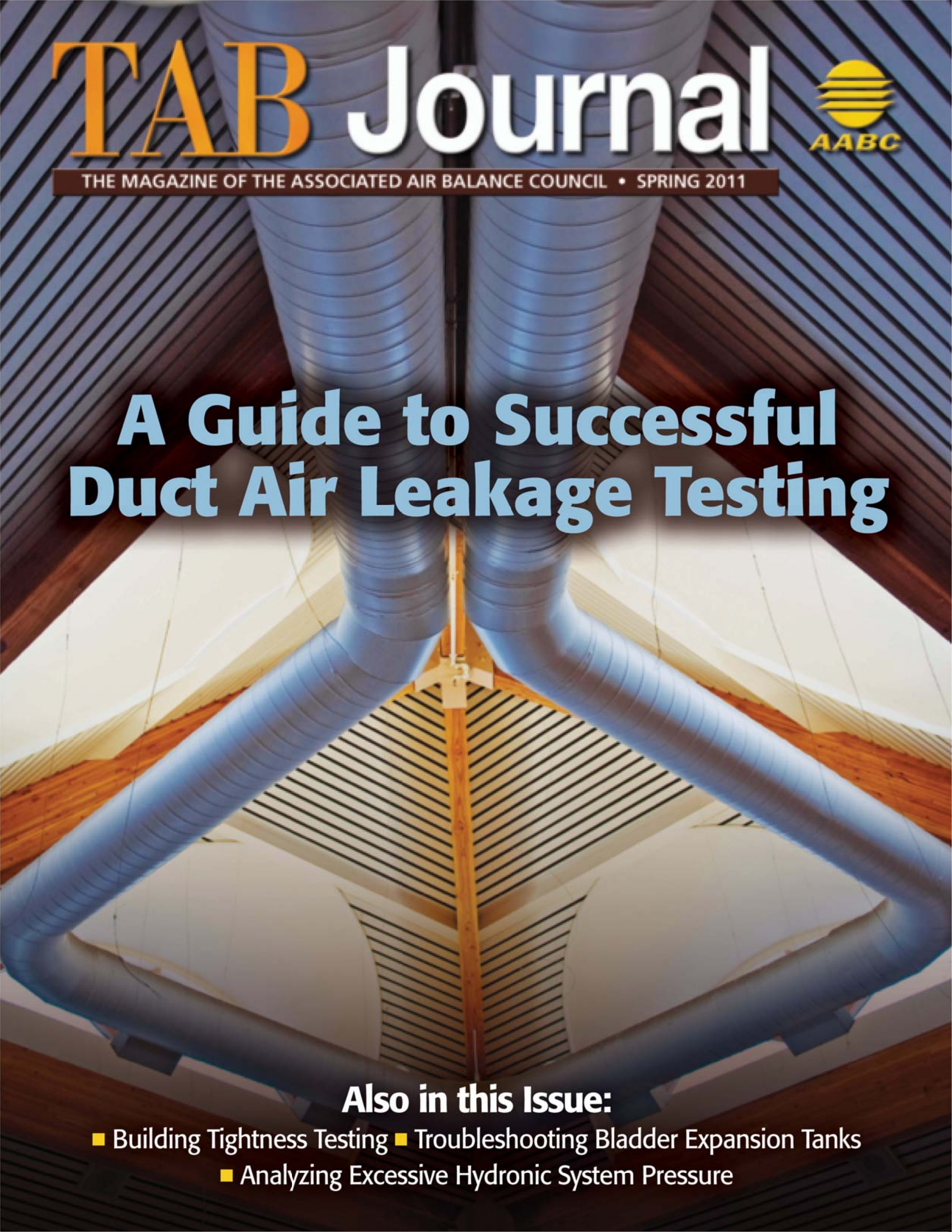
TAB Journal

AABC

THE MAGAZINE OF THE ASSOCIATED AIR BALANCE COUNCIL • SPRING 2011



# A Guide to Successful Duct Air Leakage Testing

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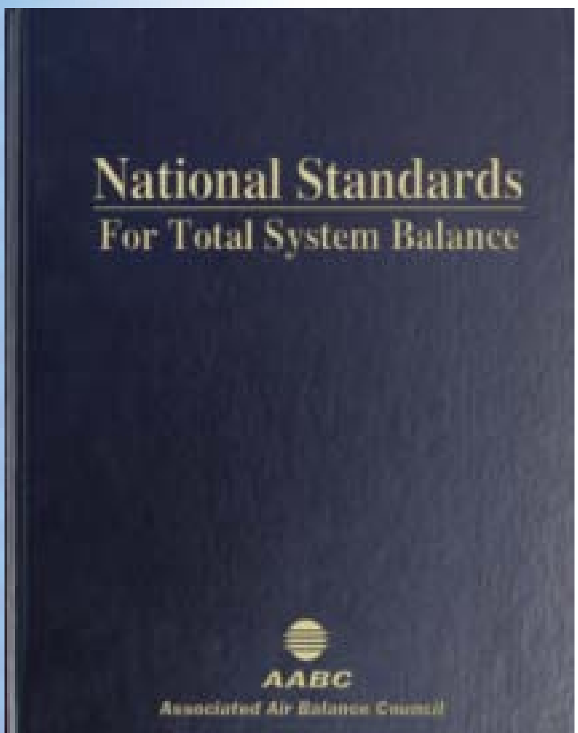


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# TAB Journal

**Associated Air Balance Council**





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# From the Publisher

The Spring 2011 issue of *TAB Journal* once again showcases the expertise of AABC test and balance engineers on a variety of issues.

This issue's feature article, "A Roadmap to Successful Duct Air Leakage Testing," by Gabriel Alejandre, TBE, of Los Angeles Air Balance Company, Inc., offers a detailed, step-by-step look at a critically important aspect of testing HVAC systems.

In other articles, Kevin Underwood, TBE, from Engineered Air Balance, Inc., discusses "Troubleshooting Heating Water Flow Problems." He makes suggestions on what may cause low water flow when all terminal valves are open.

"TAB Plan and Underfloor Air Distribution Systems," by Mat Chenvert, TBE, of Air Systems Engineering, Inc., presents an approach to creating a test and balance plan with an emphasis on issues encountered with increasingly common underfloor air distribution systems.

Also featured in this issue is "Bladder Expansion Tank Troubleshooting," submitted by David Dres, TBE, from Engineered Air Balance Co., Inc. Dres uses a case study to help explain how heating water expansion tank issues and failing boilers can coincide.

Southern Balance Company's Jonathan Young, TBE, illustrates methods for "Analyzing & Avoiding Excessive Hydronic System Pressure," while Chuck Kaupp, TBE, from Southern Independent Testing Agency, Inc., explores performance problems with hoods and small inline vertical mounted pumps in "Fan & Pump Issues."

"Air Building Tightness Testing" by Christopher A. McElwee, TBE, of Professional System Analysis, addresses the two types of testing to identify air leakage that is not visible to the naked eye.

This issue's Tech Talk provides insight on balancing valve placement, as well as when it is appropriate to install and the use of a dry-case type pressure gage.

We would like to thank all of the authors for their contributions to this issue of TAB Journal. Please contact us with any comments, article suggestions, or questions to be addressed in a future Tech Talk. We look forward to hearing from you!

# A Roadmap to Successful Duct Air Leakage Testing

**Gabriel Alejandre, TBE**
Los Angeles Air Balance Company, Inc.

Duct air leakage testing is often a source of confusion at the jobsite. Often the project's sheet metal foreman has never had to perform a duct leakage test or has done it, with a "bad experience" as the result, and is resistant to doing it again. This article is a roadmap to smooth out the whole process and make it as simple as it really is, saving time and effort for all involved.

## First Things First… Know Your Responsibilities

Duct air leakage testing is best accomplished with the coordinated effort of the mechanical engineer, the mechanical contractor and the air balance agency. A successful duct leakage test begins with the engineer clearly stating the test method and ends with the air balance agency testing a system that is properly sealed, sectioned, and pre-tested by the mechanical contractor.

This roadmap will attempt to maximize efficiency and minimize failed tests on the jobsite. Throughout this process it is essential that everyone understands their responsibilities in each phase of the duct air leakage test (or D.A.L.T.).

**It is the Mechanical Engineer's Responsibility to Specify Either A or B:**

A. Reference *AABC National Standards* To:[1]

1) Specify the systems to be tested for duct air leakage (i.e. supply, return, exhaust).

2) Select a test pressure that does not exceed the pressure rating of the duct construction.

3) Specify the system's maximum allowable duct leakage percentage.

B. Reference SMACNA HVAC Air Duct Leakage Test Manual To:[2]

1) Match the fan to the system pressure losses.

2) Designate the pressure class or classes for the construction of each duct system, as appropriate and cost effective, and clearly identify these in the contract documents.

3) Designate the seal class for each duct system.

4) Evaluate the leakage potential for ducts conforming to SMACNA or other standards and supplement the requirements therein with deletions and additions as may be prudent, giving due attention to the location of the ducts, the type of service, the equipment, dampers and accessories in the system, tolerances of volume regulating boxes, etc., independent of duct leakage.

5) Prudently specify the amount and manner of leakage testing and clearly indicate the acceptance criteria.

6) Avoid ambiguity created by references to non-specific editions of the SMACNA manual or other documents specified.

7) Ensure contract documents reflect a clear scope of work known to conform to applicable codes and regulations, including those addressing energy conservation laws.

8) Require adequate submittals and record keeping to ensure that work in progress conforms to the contract documents in a timely manner.

**It is the Mechanical Contractor's Responsibility To:**[3]

1) Prepare the duct sections to be tested.

