TAB Journal

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Understanding the Effects of Glycol on Hydronic Systems

- Environment & Instrument Readings
- Maintaining Minimum OAR
- Design & Plan Review for TAB

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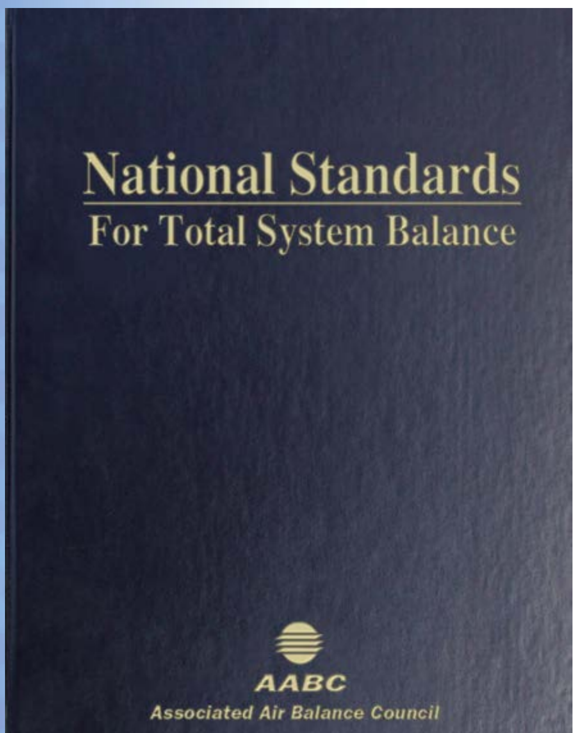

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TAB Journal

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TAB Journal is published quarterly by the Associated Air Balance Council. It is distributed free to AABC members and by subscription to non-members at $24 per year. *TAB Journal* is an open forum for the free expression of opinions and information. The views expressed are not necessarily those of AABC, its officers, directors, or staff. Letters, manuscripts, and other submissions are welcome. However, *TAB Journal* accepts no responsibility for unsolicited material.

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From the Publisher

The Summer 2013 issue of *TAB Journal* covers a variety of topics from the test and balance field. The cover story, from Joseph Hardy, TBE, of Augusta Air Balance Co., LLC, investigates the benefits of adding glycol to a hydronic system.

Sean Bunting, TBE, of National Air Balance Company, Inc., explores the effects that environmental factors such as air temperature and elevation have on instrument readings.

Kit Brockles, TBE, of Engineered Air Balance Co., Inc., looks at the complications that arise when trying to incorporate an indoor air quality plan into a construction schedule.

Thomas Bunner, TBE, of Palmetto Air & Water Balance, Inc., examines common strategies for maintaining minimum outdoor airflow rate.

Henry Long, TBE, offers some pointers for writing an effective design review, which can help identify omissions or deficiencies before construction begins.

Kahoe Air Balance Company's Robert Severin, TBE, gives us the latest installment of Tech Tips, in which he discusses what certain noises can tell a TAB agent about how a system is performing.

And last but never least, this issue's Tech Talk answers questions regarding the best way to balance grilles in multi-family dwellings with a low design CFM per space.

Understanding the Effects

Installations that introduce glycol to condenser water to help prevent freezing might still encounter problems, so one must question the effect glycol has on design, operation, and balancing.

# of Glycol on Hydronic Systems

Joseph K. Hardy, TBE, *Augusta Air Balance Co., LLC*

The higher efficiency ratings of water source heat pump systems are making them an increasingly popular choice for HVAC installations. Some of these installations utilize cooling towers/air coolers for heat rejection. Since these equipment items are exposed to freezing conditions in the winter in some regions, many installations also introduce glycol to the condenser water loop to help prevent freezing. However, problems can still be encountered, so one must question the effect glycol has on the design, operation, and balancing of a hydronic system.

Why add glycol to a system?

Water in outside piping is vulnerable when exposed to freezing conditions in cold-weather regions. For this reason, equipment and piping systems in those regions often have glycol added to the circulating water to help prevent freezing—just as antifreeze is added to automobile coolant systems. The addition of glycol to water lowers its freezing point and raises its boiling point. It does this by modifying the amount of heat that can be absorbed by the mixture. Ethylene glycol and propylene glycol are the most commonly used antifreeze fluids in piping systems.

Here are some characteristics of these types of glycol. All examples are from the standpoint of a 50% glycol/water mixture at a temperature of 80˚F.

