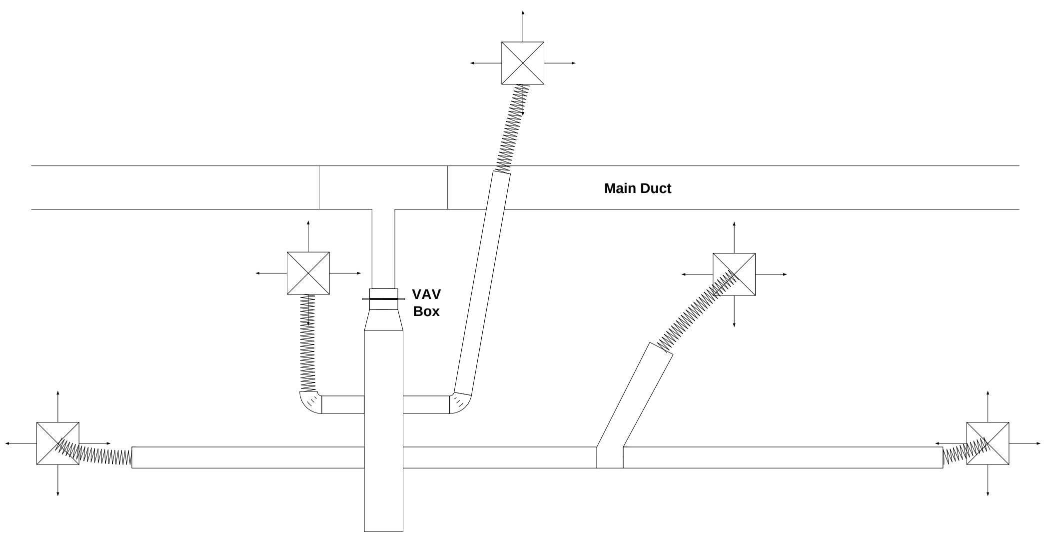

Figure 2: System section in Sacramento office building that was used for validating modified fan pressurization protocol.

The leak that we added to the test section was a perforated plate that had been calibrated in the laboratory. Its flow can be represented by the following equation:

$$Q = K_{leak}\,\Delta P^{\,0.52} \tag{10}$$

where:

Q = perforated-plate leakage flow, L/s (cfm),
K_{leak} = 11.75 L/s per $Pa^{0.52}$ (443 cfm per in.w.c.$^{0.52}$), and
ΔP = pressure drop across perforated plate, Pa (in.w.c.).

By measuring the test section leakage before and after this plate was added and by subtracting the results with and without the plate installed, we could determine the normalized leakage (i.e., leakage area) of the added leak. A sample analysis of one test in the Sacramento office building using Mode 1 and 2 data is presented in Table 1, a summary of the Sacramento office-building results is presented in Table 2, and the normalized measured leakage for each test is presented in Figure 3.

Table 1: Sample data/analysis for Sacramento office building – pressurization data for leaky duct and leaky damper. A_{damper} was adjusted until the fit to Eq. 7 also satisfied Eq. 5.

Flow Mode	P_{duct} [Pa (in.w.c.)]	$P_{upstream}$-P_{duct} [Pa (in.w.c.)]	Q_{fan} [L/s (cfm)]	Q_{damper} [L/s (cfm)]	Q_{leaks} [L/s (cfm)]
1	18 (0.07)	94 (0.38)	-31 (-66)	102 (216)	71(150)
2	48 (0.19)	64 (0.26)	37 (79)	84 (178)	121 (257)
2	53 (0.21)	59 (0.24)	47 (100)	81 (171)	128 (271)
2	61 (0.25)	51 (0.20)	62 (132)	75 (159)	138 (291)
2	67 (0.27)	45 (0.18)	75 (158)	71 (150)	145 (308)
2	75 (0.30)	37 (0.15)	91 (194)	64 (135)	155 (329)
K_{leaks}	14.5 L/s/Pa$^{0.548}$ (633 cfm/(in w.c.)$^{0.548}$)				
n	0.55 [-]				
A_{damper}	82 (13) [cm^2 (in^2)]				

Table 2: Measurement results from Sacramento office building.