2) Provide connections for duct leakage test apparatus.

**Duct air leakage testing is best accomplished with the coordinated effort of the mechanical engineer, the mechanical contractor and the air balance agency.**

3) Pre-test the systems prior to scheduling the air balance agency testing, to prevent failed tests and schedule delays.

4) Take corrective action to seal the ducts when the maximum allowable leakage rate is exceeded.

5) Allow sealant enough curing time before any duct pressurization.

**It is the Air Balance Agency's Responsibility to Include in the Duct Leakage Test Report:**

1) The date of the test.

2) The name and phase of the project.

3) A complete description of the ductwork tested, including location, sealing classification, and duct classification.

4) The test design static pressure, and the actual test static pressure.

5) The test design maximum allowable leakage rate and the actual leakage rate.

6) The calculation of the duct test section maximum allowable leakage rate.

7) The tested result, "pass" or "fail."

8) The orifice size, manufacturer, orifice tube serial #, and calibration date.

9) The actual orifice pressure differential and the actual airflow.

10) The name of the technician performing the test and any other inspectors or engineers witnessing the tests.

## Second…Determine the test criteria!

There are two common ways that the duct air leakage test criteria is specified.

The first we'll call the "engineer designated percentage" way. In the engineer designated percentage duct leakage test, the maximum allowable leakage rate is very clearly indicated as a percentage. The mechanical engineer designates the exact test criteria in the project specifications.

For example, the mechanical engineer's specification may read as follows: "The supply air main duct is to be pressurized to 1 ½ times the design fan static pressure with 1% of the system total design CFM as the maximum allowable leakage rate. Return air and exhaust air main ducts are to be pressurized to 1 ½ times the fan design static pressure with 2% of the system total design CFM as the maximum allowable leakage rate."

The second way, we'll call the "SMACNA D.A.L.T. method." This method references the SMACNA duct construction standards, pressure class, and seal class ratings (A, B & C). When using the "SMACNA D.A.L.T. method," an "allowable leakage rate" is determined by a "leak factor" given in CFM per 100 sq. ft. of ductwork surface area. Therefore, the more duct surface area your system has, the more air the system is "allowed" to leak.

This method of duct air leakage testing typically results in an "allowable leakage rate" in the range of about 5% to 10% of the system total airflow. Depending on the type of facility or application, this is often too much air leakage.

## Third….Define test sections and pre-test

Once the test criteria is determined, the mechanical contractor may choose to separate the system into "sections" such as risers, shafts, west half, east half, sub-terrain ducts, etc. The mechanical contractor should pre-test each section prior to any air balance agency testing, inspector witnessing, mechanical engineer witnessing, duct insulation installation, duct shafts are closed up, or any other structural construction that will have an impact is continued.

**When testing in "sections," coordination between the air balance agency and mechanical contractor becomes extremely important.** A dedicated working set of mechanical prints is essential to plan and track the duct testing progress and results. The various test sections and locations of the "bulk heads" (duct caps separating the various test sections) can be planned and pre-determined.

With the duct sections clearly defined on the working set of plans, each duct section's maximum allowable leakage rate can be calculated and pre-determined, allowing the mechanical contractor to pre-test any duct sections. The savvy mechanical contractor owns his own D.A.L.T. apparatus or in some cases will rent the apparatus on a weekly basis from an outside source, sometimes from the air balance agency. Some air balance agencies have multiple orifice tubes and fans for rent.

The benefit to the mechanical contractor of pre-testing is that when leaks are found, the mechanical contractor can seal the leaks and continue his duct construction without the delays that are inherent with having to call the air balance agency to test every little section of ductwork constructed (and retest each one that fails). Pre-testing allows the mechanical contractor to call the air balance agency only after several sections are complete and the mechanical contractor is sure the system (sum of the duct sections) will pass. **Pre-testing by the mechanical contractor is strongly recommended, and required by the *AABC National Standards*.**

Pre-testing of the duct sections is the key to avoid delays and/or added costs. When preparing the duct system or duct section to be tested, the mechanical contractor should consider the proper sealant application procedure, and the curing time of the duct sealant. Different sealants have different curing times based on temperature and humidity. It is important to allow adequate time for duct sealant curing to avoid "blowing bubbles out of the seams" during any pressurization of the ducts.

## Testing Duct System in Sections

When determining the allowable leakage rate for a section of ductwork using either one of these two methods, you must first calculate the total sheet metal area of the duct. When the mechanical engineer has designated the total system allowable leak rate to be a percentage of the fan total design CFM, and the entire system cannot be tested all in one duct air leakage test, the system is divided into sections.

The allowable section leakage rate is calculated by multiplying the total system allowable leak rate by the percentage of the total system to be tested. For example, if the fan system CFM total is 40,000 CFM, then 1% leakage is 400 CFM. With 400 CFM being the total system allowable leak rate and the test section representing 30% of the system total sheet metal ducting, then 30% of 400 is 120 CFM (400 * 0.30 = 120 CFM). This is the allowable leakage rate for the section under test (see equations below).

System Total Design CFM x Allowable Percentage of Leakage = System Total Allowable Leakage

$$\frac{\text{Section Total Sq Feet}}{\text{System Total Sq Feet}} \times \text{System Total Allowable Leakage} = \text{Section Allowable Leakage}$$

In the end, the summation of all of the test sections' leakage cannot exceed the total system allowable leakage.

## Positive vs. Negative Pressure Testing

The question of positive vs. negative pressure testing is often posed by mechanical contractors during the pre-testing phase. "Don't I have to test the return air ducts under negative pressure, and what about the exhaust?"

We have tested both return and exhaust ductwork sections under positive and negative pressure with little to no significant difference in the leakage rate. In the ASHRAE Handbook of Fundamentals, in the paragraph under Duct System Leakage it is stated "Sealed and unsealed duct leakage tests (AISI/SMACNA 1972, ASHRAE/SMACNA/TIMA 1985, Swim and Griggs 1995) have confirmed that longitudinal seam, transverse joint, and assembled duct leakage can be represented by Equation (37) and that for the same construction, leakage is not significantly different in the negative and positive modes."[4]

Equation (37)

$$Q = C\Delta p_s^N$$

$Q$ = duct leakage rate

$C$ = constant reflecting area characteristic of leakage path

$\Delta p_s$ = static pressure differential from duct interior to exterior, in. of water

$N$ = exponent relating turbulent or laminar flow in leakage path

With no significant difference in leakage rate results, we prefer to always test under a positive pressure. A positive pressure allows the use of smoke or a wave of your hand across duct connections to identify leaks. Under negative pressure, the identification of areas of leakage is much harder, more time consuming, and almost impossible at times.

## Fourth….TAB Testing Process/Procedures

After the duct test section has been prepared, a flexible tubing is connected from the test apparatus (radial fan and orifice tube) to the duct test section. Next, the static pressure probe is inserted into the test section (see Figure 1). The placement of the static pressure measurement point should be at least 12 inches away from the flexible tubing connection.

To prevent over-pressurization of the duct test section, the test apparatus fan inlet damper should be in the closed position before the fan is started. While monitoring the static pressure in the duct, the radial fan's inlet damper should be slowly opened.

While monitoring the duct static pressure, adjust the fan inlet damper until the static pressure in the duct test section is equal to the specified test pressure.

In a "steady state" of static pressure, the amount of CFM needed to maintain this "steady state" is your actual leakage in CFM for that duct section. In all duct leakage tests, whether it is a duct section test or a total system test, the maximum allowable leakage rate and test pressure will determine the size and number of orifice tubes and fans needed. The test equipment selected should be capable of supplying 110% of the maximum allowable leakage rate at the specified test pressure. The test equipment capacity becomes a factor when adding, removing, or combining duct sections (see Figure 2).

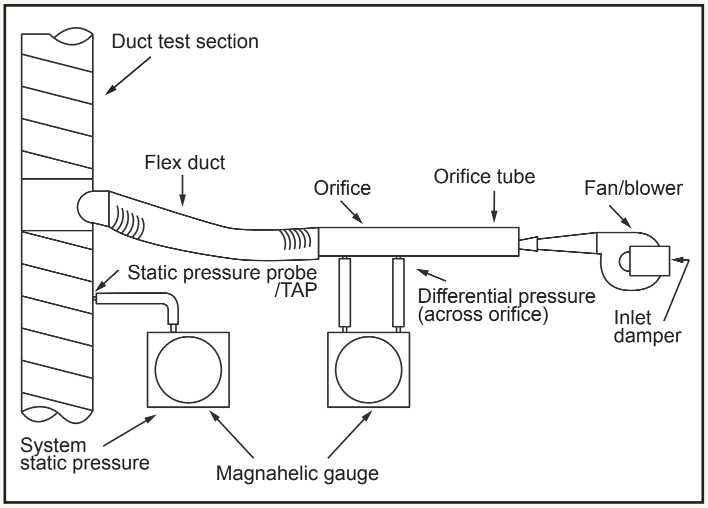


*Figure 1*

When an individual leakage test apparatus is unable to achieve the design test criteria, whether it is volume or static that is not achieved, two or more test apparatus can be operated simultaneously. Using two or more fans in a series operation, supplying a single orifice tube, will increase the static pressure at the same CFM. The use of two or more parallel test apparatus (multiple fans and orifice tubes) will increase the CFM (air leakage rate) while maintaining the same static pressure.

In parallel operation, all fans need to be running at the same time to prevent airflow through inoperative apparatus and the parallel fans should be dampered so that each is delivering roughly the same airflow. After the duct system is pressurized to the required test pressure, the total amount of air leakage out of the duct section is the sum of the CFM measured through all the orifice tubes.