Product	Freeze Temp	Boil Temp	Specific Heat	Specific Gravity
Propylene glycol	-29˚F	222˚F	.850 btu/lb˚F	1.041 at 60˚F
Ethylene glycol	-36˚F	225˚F	.815 btu/lb˚F	1.077
Water	+32˚F	+212˚F	1.00 btu/lb˚F	.998

From these characteristics, the following facts can be understood. More propylene glycol is needed to achieve the same freeze point depression. Ethylene glycol can't carry as much heat as propylene glycol, so more fluid must circulated through the loop to transfer the same energy (pump volume increases). The viscosity (resistance to flow) of both products are higher than that of water. Therefore, friction is increased in piping and pump head.

Because ethylene glycol has a high level of toxicity if ingested, it should never be used in systems where the water might be consumed by animals or in food production facilities where it may become mixed in food processing systems. Propylene glycol has a lower level of toxicity and should therefore be used in these circumstances.

Glycol's Effect on Equipment and Performance

Since the viscosity of glycol is higher and its specific heat ratio is lower than water's, one would expect some changes in pipe sizing. In general, the higher the percentage mixture, the higher the friction loss is in the piping. The following table illustrates the differences in friction loss for a 60 GPM flow of water, 50% solution of ethylene glycol, and 50% propylene glycol at 80˚F through 2" copper piping in a 1,000 equivalent linear foot loop.

Solution	Velocity	Friction Loss	Total Friction
Water	6.28 FPS	6.60 ft/100 lf	66.0 ft
50% Ethylene glycol	6.28 FPS	8.54 ft/100 lf	85.4 ft
50% Propylene glycol	6.28 FPS	9.65 ft/ 100 lf	96.5 ft

(Bell & Gossett "System Syzer" program)

Note that the pipe size in this example does not change to provide the same quantity of fluid at the same velocity. This leads to another question. Given an equal flow rate, what difference does it make in the selection process for the pumps and in their resultant operation?

Looking at the previous chart, one can see that the friction head in a system changes with the addition of a 50% glycol solution—whether ethylene or propylene. Let's look at the pump selections for the same three solutions at the same operating temperature. For now, only friction loss for equal flows will be used. While many pumps are manufactured to successfully move the fluid, this example uses a B&G Series 1510, Base Mounted, End Suction (ESP) Pump operating at 1,750 RPM and 60 Hz. Here are the results:

Selection A: Circulate 60 GPM of clean water at 80˚F through 1,000 equivalent linear feet of 2" piping.

Enter the following pump information in the program: Model 1510, 1,750 RPM, 60 Hz, 60 GPM clean water at 66 ft head at 80˚F. The program advises that a 1-1/4 BC model pump should be used. The pump will operate at 55.43% efficiency and will use 1.83 brake horsepower (BHP), which requires a 3 horsepower (HP) motor. The pump will have an 8.25" diameter impeller. At $.10/kwh power rate and a 365 day-per-year operation, this pump, exclusive of maintenance, is estimated to cost the owner $1,383.23 annually to operate.

Selection B: Circulate 60 GPM of a 50% ethylene glycol solution at 80˚F through 1000 equivalent linear feet of 2" piping. The pump information remains the same (Model 1510, 1-1/4

BC, 1,750 RPM, 60 Hz), but with a total friction for this solution of 85.4 ft. The pump will operate at 54.91% efficiency and require 2.39 BHP and a 5 HP motor. The required impeller diameter now becomes 9.125 inches. Operating under the same conditions as the clean water pump, it will cost the owner an estimated $1,727.02 per year to operate.

Selection C: Circulate 60 gpm of a 50% propylene glycol solution at 80˚F through 1,000 equivalent linear feet of 2" piping.

Again, the same pump information is used (Model 1510 1-1/4 BC, 1,750 RPM, 60 Hz), but with total friction for this solution of 96.5 ft, this pump will operate at 54.3% efficiency and will require 2.71 BHP and a 5 HP pump motor. The impeller diameter required for this solution is 9.5". Under the same conditions as the previous two pumps, this pump will cost the owner $1,958.25 in estimated annual operating costs.

So far, it's clear that glycol affects friction loss in the piping and the resultant change in pump selection. But what about other equipment such as coils? Manufacturers will normally select the equipment coils based upon the load and the operating fluid. But what if, for instance, standard water source heat pumps are used on a project? What might happen in terms of heat transfer and pressure drop across the condenser coils?

For this scenario, there is a 2-ton water source heat pump is used operating at a flow rate of 6 GPM with water and propylene glycol. Using the manufacturer's literature, we can produce the following comparison table:

Fluid	Coil PD Cooling	Coil PD Heating	Ttl Cool (mbtu)	Sens Cool (mbtu)	Ttl Heat (mbtu)
Water	11.7 ft	12.4 ft	25.2	19.91	29.76
30% PropGly	15.1 ft	16.8 ft	24.22	18.38	27.80
50% PropGly	17.4 ft	19.8 ft	23.81	17.78	25.30

The manufacturer's selection program did not include selections based upon ethylene glycol mixtures.