Inlet Damper Position	Section Tested	Fan-Induced Section Pressures [Pa (in.w.c.)]	Section Leakage Exponent	Calculated Section Leakage [L/s at 25 Pa (cfm at 0.1 in.w.c.)]	Calculated Damper Leakage [L/s at 25 Pa (cfm at 0.1 in.w.c.)]
Tight	As-is	8 to 64 (0.03 to 0.26)	0.61	27 (57)	0.1 (0.2)
Tight	As-is	-7 to -68 (-0.03 to -0.27)	0.61	28 (59)	0.1 (0.2)
Tight	w/leak	1 to 13 (0.004 to 0.05) 4 to 55 (0.02 to 0.22)	0.51 0.53	91 (192) 91 (193)	0.3 (0.6)
Tight	w/leak	-4 to -15 (-0.02 to -0.06) -7 to -59 (-0.03 to -0.24)	0.58 0.55	91 (193) 88 (187)	0.2 (0.5)
Leaky	As-is	29 to 64 (0.12 to 0.26)	0.52	33 (70)	59 (124)
Leaky	As-is	1 to -14 (0.004 to -0.06)	0.5	30 (64)	56 (119)
Leaky	w/leak	48 to 75 (0.19 to 0.30)	0.55	85 (179)	53 (112)
Leaky	w/leak	-1 to -25 (-0.004 to -0.10)	0.49	88 (186)	55 (116)
AVERAGE	As-is	n/a	0.56	30 (63)	n/a
AVERAGE	w/leak	n/a	0.53	89 (189)	n/a
Tight	**AVERAGE**	n/a	0.58	n/a	0.2 (0.4)
Leaky	**AVERAGE**	n/a	0.52	n/a	56 (118)

The results in Figure 3 suggest a number of interesting conclusions. Looking first at the results for the test section in the "as-is" leakage condition, these results indicate that the measured leakage of the test section in the as-is condition was consistent over each group of four tests: 1) *pressurization* with a fully closed ("*tight*") VAV box inlet damper, 2) *depressurization* with a *tight* damper, 3) *pressurization* with a minimum-setting (*"leaky*") damper, and 4) *depressurization* with a *leaky* damper. The agreement between pressurization and depressurization results suggests that there were no “one-way-valve” leaks in this test section, which in some cases might be caused by the grilles lifting off the T-bar ceiling during pressurization. There was one point at 93 Pa (0.37 in.w.c.) of pressurization that showed a disproportionately large leakage flow, which was not included in the analysis for Figure 3. This point, which increased the flow exponent from 0.55 to 0.66, does suggest some lifting of the grilles (or “one-way-valve” effects) at that pressure.