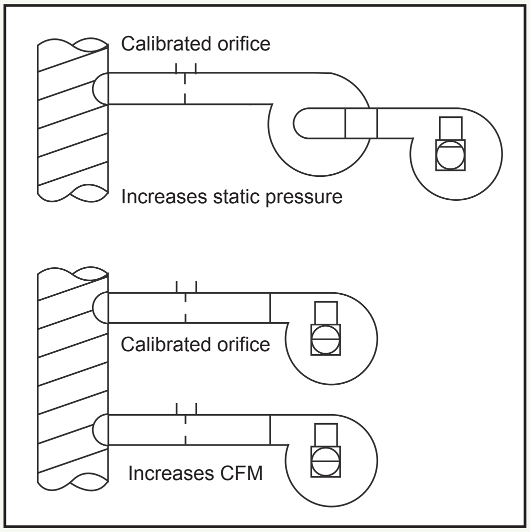


*Figure 2*

To quantify the leakage rate, a difference in pressure is measured across a calibrated orifice plate contained in the orifice tube. The pressure differential is then used to determine the airflow by calculation or by referring to the calibrated orifice tube's curve specific to that orifice tube. By comparing the "maximum allowable leakage rate" to the actual rate of leakage, a pass or fail of the duct test section can be reported.

Manufacturers are also in a position to help the duct air leakage testing process. It would be of great help to the whole industry if manufacturers of fire dampers, access doors, automatic dampers and VAV/terminal units would provide external leakage rate vs. pressure tables for their products. At a minimum, the manufacturers should provide the mechanical contractors with instructions on how to properly seal their products to reduce leakage without affecting performance or violation of the U.L. listing.

If the total system leakage is of any concern to the mechanical engineer, the engineer should require that the total system allowable leakage be calculated and submitted for approval before any D.A.L.T. is performed, especially if the "SMACNA D.A.L.T. Method" is to be referenced. If the "SMACNA D.A.L.T. Method" is not to be used the mechanical engineer should indicate what allowances were made for apparatus leakage at operating pressure, even if it is zero.

## An Observation

We have seen the "SMACNA D.A.L.T. Method" being applied to leak test various systems. The usual result is that this method benefits the mechanical contractor by allowing a higher rate of leakage on the ductwork he installs.

If the system requires additional sealing (due to this higher allowed leakage rate) after it is installed to satisfy owner operating requirements, the mechanical contractor usually benefits because he can require additional funds for additional sealing, which in turn is more labor intensive after the building, ceilings and systems are completed. The owner does not benefit, the mechanical engineer does not benefit, and there is no benefit to the air balance agency.

The only one who benefits from using the SMACNA D.A.L.T. method is the installing mechanical contractor. See the following for Energy Evaluation and Cost Comparison for various amounts of leakage.

**Energy Evaluation and Added Cost Due to Leakage**[5]

$$\text{Cost/Yr} = \text{BHP} \times \frac{0.746\ \text{kW}}{1\ \text{BHP}} \times \frac{\text{Hrs}}{\text{Yr}} \times \frac{\text{Cost}}{\text{kWh}}$$

**Assumptions:**

Fan CFM: 39,600

Fan Operation: 7000 Hrs/Yr (80%)

Utility Rate: 0.20 $/KWH

| % LEAKAGE | FAN CFM | FAN SP | FAN BHP | OPERATING COST ($/YR) | ADDITIONAL COST ($/YR) (FOREVER) | % COST INCREASE (FOREVER) |
|---|---|---|---|---|---|---|
| 0% | 39,600 | 4.3 | 36 | $37,600 | $0 | 0% |
| 1% | 40,000 | 4.4 | 37 | $38,600 | $1,000 | 3% |
| 5% | 41,600 | 4.6 | 42 | $43,800 | $6,200 | 16% |
| 10% | 43,600 | 4.8 | 46 | $48,000 | $10,400 | 28% |
| 15% | 45,500 | 5.0 | 50 | $52,200 | $14,600 | 39% |


Note that the comparison between 0% and 15% leakage rates reflects a 39% annual cost increase to operate the fan!

What is the significance of our chosen leakage rates? SMACNA indicates that if a duct system is not properly sealed, the designer "must make allowances in his calculations for a minimum of 15% duct leakage".[6] Thus we have established our upper limit at 15%. For a medium or high pressure design (2" W.G. and up) SMACNA specifies that: "Total allowable leakage should not exceed one (1) percent of the total system design air flow rate."[7]

However, United McGill maintains that a quality fabricated duct system, properly installed and sealed, can achieve leakage rates as low as ½ of 1%, and we recommend sealing all sections of a system, regardless of pressure classification. Therefore, proper installation and sealing of duct systems can cut leakage rates from 15% or more down to almost nothing, resulting in substantial operating cost savings.

Since the power required to operate HVAC air handlers is normally the largest single contributor to building energy costs, every effort should be made to reduce this expenditure. As this analysis has shown, control of leakage can play a very significant part in reducing these costs.

## Conclusion

Duct leakage testing may seem confusing at first, but nothing could be further from the truth. It cannot be stressed enough that testing starts with the mechanical engineer. By clearly defining the test criteria, the mechanical engineer provides needed guidelines for the mechanical contractor to adequately seal and section the ductwork for duct leakage pre-testing.

Once pre-testing is successfully completed, the mechanical contractor allows the air balance agency to complete their duct leakage testing in a timely fashion by eliminating additional charges/delays due to failed duct leakage tests. With all parties of the project aware of their roles, actively participating and following a coordinated roadmap, you can turn a potential headache into something that saves the building owner money, satisfies the mechanical engineer's requirements, and keeps the project progressing on schedule.

[1] *AABC – National Standards for Total System Balance (AABC, 2002) 42.*
[2] *SMACNA HVAC Air Duct Leakage Test Manual-1st Ed. (SMACNA,1985) 2-1.*
[3] *AABC – National Standards for Total System Balance (AABC, 2002) 42.*
[4] *2005 ASHRAE Handbook of Fundamentals (ASHRAE, 2005) 35.14*
[5] *Engineering Report: Duct Leakage and System Performance (Ohio:United McGill) No. 145-4 (updated utility rate, and calculations based on rate, to 2010 amounts)*
[6] *HVAC Duct System Design (1981), SMACNA, p.2.4*
[7] *High Pressure Duct Construction Standards (1975), SMACNA, p.65*

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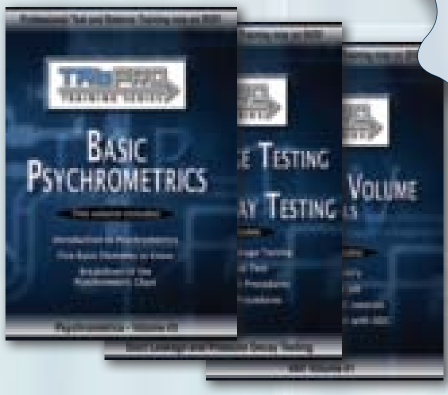


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# Troubleshooting Heating Water

**Kevin Underwood, TBE**

**Engineered Air Balance Co., Inc.**

A recent project presented problems with heating water flow on the lowest level of a 10-story building (Level-LL2). The heating water pumps (HWP) and heat exchangers are located in the penthouse (seventh floor) and the other remaining floors were still under construction.

By reading out all the heating coils on Level-LL2 with the HWP operating at 60 HZ, it was determined that the water flow on the floor was very low. After this was established, each level of the building was examined to see what was actually open to the system. All the isolation valves to each level were found shut.

This data still did not resolve the problem. Level-LL2 only had a total of 97 gpm and the HWP was submitted for 730 gpm at 90' head. At this time, pressures were measured throughout the building to determine where the loss of flow was occurring. The schematic of the heating water system for this building (see Figure 1) will help through this troubleshooting process.

The first pressures taken were leaving and entering the heating pump. The leaving (1) pump pressure = 81 psi and the entering (2) pump pressure = 34 psi. These pressures were plotted on the pump curve and indicated that the pump was not moving enough water. The analysis continued back through the system to the heat exchanger. The leaving (3) pressure = 34.0 psi and the entering (4) pressure = 34.0 psi, indicating no ΔP. Continuing on back to the strainer entering the heat exchanger the leaving (4) pressure = 34.0 psi and the entering (5) pressure = 35.5 psi. This was not an excessive ΔP across the strainer the analysis was continued down the building riser.

It was important to consider the elevation of pressures because the next closest place to take a pressure was on Level-4. The elevation difference between entering the strainer (5) and the closest terminal box on Level-4 (6) was 45.4 ft., which equates to a 19.6 psi pressure gain from the Penthouse Level down to Level-4. Since it was already known that the pressure entering the strainer (5) was 35.5 psi, pressure was taken in the return line at the closest terminal box to the riser (6) of 55 psi. Subtracting the pressure gain by elevation the ΔP between the two points was approximately zero.

Next, pressures were taken at the first three terminal boxes (7) of Level-LL2 that were closest to the riser. All three terminal boxes had a return pressure of 95.2 psi. The elevation from Level-LL2 (7) to the Penthouse Level (5) is approximately 137.8 ft., which equates to 59.7 psi pressure gain between the two points. Subtracting the pressure gain by elevation the ΔP between the two points was approximately zero.