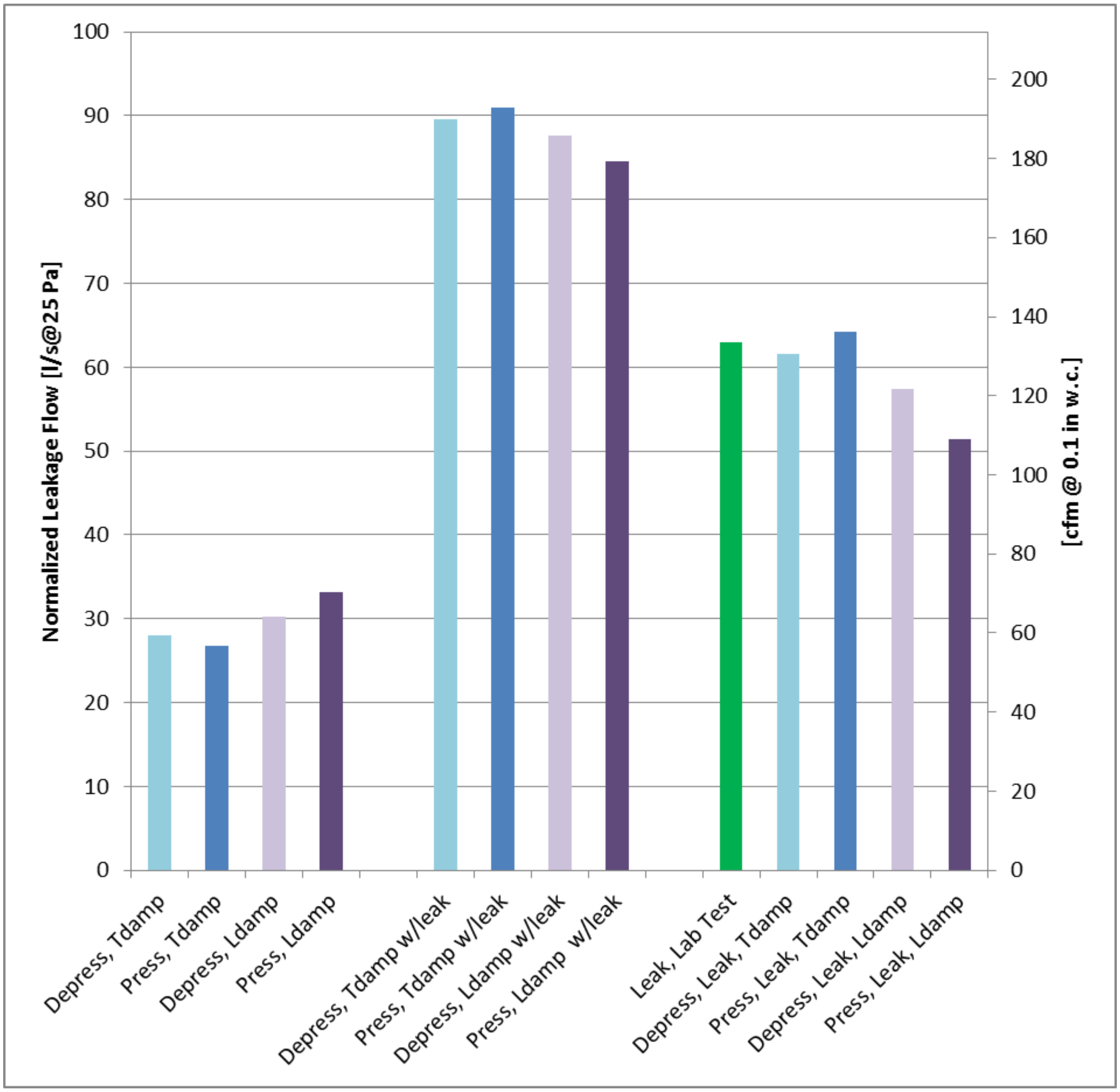

Figure 3: Section leakage determined using the modified fan pressurization protocol under different conditions: pressurization (Press) and depressurization (Depress), tight (Tdamp) and leaky (Ldamp) VAV box inlet damper, with and without added leak.

The fact that the results were not affected by a significant change in inlet damper leakage suggests that the protocol is able to appropriately handle damper leakage. In this case, the tight-damper scenario was essentially airtight, which means that the modified fan pressurization test was effectively equivalent to a standard fan pressurization test (i.e., block all openings and measure the flow required to maintain a given pressure). The leaky-damper scenario corresponded to damper leakage roughly twice as large as the section leakage in the as-is condition, 56 L/s at 25 Pa (118 cfm at 0.1 in.w.c.) versus 30 L/s at 25 Pa (63 cfm at 0.1 in.w.c.).

Turning to the second set of results in Figure 3, the same tests were performed on the same test section, except in this case with the addition of a perforated plate whose flow as a function of pressure difference follows Equation 10. The normalized leakage of the perforated plate is roughly the same magnitude as the average damper leakage in the leaky-damper case (63 L/s at 25 Pa (133 cfm at 0.1 in.w.c.) versus 56 L/s at 25 Pa (118 cfm at 0.1 in.w.c.)). The results in Figure 3 indicate that the measured leakage of the test section plus the perforated plate ranged between 85 L/s at 25 Pa (179 cfm at 0.1 in.w.c.) and 91 L/s at 25 Pa (193 cfm at 0.1 in.w.c.).

Turning to the third set of results in Figure 3, the leakage for the perforated plate based on the "lab test" (Equation 10) is 63 L/s at 25 Pa (133 cfm at 0.1 in.w.c.). The same leakage calculated as the difference between each pair of field tests (pressurization: with and without the plate; depressurization: with and without the plate; both with and without a leaky VAV damper) ranged from 51 L/s at 25 Pa (109 cfm at 0.1 in.w.c.) to 64 L/s at 25 Pa (136 cfm at 0.1 in.w.c.). The RMS error for all four field test points was 10%, with the average value being 59 L/s at 25Pa (124 cfm at 0.1 in.w.c.), or 4 L/s at 25Pa (8 cfm at 0.1 in.w.c.) (7%) low compared to the reference. Relative to the design inlet flow for the test section 684 L/s (1448 cfm), the reference leakage percentage for the added leak is 9.2% and the average field test value is 8.6%. For practical purposes, these values are identical and we can conclude that the new technique was able to correctly determine the leakage of the added leak.

RESULTS: FIELD APPLICATION OF MODIFIED FAN PRESSURIZATION PROTOCOL

The modified fan pressurization protocol also was used to measure leakage downstream of VAV box inlet dampers at nine other buildings distributed around the United States. In particular, the protocol was used to determine whether the low-pressure sections of the air-handling systems in these particular buildings were worth sealing. The buildings were constructed during the 1980s and 1990s, and are located on university campuses and military bases. .However, although the choice of buildings to test was not strictly random, there was no prior knowledge of leakage levels in the buildings. Table 3 presents the measurement results.

The first thing to note in Table 3 is that the leakage is presented as percentage flow, yet the modified fan pressurization protocol measures leakage as a function of the applied pressure differential, which is not necessarily operating pressure. However, because the flow exponent of the leakage was also measured, leakage flows can be calculated at other pressures. Thus, the leakage percentages in Table 3 were determined from calculated leakage flows combined with either design flows or flows measured at the grilles. The leakage flows were calculated by measuring the pressures at the grilles and directly downstream of the VAV box inlet dampers under fully-open operation. Best estimates of the average pressures seen by the leaks were determined from these measurements.

The flow entering the VAV box was based upon design flows for all of the buildings in Table 3 other than for Buildings 3, 6, and 7. The flows for those three buildings were determined by measuring and summing the flows through each grille at fully-open VAV-damper position, and adding the calculated leakage flow to this sum. The grille flows were measured using a powered flow capture hood, which is based on the same portable fan pressurization device used for the leakage measurements. In particular, to measure each grille flow, the calibrated fan was connected to a capture hood that covered the grille, and then the fan was controlled to zero out the static pressure difference between the hood interior and the room. The pressure used to calculate leakage flows is the “Estimated Average Pressure at Leaks”, which is the average of the grille pressure and the pressure downstream of the VAV box inlet damper. Per the analysis presented in Equation 1, the percentage leakage flows in Table 3 should be valid over a full range of operating conditions (varying with the pressure to the power 0.15).

Table 3: Leakage measured downstream of VAV box inlet dampers using modified fan pressurization protocol

Building	State	Year of Const.	Flow Exponent	Upstream Pressure [Pa (in w.c.)]	Minimum Leak Pressure (at grilles) [Pa (in w.c.)]	Estimated Average Pressure at Leaks [Pa (in w.c.)]	Best Estimate Percentage Leakage
1	CA	1988	0.80	250 (1.0)	8 (0.03)	25 (0.1)	8
2	WA	1997	0.72	375 (1.5)	10 (0.04)	25 (0.1)	15
3	RI	1988	0.54	300 (1.2)	20 (0.08)	25 (0.1)	13
4	RI	1988	0.64	108 (0.43)	15 (0.06)	25 (0.1)	11
5	FL	1988	0.61	550 (2.2)	50 (0.20)	50 (0.2)	19
6	TX	n/a	0.61	155 (0.62)	10 (0.04)	40 (0.16)	6
7	TX	1995	0.78	155 (0.62)	10 (0.04)	40 (0.16)	4
8	CA	n/a	0.53	375 (1.5)	10 (0.04)	67 (0.27)	9
9a	CA	1996	0.51	488 (2.0)	20 (0.08)	50 (0.2)	6
9b	CA	1996	0.51	488 (2.0)	20 (0.08)	50 (0.2)	13
Average			**0.62**	**324 (1.3)**	**17 (0.07)**	**40 (0.16)**	**10%**
Standard Dev. [%]			**17**	**48**	**73**	**37**	**45**
Std Error in Mean [%]			**6**	**15**	**23**	**12**	**14**

Notes: Buildings 9a and 9b represent two different VAV boxes in the same building.