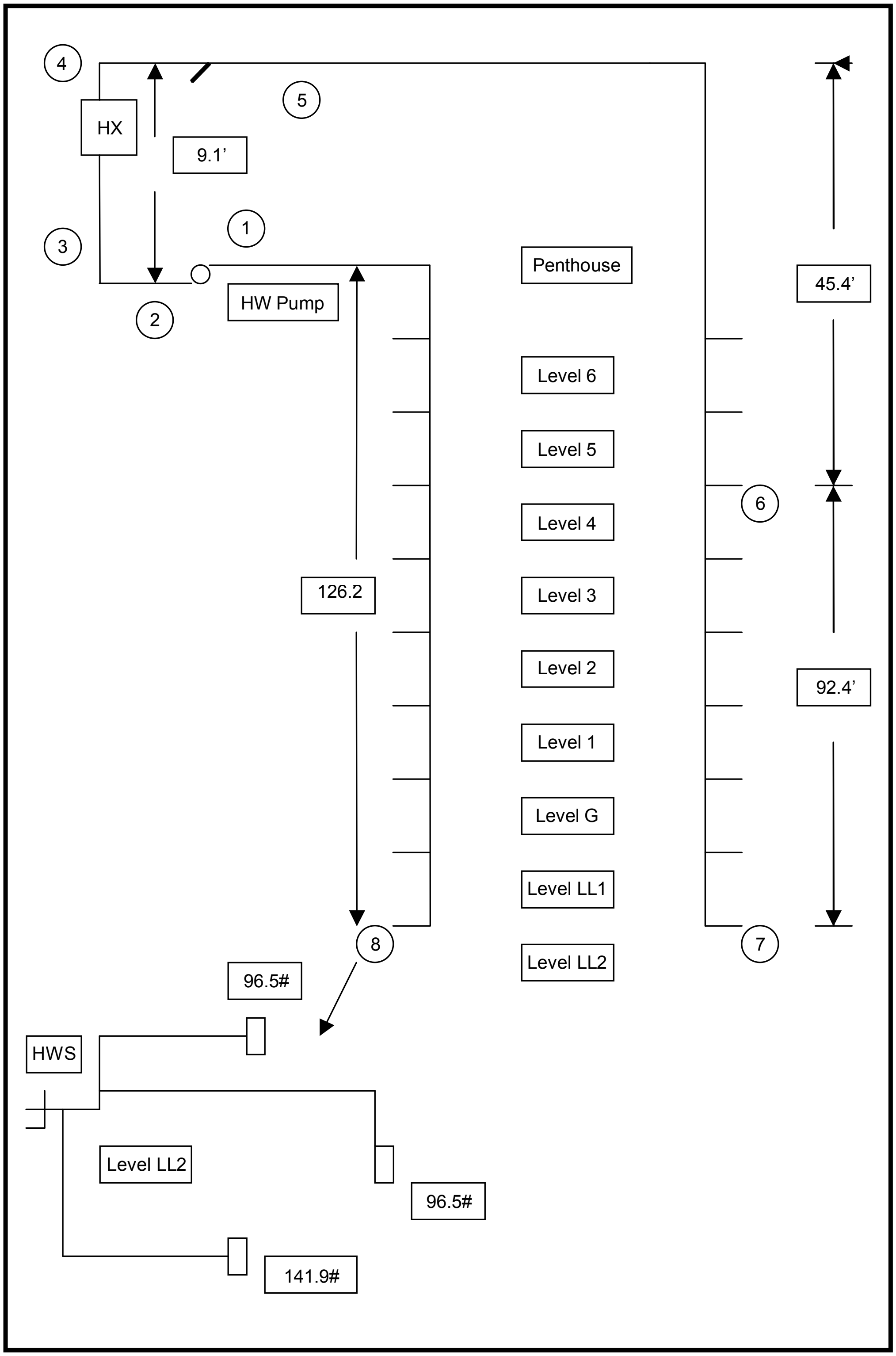
Next, pressure was measured in the supply line at the same three terminal boxes (8). The first terminal box from the riser had a pressure of 141.9 psi. This box was tapped from the vertical header right after the floor isolation valve. Then the next two terminal boxes were measured with a supply pressure of 96.5 psi. This was a 45.5 psi pressure loss between the box tapped in the vertical header and the terminal boxes tapped from the horizontal header.

At this time, it was suggested that the mechanical contractor cut open the supply header at the transition from vertical to horizontal. Doing this revealed that there was nothing in the horizontal pipe. A ceiling tile wire twisted into a hook was inserted up past the 90° turn up into the header. After a couple of feet, it hit something hard and when it was pulled out, a bunch of slag fell down.

It was suggested that the mechanical contractor break open the vertical header and locate the obstruction. All this testing and pressure calculating determined that a 3" coupon (a cut out from a pipe) was lodged in the 90° turn from vertical to horizontal.

In conclusion, if water flow is low in a system and terminal valves are wide open, start back to the basics and perform a pressure profile of your system. In most cases this will determine at least if there is an obstruction within your system.

# Flow Problems

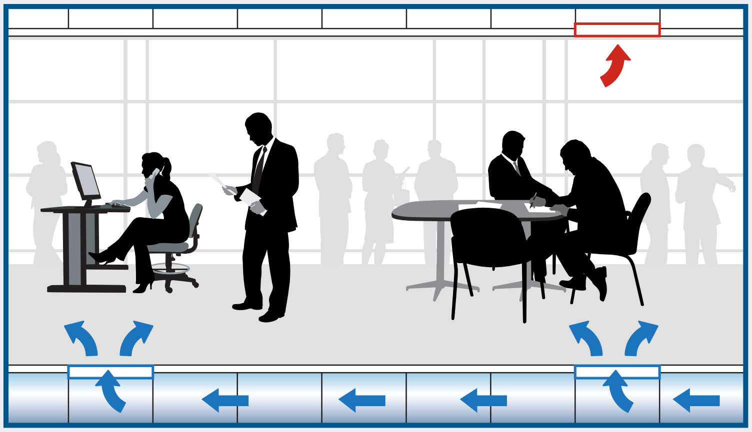



*If water flow is low in a system and terminal valves are wide open, start back to the basics and perform a pressure profile of your system.*

**Figure 1.**
Schematic of the heating water system for a 10-story office building.

# TAB Plan and Underfloor Air Distribution Systems

**Mat Chenvert, TBE**
Air Systems Engineering, Inc.



Underfloor air distribution (UFAD) systems are becoming more common in the United States as alternative HVAC systems. This type of system uses the structural concrete slab and a raised access floor to create a plenum. The pressurized plenum is used to deliver conditioned air to the occupied space through a variety of floor diffusers.

This type of system was introduced in West Germany in the 1970s and was used in other areas of Europe prior to the 1990s. These systems became very popular in Japan in the early 90s in new office buildings.

## TAB PLAN

These UFAD systems present some unique challenges to the test and balance professional. In order to make the project go as smoothly as possible, the TAB professional should develop a detailed TAB plan. The TAB plan should include the systems to be tested and the strategies and procedures to be used. Particular attention should be paid to the following issues unique to UFAD systems:

1. Scheduling of floor system installation, floor diffuser installation, carpet installation, and cubicle/furniture delivery. All of these items affect the removal of floor panels for access.
2. Emphasis of the underfloor access required for testing and balancing and when access will be required. It is important that all systems are ready for TAB while underfloor access is still available.
3. Unique tests which may be required include an underfloor air leakage test, air delivery test, and required plenum static pressure test.

This TAB Plan should be distributed to the construction manager and mechanical contractor as early as possible. The scheduling and access issues should be discussed so that the construction schedule can be altered if necessary to facilitate proper and accurate testing and balancing of the systems.

The very nature of the system requires that much of the equipment that is tested is located under the access floor. This equipment can include:

1. Main supply air ducts
2. Air moving equipment such as fan terminal units
3. Balance dampers
4. Hydronic balance valves

The raised access floor is typically installed before the systems are ready for testing. As a result the equipment listed above will be accessed through removable floor panels. These panels are usually held in place with screws. It is important that the construction manager understands that access to the all of the above equipment is required. Be sure they understand this includes ALL of this equipment. There is a tendency of many construction managers to underestimate the scope of testing and balancing. A system cannot be properly balanced when access is only provided to most of the equipment. All of this access is required after equipment start-up, and the temperature controls are finished so the system can be properly balanced.

When a construction schedule is being “squeezed” it is common to forget about TAB. Often with UFAD projects there is a rush to get the floor closed up and get the carpet installed. These steps make it difficult to locate the equipment and select the correct floor panel to remove, making access to the underfloor equipment system difficult and time consuming. Once heavy furniture and cubicles are moved in and installed, access to some areas is nearly impossible.

These issues can easily be avoided by submitting a thorough TAB plan and attending construction meetings in order to make everyone aware of the TAB requirements necessary to provide the owner with a properly functioning HVAC system.

# Tech Talk

Facilitating better understanding of proper balancing procedures has been part of AABC's mission for more than 40 years and helps to produce buildings that operate as designed and intended. Tech Talk is a regular feature in which AABC shares questions we've received and the responses from the association's experts. We hope that others have had similar questions and, therefore, will benefit from the answers. Readers are encouraged to submit their own questions about test and balance issues.

## Have a Question?

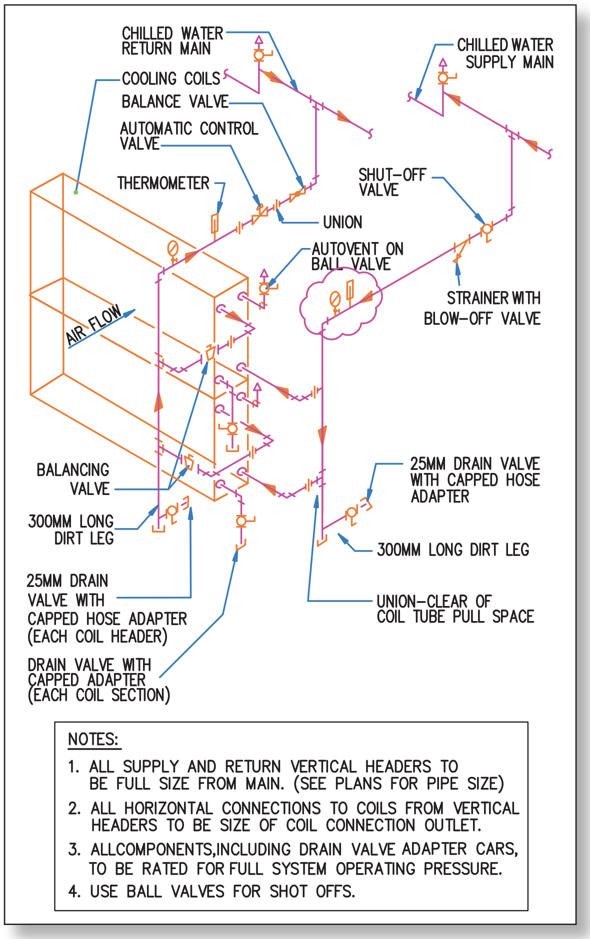
***To submit a question for Tech Talk, email us at info@aabc.com***

**The Associated Air Balance Council** frequently fields technical questions from engineers, contractors, owners and others regarding proper air and water balancing procedures.

These questions are answered by the most qualified people in the industry: **AABC Test & Balance Engineers (TBEs).**

## Balance Valve Placement for Coils

**QUESTION:** ***The figure detail (below) depicts a balancing valve on the return line of each coil section, as well as another on the common return line. Therefore for each coil it seems to require 3 balancing valves. Why do I need to install 2 more balance valves on each return pipe of coil section if there is already a balance valve on the common return pipe? My opinion is that the water flow can be balanced via the balance valve on the common return line, and the other 2 balance valves may be deleted. Do you agree?***




**AABC:** The balancing valves at the coils are needed, but in addition, we recommend pressure taps on the supply and return pipe connections at the coils (4 locations). The common return balancing valve should be a "calibrated" type valve for measuring and setting the total flow. The coil balancing valves can be ball valves or plug cocks with memory stops if pressure taps at the coils are added, if no pressure taps are added, calibrated balancing valves on each coil will be needed.

*-Steve Young, TBE, The Phoenix Agency Inc.*

## Pressure Gauges

**QUESTION:** ***On a U.S. embassy project, the U.S Government specifications do not require analog dry case pressure gauges at inlet and outlet piping of the hydronic coils of air handling units. However, another U.S. Government spec for testing, adjusting and balancing, in section 3.17 Heat-Transfer Coils, calls for measurement of the water pressure drop for each coil.***

***As far as I know, the water pressure drop is measured by means of a special device or equipment by the TAB technician, and not with an analog dry-case type pressure gage. Hence, there is no need to install a dry-case type pressure gage at the inlet and outlet of hydronic coils of air handling unit. Is that correct?***

**AABC:** We would use our own calibrated meter to measure the pressure. We would remove any installed gauge, take the measurement with our meter, and reinstall the gauge previously removed.

However, if the contract drawing detail shows a pressure gauge we cannot say it does not need to be installed.

*-Joseph E. Baumgartner III, P.E., TBE, Baumgartner Inc.*

**AABC:** The spec provided requires a water pressure drop, it does not mention the installation of any type of pressure gauges. We would read the ΔP at each coil with our calibrated digital water meter. If the coil is required to have analog gauges installed, that is usually a requirement of the mechanical contractor under hydronic specialties.

*-Gaylon Richardson, TBE, Engineered Air Balance Co., Inc.*

# Bladder Expansion Tank Troubleshooting

**David Dres, TBE**
Engineered Air Balance Co., Inc.

A recent project had many problems with the heating water expansion tank as well as issues with the boilers which emphasized the problem. The boilers would fail on internal errors several times a week. When the boiler failed, the heating water supply temperature would drop to about 60ºF in 6 – 8 hours. The mechanical contractor would reset the boiler and as the heating water supply temperature increased, the system pressure would also increase. The pressure relief valve would then pop off, flooding the mechanical room.

The expansion tank was installed on the ground floor of a multi-story building. It was determined that the make-up water pressure regulating valve needed to be set at 55 psi to maintain 10 psi at the top of the system. The pressure relief valve installed had a rating of 75 psi.

*The problem with the expansion tank and the pressure relief valve popping off was due to the expansion tank being undersized, instead of a problem with the expansion tank pre-charge or a defective bladder.*

The bladder expansion tank was factory charged to 12 psi and the charge pressure had not been modified. The mechanical contractor increased the expansion tank charge pressure to 55 psi. The problem with excessive system pressure continued as the system was heated.

It was discovered that the mechanical contractor had not isolated the expansion tank from the system when the tank charge was changed, as recommended by the expansion tank manufacturer. The expansion tank was isolated and the water side of the tank was opened to atmosphere. The tank charge pressure was reset to 55 psi. The problem continued with little improvement.

During the testing of the expansion tank operation, the heating water supply temperature and the system pressure was logged.

The expansion tank manufacturer representative was contacted by the mechanical contractor, and they suggested testing the system with the expansion tank charge set 10 psi below the system pressure (45 psi) and 10 psi above the system pressure (65 psi). No improvement was noted with the different bladder pressures. The pressure relief valve continued to pop off at 75  psi. The operation of the pressure relief valve was verified. The mechanical contractor replaced the pressure relief valve with no improvement as a precautionary measure.

The consulting engineer was contacted to aid in troubleshooting the system. They verified the sizing of the expansion tank and suggested changing the pressure relief valve to a valve with a 100 psi setting. The pressure relief valve was changed to a valve with 100 psi set point, but the problem continued.

To aid in the troubleshooting, some research on bladder expansion tank sizing was conducted with the ASRAE HVAC Systems and Equipment Handbook as a reference. The formula for bladder expansion tank sizing is:

$$V_t = V_s \frac{[(v_2 / v_1) - 1] - 3\alpha\Delta t}{1 - (P_1 / P_2)}$$

**Where**

$V_t$ = volume of expansion tank, gallons

$V_s$ = volume of water in system, gallons

$v_1$ = specific volume of water at lower temperature, $ft^3/lb$

$v_2$ = specific volume of water at higher temperature, $ft^3/lb$

$\alpha$ = linear coefficient of thermal expansion, in/in · ºF

= 6.5 x 10-6 in/in · ºF for steel

= 9.5 x 10-6 in/in · ºF for copper

$\Delta t = (t_2 - t_1)$, ºF

$t_1$ = lower temperature, ºF

$t_2$ = higher temperature, ºF

$P_1$ = pressure at lower temperature, psia

$P_2$ = pressure at higher temperature, psia

*(2008 ASHRAE Handbook – HVAC Systems and Equipment)*

The heating water system had a volume of 6800 gallons, with a low temperature of 60ºF and a high temperature of 180 ºF, with a low pressure of 55 psi (or 69.7 psia) and a high pressure of 75 psi (or 89.7 psia), thus:

$$V_t = 6800 \times \frac{[(0.01651 / 0.01603) - 1] - 3\,(6.5 \times 10^{-6})\,(180 - 60)}{1 - (69.7 / 89.7)}$$

Vt (expansion tank volume) = 842 gallons

The bladder expansion tank on this project was scheduled for 400 gallons.

The expansion tank size was calculated with several different parameters:

1. The high pressure was increased from 75 psi to 125 psi.

Under this condition Vt = 375 gallons

2. The low pressure was reduced to 10 psi (like the expansion tank was installed at the high point in the system) and the high pressure was adjusted to 20 psi above the low pressure (or 30 psi)

Under this condition Vt = 420 gallons

It appeared the problem with the expansion tank and the pressure relief valve popping off was due to the expansion tank being undersized, instead of a problem with the expansion tank pre-charge or a defective bladder.

The consulting engineer was contacted to discuss the situation and shortly a price request was issued to install another 400 gallon bladder expansion tank in parallel with the original 400 gallon expansion tank. When the second expansion tank was installed the system pressure would vary at the pressure relief valve from 55 psi with a cold system to 76 psi at design set point (180ºF).

# Analyzing & Avoiding Excessive Hydronic System Pressure

**Jonathan Young, TBE**

**Southern Balance Company**

A condominium building owner wonders, "What can I do to address a problem with my secondary chilled water pump? Ever since it was installed last fall, it has been breaking tenant water-usage meters and flexible pipe couplings due to its extremely high pressure."

A short site meeting revealed that the fundamental requirements for the pump and the piping system did not match, and that the system had been balanced with pressures far exceeding the system's rated working pressure of 150 psi. In fact, the balancing report had pressure readings above 200 psi at numerous condominium units on the lower levels.

The 26-story building had 332 feet of elevation between the pump and the highest chilled-water pipe, with 20 feet of elevation between the pump and the lowest coil. Therefore it took about 143 psi at the pump just to fill the system with water.

All of the components and pipe were pressure rated for 150 psi, and all of the coils had two-way control valves. Obviously, the next step was to test the pump, which had design requirements of 680 gpm at 270 feet of head.

A quick discussion with the building engineer established that the pump was being operated at 60 Hz in order to maintain 15 psi at the pressure differential transmitter. Upon examination, the transmitter was found to be inoperable, causing the pump to always run at 60 Hz.

Therefore, during the recent winter, the control valves had all closed at some point, allowing the pump shut-off head of 134 psi to increase the minimum system static pressure from 143 to 277 psi at the pump. No wonder this building had a lot of water leaks.