There are a number of observations that can be made based upon the results in Table 3. One observation is that the average leakage downstream of VAV box inlet dampers in these buildings is substantial (10% of the VAV box inlet flow). Moreover, although not shown in Table 3, the standard error in this mean value is 1.5 percentage points, suggesting that the probability of the true mean value being between 8.5 and 11.5% is about two thirds.

Another observation is that the standard deviation between buildings is also substantial (roughly ±5 percentage points or almost 50% of the mean value). Such a large deviation suggests that leakage testing is needed to determine whether a particular system requires sealing.

Table 3 also illustrates the large difference in the pressures experienced by the leaks upstream and downstream of VAV box inlet dampers. In addition, Table 3 also indicates that leaks downstream of these dampers are exposed to a large range of pressures, which increases the uncertainty in estimating the leakage flow downstream of these dampers (the non-linearity of the flow through those leaks means that the average pressure seen by the leaks may or may not provide a good estimate of the leakage flow).

All of the tests presented in Table 3 were performed in pressurization mode only, so it is difficult to know whether there was any pressurization-induced lifting of grilles off the T-bar ceiling. However, an examination of the results obtained by removing the highest pressure point(s) from the analysis was used to investigate whether and how the highest pressure point(s) impacted the measured flow exponent. In addition, one system section was tested in depressurization mode. These results are shown in Table 4.

Table 4: Impact of Test Pressure on Flow Exponent

Building/ System	Full Test Pressure Range [Pa] (in.w.c.)	Flow Exponent [-]	Lower Test Pressure Range [Pa] (in.w.c.)	Flow Exponent [-]	Change in Flow Exponent [-]	Notes
1	72 to 123 (0.29 to 0.49)	0.80				only 3 points
2	12 to 62 (0.05 to 0.25)	0.72	12 to 27 (0.05 to 0.11)	0.67	-0.06	big pressure change
3	42 to 92 (0.17 to 0.37)	0.54	42 to 75 (0.17 to 0.30)	0.49	-0.05	1 point removed
4	31 to 82 (0.12 to 0.33)	0.64	31 to 53 (0.12 to 0.21)	0.61	-0.03	big pressure change
5	12 to 137 (0.05 to 0.55)	0.61	12 to 44 (0.05 to 0.18)	0.61	0.00	slot diffusers
6	16 to 58 (0.06 to 0.23)	0.61	16 to 42 (0.06 to 0.17)	0.61	0.00	1 point removed
7	19 to 56 (0.08 to 0.22)	0.78	19 to 43 (0.08 to 0.17)	0.67	-0.11	depressurization produced n=0.63
8	10 to 40 (0.04 to 0.16)	0.53	10 to 30 (0.04 to 0.12)	0.53	0.00	all low pressure
9a	25 to 65 (0.10 to 0.26)	0.51	25 to 50 (0.10 to 0.20)	0.50	-0.01	1 point removed
9b	27 to 50 (0.11 to 0.20)	0.51				only 3 points
Average		**0.62**		**0.59**	**-0.032**	
Standard Deviation [%]		**17%**		**12%**	**117%**	
Standard Error in Mean [%]		**6%**		**4%**	**37%**	

Table 4 shows that the flow exponent either remained the same or decreased when the higher pressure points were removed from the analysis. This means that the higher-pressure points increased the flow exponent, which can also be interpreted as the measured flows at the higher pressures being higher than that predicted by measurements at lower pressures. This in turn suggests that the leakage area was increased by the higher pressures, which is consistent with some pressurization-induced lifting of the grilles. It is worth noting that for Building 5, which had slot diffusers, the flow exponent was not impacted by pressure, which would be expected for these diffusers. On the other hand, for Building 7, the depressurization test shows an even lower flow exponent, providing further evidence of lifting for this building. It is also worth noting that the calculated leakage flows at operating pressures generally increased when using the testing based on low pressures only. This is because operating pressures are generally lower than the test pressures, and therefore smaller flow exponents result in higher flows at operating pressures. Based upon these results, it appears that keeping the pressurization below 40 Pa (0.16 in w.c.), and measuring in both pressurization and depressurization would improve the accuracy of the test protocol.

In addition to determining the leakage downstream of the VAV box inlet damper, the test procedure also produced the VAV box inlet damper leakage results presented in Table 5. The results in Table 5 indicate that the damper opening ranged from airtight to almost as large as the leaks in the downstream system section, with the flow through that minimum opening being as much as 23% of the design flow.

Table 5: VAV Box Characterization

Building/ System	Normal-Operation Box Flow [L/s (cfm)]	Minimum VAV Damper Opening [cm^2 (in^2)]	VAV Damper Opening [% of downstream leakage area]	VAV Damper Minimum Flow [% of Design Flow]	Normal-Operation Flow Determination
1	326 (690)	60 (9.3)	88%	23%	plans
2	262 (555)	9.9 (1.5)	10%	6%	plans
3	300 (635)	36.3 (5.6)	33%	16%	Grilles measured with fan-powered hood
4	198 (420)	22.7 (3.5)	41%	11%	plans
5	372 (787)	6.4 (1.0)	5%	3%	plans
6	1096 (2320)	27 (4.2)	16%	2%	Grilles measured with fan-powered hood
7	756 (1600)	0	0%	0%	Grilles measured with fan-powered hood
8	360 (763)	3.1 (0.5)	4%	1%	plans
9a	444 (941)	2.5 (0.4)	5%	1%	plans
9b	475 (1006)	4.0 (0.6)	4%	1%	plans
Average	**459 (972)**	**17 (2.7)**	**21%**	**6%**	
Standard Deviation [%]	**59%**	**113%**	**133%**	**118%**	
Standard Error in Mean [%]	**19%**	**36%**	**42%**	**37%**	

DISCUSSION

The modified fan pressurization protocol described herein could have a number of applications, typically in existing buildings. One scenario is to use it as a debugging tool for when an operating HVAC system fails a performance test (e.g., measured diffuser flows do not match specifications). The protocol could be used to determine if leakage from any portion of the system downstream of an individual VAV box is the cause. Another potential application would be to test for excessive leakage through closed VAV terminal unit dampers when zones are not in use during normal HVAC system operation. Finally, as noted above, the protocol has been used to determine if there is enough leakage downstream of VAV dampers to merit corrective action (e.g., sealing leaks).

Energy and Cost Impacts of Post-VAV Leakage

Although 10% downstream leakage is likely to impact system performance and system capacity, the decisions as to whether it makes economic sense in a particular building to test for this leakage or to reduce it by sealing are often also based on the energy and cost impacts of that leakage. Leakage downstream of VAV box inlet dampers increases supply fan power and adds cooling load due to that extra fan power. If there is a fixed-size aperture used for ventilation air, outdoor air being introduced into the building for ventilation will increase whenever the supply airflow is increased, and in turn heating and cooling loads will increase due to the increased ventilation. This energy impact of ventilation air is increased considerably in 100% outdoor air applications, such as hospitals and laboratory buildings. However, downstream leakage can also decrease VAV box fan operation[2] and the need for reheat. Leakage also changes the temperature of the ceiling plenum, which in turn affects zone thermal loads, both positively and negatively.

Although a detailed analysis of the combined effects of downstream leakage on energy use (e.g., Franconi et al. 1998, Wray and Matson 2003) is beyond the scope of this paper, one can use simple calculations to quantify at least some of the effects.