As it turned out, on the warm summer day that the system was being evaluated, the pump was running at only 120 feet of head at 60 Hz. However that low head still delivered 1180 gpm—which was a startling 73% above design. At this point, a simple plot of the system curve *(see Figure 1)* indicated the pump would deliver the design chilled water flow of 680 gpm with only 40 feet of head at about 35 Hz pump speed.

The key to operating this system was to determine where to run the pump so it would not exceed 150 psi at the lowest coil and still deliver adequate water flow. This involved two simple steps:

1) Reduce the system static pressure by 9 psi and still maintain 5 psi at the highest level, which is all that is necessary to prevent air from entering the system.

2) Determine the pressure change between the pump and the lowest coil, which was 20 feet above the pump, or about 9 psi. Therefore, the pump can be run at 159 psi discharge system pressure without risking further pipe damage.

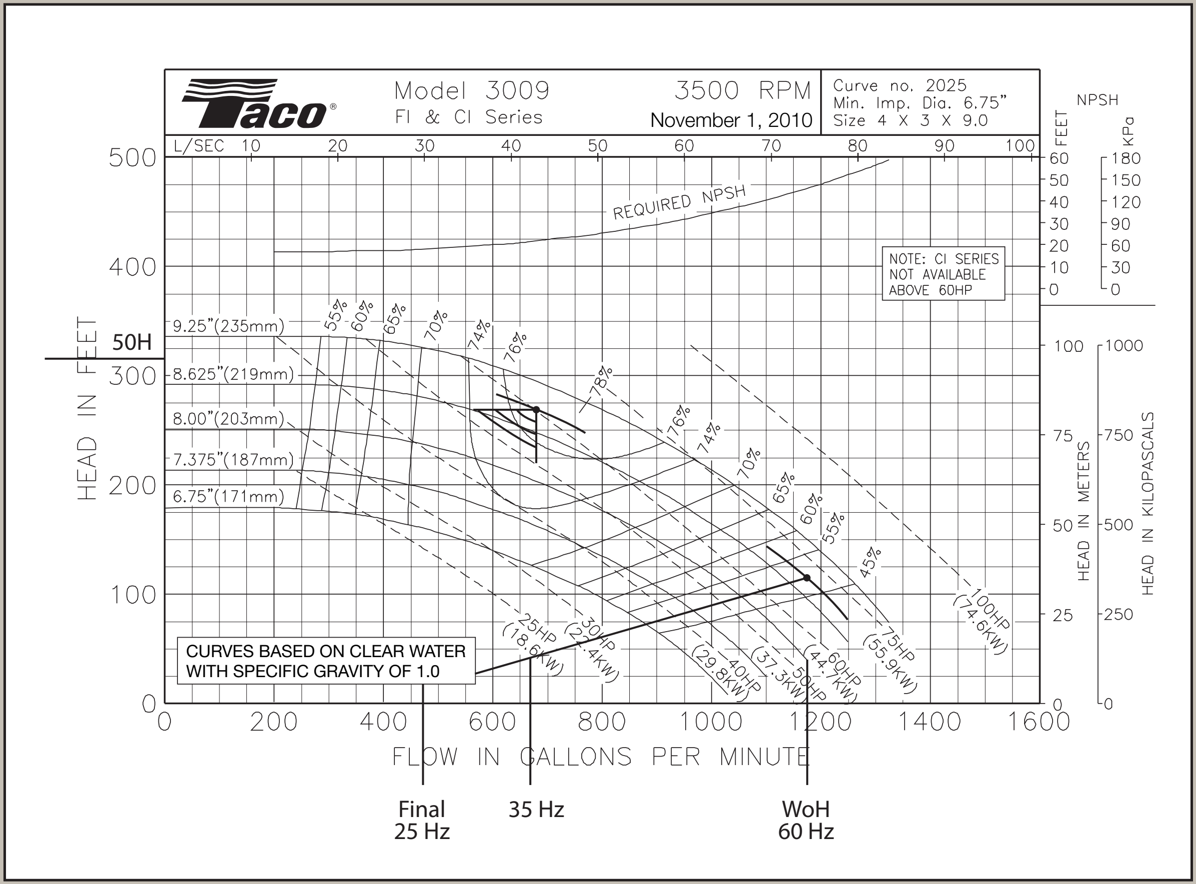
After reducing the pump speed to 35 Hz to obtain the 159 psi, the waterflow was at 680 gpm. This was right on design, but at only 40 feet of head. The design requirement of 270 total dynamic head must have erroneously included an allowance for building height in the total head calculations used to select this pump.

Now, the return water temperature rise during summer operation was still below design, indicating too much water flow. Therefore the pump speed was gradually reduced to 25 Hz, resulting in a new return water temperature rise of 11 degrees with total flow of 480 gpm. At this point, the discharge pressure was only 143 with 134 psi at the lowest coil while still maintaining good comfort conditions in the building.

Obviously, this building will operate with substantial diversity. Unfortunately for the owner, selection of a smaller, much more efficient pump would have reduced the first cost of the pump and saved a tremendous amount of energy over its life. On the other hand, the owner was extremely lucky to have oversized pipe mains that resulted in very low friction loss—allowing adequate water flow without exceeding the equipment pressure limits.

The final operating point for this system eliminated further pipe damage, reduced the water flow from 173% to 70% of design, and cut the horsepower from 70 to 26 HP. An important safety feature was implemented by programming the variable frequency drive to limit the speed to 35 Hz. Of course, the shut-off discharge pressure can still go up about 9 psi higher, but as long as the pressure transmitter is working, the pressure should never go over 159 psi.

**Figure 1. System Plot on Pump Curve**




# Fan & Pump Issues

**Charles W. "Chuck" Kaupp, TBE**
Southern Independent Testing Agency, Inc.

Equipment test results from a fan or a pump do not always follow the performance curves. In the last few years, two unusual issues arose, relating to an exterior wall mounted kitchen hood exhaust fan and a small 30 GPM circulator pump.

## Case #1 - Kitchen Hood Exhaust Fan

This is an issue relating to an exterior wall mounted, large kitchen hood exhaust fan. The following information (nameplate and actual) was obtained.

| | DESIGN CFM | ACTUAL CFM |
|---|---|---|
| Total Air Volume | 11,000 | 10,409 (Hood Face Reading) |
| Fan RPM | 1164 | 1250 |
| Horse Power | 7.5 | 7.5 |
| Brake Horse Power | 6.5 | 7.25 |
| Volts | 460 | 478 |
| Amperage | 9.7 | 9.03 |
| Motor Sheave | 4-3/4" | 4-1/4" |
| Fan Sheave | 6" | 6" |
| Motor RPM | 1760 | 1770 |

There were no problems with these test results and no operating complaints.

The hood was in operation for two months when the restaurant started to complain of the hood performance and requested that the contractor improve the air to or above design. The contractor increased the fan RPM to the maximum sheave adjustment and changed to the next larger motor, due to increased amperage draw, but the owner still had complaints related to the hood performance.

The test and balance firm was then informed of the conditions several months later, and was requested to re-test the air volume and provide suggestions for correction. To ensure accurate readings, it was suggested that the welded exhaust duct be drilled to allow for a duct traverse test. This was done resulting in a total air volume of 9413 CFM at the traverse and 9233 CFM at the filter face. These results were baffling, considering the first results at the filter face were 10,409 and now with a sheave adjustment and motor change, the results reduced to 9413 CFM.

In an attempt to solve the problem, fan and duct static pressures were taken and compared to the fan curve, which did not lead in a corrective direction. Finally, the contractor provided a small statement that they had been finding thrown belts on occasion. A disassembly of the fan was requested to check the sheave alignment and the distance of the fan wheel to the fan inlet volute.

It was found that the fan wheel and fan shaft had shifted and actually moved over ¾" away from the inlet volute. A possible analogy to this condition is, if your lips are not on a straw, you cannot suck any fluid. The manufacturer of this fan recommends for the fan wheel to actually be over the inlet volute by ¼". The fan was properly realigned, reassembled and retested with final results of 11,470 CFM. Problem and complaint solved.

## Case #2 - Small Circulator Pump

This is an issue relating to a small inline vertical mounted pump, rated at 30 GPM at 35 foot head. The pump was utilized for a test board with five circuits for balancing. To prepare the test board, a pump curve is mounted on the test board to allow plotting of the actual curve, based on the dead head (shut-off) of the pump.

The pump dead head was performed, however the actual plotted curve was greater than the actual nameplate installed impeller. The pump was disassembled to measure the installed impeller, which was per the nameplate. The taps on the pump house were checked and found to be proper.

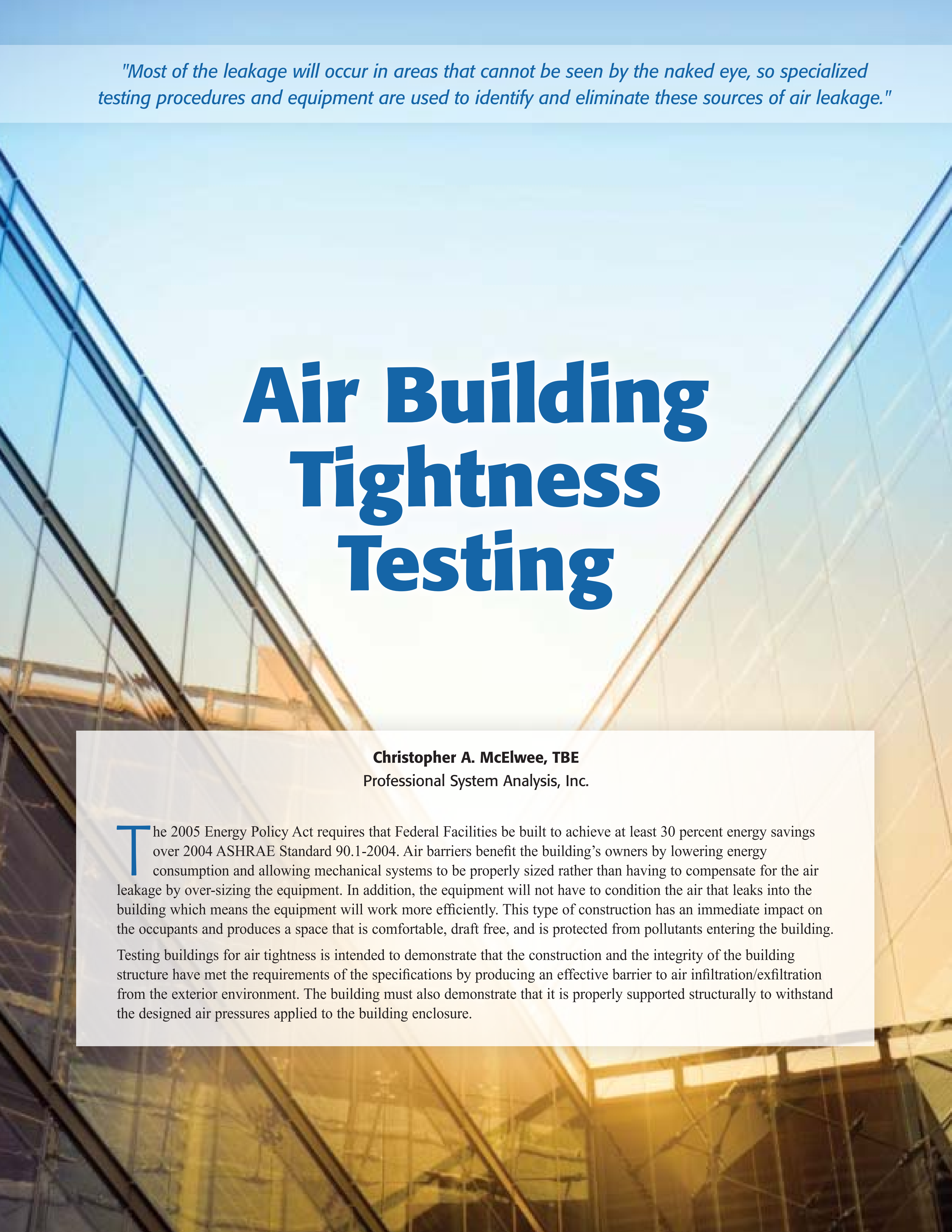
The pump was reassembled and retested, resulting once again with a nonrelating pump curve. Four experienced individuals assigned on site had no explanation. Therefore, a replacement pump was obtained, installed and then retested. The results of the dead head were proper in relation to the manufacturer. No explanation as to why the first pump would not plot correctly was ever found.

Most recently, a local manufacturer/supplier of packaged pumping systems (during a seminar), stated that they test all pumps for actual performance prior to shipment. With this statement, the question was presented: Do you ever find pumps that will not perform to the manufacturer's nameplate data for capacity relating to the pump curve? If so, what is the solution?

The answer from the local manufacturer was that they have about one out of one hundred pumps that, when tested, do not relate to the pump curve when dead headed nor follow the performance curve when having the correct impeller. They test approximately 1,200 pumps per year, prior to shipment.

What they have found as the problem are imperfections in the pump casting, which causes water turbulence producing misleading pump differential readings. The solution is to grind out the casting imperfections, which corrects this condition.

It is hoped that anyone confronted with similar situations to these two issues will find some helpful information to consider from this article.



*"Most of the leakage will occur in areas that cannot be seen by the naked eye, so specialized testing procedures and equipment are used to identify and eliminate these sources of air leakage."*

# Air Building Tightness Testing

**Christopher A. McElwee, TBE**
Professional System Analysis, Inc.

The 2005 Energy Policy Act requires that Federal Facilities be built to achieve at least 30 percent energy savings over 2004 ASHRAE Standard 90.1-2004. Air barriers benefit the building's owners by lowering energy consumption and allowing mechanical systems to be properly sized rather than having to compensate for the air leakage by over-sizing the equipment. In addition, the equipment will not have to condition the air that leaks into the building which means the equipment will work more efficiently. This type of construction has an immediate impact on the occupants and produces a space that is comfortable, draft free, and is protected from pollutants entering the building.

Testing buildings for air tightness is intended to demonstrate that the construction and the integrity of the building structure have met the requirements of the specifications by producing an effective barrier to air infiltration/exfiltration from the exterior environment. The building must also demonstrate that it is properly supported structurally to withstand the designed air pressures applied to the building enclosure.

The major factor in achieving an effective barrier is stopping air leakage through the building envelope. The building needs to be designed with a continuous air barrier to control the air leakage into or out of the conditioned space. This means special consideration needs to be applied where the floors meet the walls, the walls meet the roofs or ceilings, door assemblies, window assemblies, overhangs, joints, junctures, and transitions. Consideration must also be given to all piping and venting entering the building. The air barrier must be structurally supported to withstand positive and negative air pressures applied to the building.

Most of the leakage will occur in areas that cannot be seen by the naked eye, so specialized testing procedures and equipment are used to identify and eliminate these sources of air leakage.

Most specifications have detailed test procedures that specify the testing apparatus, equipment, testing methods, procedures and analysis of the data.

There are two major components to this testing: thermography and pressure testing.

If there is a difference of more than 10% between two tests at the same pressure, either positive or negative, the cause will need to be identified and the problem rectified.

## Thermography

The first test is thermography. Once the building has been deemed ready for air tightness testing, an infrared camera should be brought in to test the entire envelope. Most specifications call for a camera that has a resolution of 0.1° C. The emphasis for this testing is to determine if insulation is missing, and if the installation of these components has been done properly. Even when all the proper procedures for installation have been followed, the camera will show many areas that need to be caulked, sealed, and fixed before the pressure testing can begin.

Once the thermography testing has been completed and repairs have been made, it is time set the building up to do the pressure testing.

## Pressure Testing

The following must be completed before testing of the building envelope commences.

The building envelope must be enclosed. All exterior doors and windows shall be closed with the exception of the mechanical room door to the outside which shall be open. All interior doors including stairway doors shall be open with the exception of any mechanical room doors to the inside which shall be closed. The mechanical room doors are closed to inside adjacent spaces because inherently mechanical rooms, boiler rooms, etc… are deigned to be negative to adjoining spaces. The outside door to the mechanical room is propped open during the testing to ensure that the mechanical room pressure is equal to the outside. This allows an effective test of the mechanical room barrier to the internal space.

Ensure that all combustion appliances and exhaust fans are de-energized so they do not start up during testing. Ensure that all paths of air infiltration or exfiltration have been sealed airtight. This includes all joints, junctures, and transitions between the foundation, walls, windows, piping, ductwork, exhaust and outside air ducts, and any other roof penetrations have been sealed airtight. Finally, all plumbing traps shall be filled with water.

Once the building is setup, it is ready to be pressure tested. This is done in two steps, with a positive and a negative pressure test. During the positive pressure test, the air blower fans supply air into the building until a predetermined positive pressure has been reached, (usually 75 pa). During the negative test, the air blower fans exhaust air out of the building until a predetermined negative pressure has been reached (usually 75 pa).

When doing either test, the initial bias pressure of the building needs to be recorded after the building has been set up. ASHRAE and the Canadian Building Code use testing points up to 75 pa. The range should be from 75 pa to 15 pa. The pressures should be recorded for a minimum of five and up to twelve steps, depending on the standards being used.

Once this work has been completed, the two tests need to be compared and analyzed. If there is a difference of more than 10% between the two tests at the same pressure, either positive or negative, the cause will need to be identified and the problem rectified. Once again the use of the infrared camera may be necessary. While the pressurization fans are running and holding pressure, the camera should be used to find the problem areas. If there is no camera, fog agents should be used.

Once the sources of leakage are identified and corrected additional pressure testing will need to be done to verify that the problems have been resolved. Once the tests are within 10% of each other they are then averaged to eliminate bias pressure in the building. Bias pressures are pressures inside the building caused by stack, wind, flues, or HVAC systems.

*References:*

*CAN/CGSB-149.15-96, Determination of the Overall Envelope Airtightness of Buildings by the Fan Pressurization Method Using the Building's Air-Handling Systems, Ottawa, Canadian General Standards Board, as modified by Amendment No. 1, April 1999, Canadian General Standards Board, Ottawa, Canada*

*ASTM E 779-03 Standard Test Method for Determining Air Leakage Rate by Fan Pressurization Method, West Conshohocken, PA, American Society For Testing and Materials*

*U.S. ARMY CORPS OF ENGINEERS AIR LEAKAGE TEST PROTOCOL FOR*

*MEASURING AIR LEAKAGE IN BUILDINGS Alexander Zhivov, David Bailey, and Dale Herron Construction Engineering Research Laboratory (CERL) U.S. Army Engineer Research and Development Center*

*ASTM C1060 - 90(2003) Standard Practice for Thermographic Inspection of Insulation Installations in Envelope Cavities of Frame Buildings*

*ASTM E1827 - 96(2007) Standard Test Methods for Determining Airtightness of Buildings Using an Orifice Blower Door*

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Ellicott City, Maryland
(800) 535-5594