The most straightforward calculation is for the fan power implications of the observed leakage levels. In brief, assuming that a fixed amount of air is needed to cool or heat a zone, leakage downstream of the VAV box, by short circuiting the leaked air back toward the supply fan, makes the VAV damper open further to supply the same airflow to the room[3]. At 10% leakage, the required flow to the VAV box is $1/(1-0.1) = 1.11$ times the no-leakage flow scenario. The fan-power implication of an 11% increase in supply airflow is a 29% increase in fan power based upon the supply-fan power scaling with the airflow raised to the power 2.4 (Franconi et al. 1998). Although this average 29% increase in fan power appears to be significant, it is important to note that the range of impacts based upon a one-standard-deviation range of leakage levels is substantial (15% to 46% increase in fan power based upon the measured 45%

[2] If the amount of primary airflow entering a fan-powered VAV box is lower than a specified threshold (e.g., for the test building, less than 40% of the box's cooling maximum airflow), the control system turns on the fan in the VAV box. Except when dampers are commanded to minimum position during reheat periods, increased downstream leakage causes the powered VAV boxes to operate at higher primary airflows to compensate for the leakage and the fans do not need to run as often.
[3] Note that some of the leaked air is likely exhausted as relief air, which would also result in a thermal load that is not being considered here.

standard deviation in Table 3). Thus, the mean leakage value may not be good enough for decision making, particularly in retrofit sealing applications.

Fan-energy impacts of leakage downstream of VAV boxes have been measured in the same building in which we conducted our measurement-protocol testing. In particular, Diamond et al. (2003) measured the energy use of supply and VAV box fans[4] during a summer cooling period with "as found" leakage and then with added leaks (calibrated perforated plates). With the air-handling system operating in a fixed mode for whole-system leakage testing[5], the total supply fan inlet airflow was 11,500 L/s (24,400 cfm). In this mode, "as found" system leakage was 428 L/s (907 cfm, 3.7%); with leaks added downstream of VAV boxes, system leakage increased to 913 L/s (1,935 cfm, 7.9%). During normal operation with supply flows varying in response to room loads or daily special operation requests (e.g., morning smoke control operating-mode tests and building pre-cooling cycles), the added downstream leakage increased average total supply fan power from 5.4 kW to 6.8 kW (26%), but also reduced average total VAV box fan power from 1.3 kW to 0.8 kW (38%). The net result was a 16% increase in combined supply and VAV-box fan energy.

The energy impacts of leakage described above raise questions of energy cost and test/mitigation cost effectiveness. If we use an average supply air flow of 3 L/s per m^2 floor area (0.6 cfm/ft^2) (reasonably typical for an office-building VAV system), and an average fan power of 1.25 W per L/s (0.6 W per cfm) (40% overall wire-to-air efficiency for a fan operating at a 500 Pa (2 in.w.c.) pressure differential), the average fan power is 3.8 W/m^2 (0.35 W/ft^2). Thus, assuming that a single VAV box serves 50 m^2 (540 ft^2), and that it operates for 3000 hours per year, the annual fan energy consumption associated with that VAV box would be 3.8 W/m^2 x 50 m^2 x 3000 h / 1000 W/kW = 570 kWh/year. This corresponds to 11.4 kWh/m^2/year (570 kWh/50 m^2) or 1.06 kWh/ft^2/year. For comparison purposes, a U.S. Department of Energy study (Westphalen and Koszalinski 1999) indicated an average fan energy use of 13 kWh/m^2/year (1.2 kWh/ft^2/year) for VAV systems. Using the 13 kWh/m^2/year (1.2 kWh/ft^2/year) value, the average energy waste associated with the average 10% leakage would be 13 kWh/m^2/year x 50 m^2 x 29% savings = 190 kWh/year, which at $0.1/kWh translates to a cost of about $19 per year. Note that this does not include any costs associated with electricity demand charges.

[4] The building's relief fans operated with an irregular pattern to maintain room pressure set points (ran as needed to balance outdoor airflow supplied for ventilation, economizer use, and pre-cooling). There was no discernible correlation between system leakage and relief fan operation.