Baltimore Air Balance Company
Bowie, Maryland
(301) 262-2705

Baumgartner, Inc.
Hunt Valley, Maryland
(410) 785-1720

Baumgartner, Inc.
Easton, Maryland
(410) 770-9277

Chesapeake Testing & Balancing Engineers, Inc.
Easton, Maryland
(410) 820-9791

Environmental Balancing Corporation
Clinton, Maryland
(301) 868-6334

Protab Inc.
Hampstead, Maryland
(410) 935-8249

Test & Balancing, Inc.
Laurel, Maryland
(301) 953-0120

Weisman, Inc.
Towson, Maryland
(410) 296-9070

## MASSACHUSETTS

Thomas-Young Associates, Inc.
Marion, Massachusetts
(508) 748-0204

## MICHIGAN

Aerodynamics Inspecting Company
Dearborn, Michigan
(313) 584-7450

Airflow Testing, Inc.
Lincoln Park, Michigan
(313) 382-TEST

## MINNESOTA

Air Systems Engineering, Inc.
Minnetonka, Minnesota
(952) 807-6744

Mechanical Data Corporation
Bloomington, Minnesota
(952) 473-1176

Mechanical Test and Balance Corporation
Maple Plain, Minnesota
(763) 479-6300

SMB of Minnesota
Center City, Minnesota
(651) 257-7380

## MISSISSIPPI

Capital Air Balance, Inc.
Terry, Mississippi
(601) 878-6701

Coastal Air Balance, Corp.
Terry, Mississippi
(228) 392-8768

## MISSOURI

Envirosystem Analysis, Inc.
St. Charles, Missouri
(636) 661-5252

Miller Certified Air, Inc.
St. Louis, Missouri
(314) 352-8981

Precisionaire of the Midwest, Inc.
Grain Valley, Missouri
(816) 847-1380

Senco Services Corporation
St. Louis, Missouri
(314) 432-5100

Testing & Balance Co. of the Ozarks, LLC (TABCO)
Ozark, Missouri
(417) 443-4430

## NEVADA

American Air Balance Co., Inc.
Las Vegas, Nevada
(702) 255-7331

Environmental Testing and Balancing of Nevada, Inc.
Las Vegas, Nevada
(702) 227-6950

Mechanical Test & Balance Corporation
Las Vegas, Nevada
(702) 737-3030

National Air Balance Company, Inc.
(702) 871-2600

Penn Air Control, Inc.
Las Vegas, Nevada
(702) 221-9877

Raglen System Balance, Inc.
Reno, Nevada
(775) 747-0100

RSAnalysis, Inc.
Las Vegas, Nevada
(702) 740-5537

RSAnalysis, Inc.
Reno, Nevada
(775) 323-8866

## NEW JERSEY

Effective Air Balance, Inc.
Totowa, New Jersey
(973) 790-6748

National Air Balance Co., LLC.
Paramus, New Jersey
(201) 444-8777

## NEW YORK

Air Conditioning Test & Balance Co.
Great Neck, New York
(516) 487-6724

Mechanical Testing, Inc.
Waterford, New York
(518) 328-0440

Precision Testing & Balancing, Inc.
Brooklyn, New York
(718) 994-2300

## NORTH CAROLINA

Air Balance Corporation
Greensboro, North Carolina
(336) 275-6678

Building Environmental Systems Testing, Inc.
Wilson, North Carolina
(252) 291-5100

e-n Tech Independent Testing Services, Inc.
Winston-Salem, NC
(336) 896-0090

Palmetto Air and Water Balance
Asheville, North Carolina
(828) 277-2256

Palmetto Air and Water Balance
Charlotte, North Carolina
(704) 587-7073

Palmetto Air and Water Balance
Raleigh, North Carolina
(919) 460-7730

The Phoenix Agency of North Carolina, Inc.
Winston-Salem, North Carolina
(336) 744-1998

## NORTH DAKOTA

Design Control, Inc.
Fargo, ND
(701) 237-3037

## OHIO

Air Balance Unlimited, Inc.
Gahanna, Ohio
(614) 595-9619

Heat Transfarr, Inc.
Columbus, Ohio
(614) 262-6093

Kahoe Air Balance Company
Cleveland, Ohio
(440) 946-4300

Kahoe Air Balance Company
Cincinatti/Dayton, Ohio
(513) 248-4141

Kahoe Air Balance Company
Columbus, Ohio
(740) 548-7411

PBC, Inc. (Professional Balance Company)
Willoughby, Ohio
(440) 975-9494

Precision Air Balance Company, Inc.
Cleveland, OH
(216) 362-7727

R.H. Cochran and Associates, Inc.
Wickliffe, Ohio
(440) 585-5940

### OKLAHOMA

Eagle Test & Balance Company
Cushing, Oklahoma
(918) 225-1668

### OREGON

Pacific Coast Air Balance Co.
Newberg, Oregon
(503) 537-0826

### PENNSYLVANIA

Butler Balancing Company
Thorndale, Pennsylvania
(610) 873-6905

Flood & Sterling, Inc.
New Cumberland, Pennsylvania
(717) 232-0529

Kahoe Air Balance Company
Pittsburgh, Pennsylvania
(724) 941-3335

WAE Balancing, Inc.
Mercer, Pennsylvania
(724) 662-5743

### PUERTO RICO

Penn Air Control, Inc.
Naguabo, Puerto Rico
(787) 504-8118

### SOUTH CAROLINA

Palmetto Air and Water Balance
Greenville, South Carolina
(864) 877-6832

### TENNESSEE

Environmental Test & Balance Company
Memphis, Tennessee
(901) 373-9946

Systems Analysis, Inc.
Hermitage, Tennessee
(615) 883-9199

United Testing & Balancing, Inc.
Nashville, Tennessee
(615) 331-1294

United Testing & Balancing, Inc.
Knoxville, Tennessee
(865) 922-5754

### TEXAS

Aerodynamics Inspecting Company
Weslaco, Texas
(956) 351-5285

Air Balancing Company, Ltd.
Fort Worth, Texas
(817) 572-6994

AIR Engineering and Testing, Inc.
Dallas, Texas
(972) 386-0144

Austin Air Balancing Corporation
Austin, Texas
(512) 477-7247

Delta-T, Ltd.
Garland, Texas
(972) 494-2300

Engineered Air Balance Co., Inc.
Addison, Texas
(972) 818-9000

Engineered Air Balance Co., Inc.
San Antonio, Texas
(210) 736-9494

Engineered Air Balance Co., Inc.
Spring, Texas
(281) 873-7084

Online Air Balancing Company
Houston, TX
(713) 453-5497

PHI Service Agency, Inc.
San Antonio, Texas
(210) 224-1665

PHI Service Agency, Inc.
Austin, Texas
(512) 339-4757

PHI Service Agency, Inc.
Alamo, Texas
(956) 781-9998

PHI Service Agency, Inc.
Corpus Christi, Texas
(361) 248-4861

Professional Balancing Services, Inc.
Dallas, Texas
(214) 349-4644

TAB Solutions, Inc.
Lakeway, TX
(720) 220-1062

Technical Air Balance, Inc.
Spring, Texas
(281) 651-1844

Texas Precisionaire, Inc.
Houston, Texas
(281) 449-0961

Texas Test and Balance
Houston, Texas
(281) 358-2118

### UTAH

RSAnalysis, Inc.
Sandy, Utah
(801) 255-5015

### VIRGINIA

Arian Tab Services, Inc.
Herndon, Virginia
(703) 319-1000

C&W-TESCO, Inc.
Richmond, Virginia
(804) 379-9345

Mid-Atlantic Test and Balance, Inc.
South Boston, Virginia
(434) 572-4025

### WASHINGTON

Eagle Test & Balance Company
Bellevue, Washington
(425) 747-9256

Penn Air Control, Inc.
Vancouver, Washington
(253) 472-6804

TAC Systems, LLC
Blaine, Washington
(360) 332-4789

### WISCONSIN

Professional System Analysis, Inc.
Germantown, Wisconsin
(262) 253-4146

## AABC CANADIAN CHAPTER

### MANITOBA

A.H.S. Testing & Balancing Ltd.
Winnipeg, Manitoba
(204) 224-1416

Air Movement Services Ltd.
Winnipeg, Manitoba
(204) 233-7456

AIRDRONICS, Inc.
Winnipeg, Manitoba
(204) 253-6647

D.F.C. Mechanical Testing & Balancing Ltd.
Winnipeg, Manitoba
(204) 694-4901

### NEW BRUNSWICK

Controlled Air Management Ltd.
Moncton, New Brunswick
(506) 852-3529

Scan Air Balance 1998 Ltd.
Moncton, New Brunswick
(506) 857-9100

Source Management Limited
Fredericton, New Brunswick
(506) 443-9803

### NOVA SCOTIA

Griffin Air Balance Ltd.
Dartmouth, Nova Scotia
(902) 434-1084

Scotia Air Balance 1996 Limited
Antigonish Co., Nova Scotia
(902) 232-2491

### ONTARIO

Accu-Air Balance Co. (1991) Inc.
Windsor, Ontario
(519) 256-4543

Airwaso Canada Inc.
London, Ontario
(519) 652-4040

Caltab Air Balance Inc.
Tecumseh, Ontario
(519) 259-1581

Designtest & Balance Co. Ltd.
Richmond Hill, Ontario
(905) 886-6513

Dynamic Flow Balancing Ltd.
Oakville, Ontario
(905) 338-0808

Kanata Air Balancing & Engineering Services
Ottawa, Ontario
(613) 592-4991

Pro-Air Testing Inc.
Toronto, Ontario
(416) 252-3232

Vital-Canada Group Inc.
Mississauga, Ontario
(905) 848-1000

VPG Associates Limited
King City, Ontario
(905) 833-4334

### QUEBEC

Groupe Danco TeleVac Inc.
Sherbrooke, Quebec
(819) 823-2092

## AABC INTERNATIONAL MEMBERS

### SOUTH KOREA

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82-2-408-2114

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