[5] In this leakage testing mode, both supply fans serving the system were operated at full-speed and each VAV box inlet damper was adjusted until the design duct static pressure of 1 in.w.c. (250 Pa) upstream of VAV box inlet dampers was achieved (in this case, each damper was adjusted to 75% of cooling maximum flow for that box). In addition, return dampers were fully open, outdoor air dampers were fully closed, relief fans were off, and VAV box fans and reheat coils were off. The air-handling system never operated in this mode during normal operation, however. The specified leakage test conditions were used only to provide an operation-independent reproducible condition for determining system leakage flows.

Using a discount rate (cost of money – inflation of electricity rate) of 3% over 30 years, the present value of this leakage is about $370. This implies that, based upon these assumptions (11% average leakage, 50 m^2 (540 ft^2) per VAV box, fan power savings only) one could spend up to $370 to test and seal that system section and still break even according to this economic yardstick. At the other extreme, to obtain a 3-year simple payback, the duct sealing and testing would need to cost $57. Once again, this does not include any savings in electricity demand charges, nor any impact on thermal loads, which can be quite significant and strongly dependent upon the outdoor air fraction (typically between 20% to 100%) and the climate.

Turning to the decision of whether to test before making a decision about sealing, if one assumes that it takes two people one hour (or one person two hours) to perform the modified fan pressurization test, and that those people cost $100/hour fully burdened, then each test costs $200, which translates to $20 per duct section at a 10% sampling rate. This testing cost needs to be compared to the value associated with differentiating systems with 5% leakage from systems with 15% leakage. One way to do this is to look at the maximum that could be spent for system sealing at 5% leakage, versus what could be spent at 10% or 15% leakage. Considering fan energy savings only (no demand charges or thermal energy implications), the calculation procedure above suggests that the maximum that should be spent to seal the downstream section serving 50 m^2 (540 ft^2) of floor area is $170 at 5% leakage, versus $370 at 10%, and $610 at 15%. This suggests that if sealing costs less than $170 / 50$m^2$ = $3.4/$m^2$ of floor area ($0.31/$ft^2$), it probably does not make sense to test, but rather makes more sense to always seal.

On the other hand, the financial decision between testing and sealing for an existing building should realistically be based upon the relative of cost of mobilizing to perform sealing and then discovering leakage that is too small to be sealed cost-effectively, as compared to the cost of performing exploratory leakage tests. It should be noted that the testing costs used for this analysis are based upon the test protocol described in this paper, which is designed for existing systems. Both testing and sealing costs should be lower in new construction, because of easier access to system components. Finally, these economic analyses are based only upon average central supply-fan annual electricity savings (no demand charges or thermal energy implications).

CONCLUSIONS

The results presented in this paper provide some tentative conclusions and some indications of needs for future work. First, the modified fan pressurization test procedure seems to produce reasonably accurate estimates of leakage downstream of VAV box inlet dampers in commercial buildings. This was found to be the case, irrespective of the magnitude of the leakage through the VAV box inlet damper, and irrespective of whether the testing was done in pressurization or depressurization mode. Second, large differences in system pressure upstream and downstream of box inlet dampers were observed when applying this technique, reinforcing the need to separately determine the leakage upstream and downstream of these dampers. Third, the leakage measured downstream of box inlet dampers was both substantial (on average, 10% of flow), and quite variable between buildings (48% standard deviation). This large variance between buildings supports the idea that testing is justified, at least when determining whether it is worth sealing leakage in an existing building.

Looking forward, it is clear that more systematic measurements of leakage downstream of VAV box inlet dampers should be conducted, including analysis of the costs of both testing and sealing in newly constructed and existing buildings. One specific question that should be answered is the statistical variability of leakage levels within a given building, as this would determine the sampling required to make go/no-go decisions with respect to sealing. Such testing could be performed in existing buildings, but would also be informative for new construction. Another issue that needs to be addressed is to identify the importance of different leakage sources, including equipment and accessory issues (e.g., the VAV box casings themselves, access doors, air-handler cabinets, grilles) versus construction quality issues (joints between duct sections and with equipment and accessories). Yet another area for research would be a comprehensive analysis of the complete economic implications of sealing, including the impact of sealing on electricity demand charges and thermal energy consumption. Finally, the validity of the protocol should be tested further on different systems and possibly by different practitioners.

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