From this chart, it becomes clear that not only is the performance of the unit affected by the use of glycol (cooling up to 6%, heating up to 15%), but also the coil pressure drops increase by as much as 60%.

Glycol and Circuit Setter Readings

Circuit setters are commonly used to balance flow through coils, heat exchangers, and other equipment. How does the introduction of glycol to the system affect their set points? For this example, assume there are ten of the previously referenced water source heat pumps installed on the project. The technician is to balance these and takes preliminary readings with the installed ¾" circuit setters in the wide open position. For the purpose of illustration, he will read the same pressure drop with each fluid. What happens with the resultant flow reading?

Fluid	CS Set Pt	DeltaP	Flow
Water at 80˚F	open	7.3 ft	6.0 GPM
30% PropGly at 80˚F	open	7.3 ft	5.9 GPM
50% PropGly at 80˚F	open	7.3 ft	5.8 GPM

If the technician doesn't happen to have a computer in the field with him, Bell & Gossett offers the following information for use:

GPMf = GPMs*(VC/Sqrt(SG))

GPMf = Corrected Flow

GPMs = Circuit Setter flow

VC = Velocity Correction (from chart)

SG = Specific Gravity of the fluid (obtained in field)

A chart is available showing correction factors for fluids of various specific gravity and viscosities. It should be understood that as the temperature of a fluid increases, its viscosity decreases; in other words, the hotter the fluid, the easier it moves through the pipe circuit. Referring back to our 50% ethylene glycol mixture, the chart below illustrates the viscosity changes at various temperatures:

Temperature	Dynamic Viscosity
40˚F	6.5
80˚F	2.8
120˚F	1.5
160˚F	.95
200˚F	.70

information not only to properly balance a system containing glycol, also to recognize the possible causes of problems that may arise."

Viscosity changes at different operating temperatures causes a change in the friction loss at those same temperatures. The chart below shows the pressure drop correction factors for our 50% ethylene glycol solution at various temperatures:

Temperature	Delta P Correction Factor
40˚F	1.45
100˚F	1.10
140˚F	1.00
180˚F	0.94

Throughout this investigation, the viscosity of the fluid at specific temperature has been the only consideration. And, it has been shown that the pipe friction loss changes due to the type and mixture rate of the anti-freeze products. Heat transfer properties of the various anti-freeze mixtures and the resultant affect on the system(s) should also be considered.

While it is true that TAB technicians do not normally deal with heat exchanger sizing, it is worth analyzing to get a more complete picture of system components and their performance. Continuing to use the 50% ethylene glycol solution, flow rate must be increased to compensate for the differences in heat exchange rates. The following table shows the increased flow requirement for the same heat transfer of a 50% ethylene glycol mixture as compared with water:

Fluid Temp	Flow Increase Needed For 50% Glycol as Compared with Water
40˚F	1.22
100˚F	1.16
140˚F	1.15
180˚F	1.14

For example, if a system is to operate at a mean temperature of 100˚F and requires a flow of 60 GPM when sized using water, then the flow rate needed to adjust for the lower heat transfer rate of the 50% mixture becomes 60 GPM x 1.16, which equals 69.6 GPM. This is the new flow rate for the pump. The pump must overcome the additional friction loss required to move this volume of the 50% mixture through the system. The combined effect of the increased flow rate due to decreased heat transfer rate and increased friction due to the glycol mixture over that of water is:

Fluid Temp	Combined Pressure Drop Correction; 50% Glycol Flow Increase
40˚F	2.14
100˚F	1.49
140˚F	1.32
180˚F	1.23

A TAB technician can use this information not only to properly balance a system containing glycol, but also to recognize the possible causes of problems that may arise.

References:

1. *EngineeringToolbox.com/ethylene-propylene-glycol-d_904.html*
2. *EngineeringToolbox.com/ethylene-glycol-d_146.html*
3. *Bell & Gossett "SystemSyzer" program*
4. *Bell & Gossett "ESP" pump selection program*
5. *Bell & Gossett Instruction Manual G95873(Rev. L), page 4*
6. *Bell & Gossett Bulletin #TEH-176 "Hydronic Systems Anti-Freeze Design", page 15*

CONSIDER THE CLIMATE:
How Environment Affects Instrument Readings

Sean Bunting, TBE
National Air Balance Company, Inc.

In today's hi-tech world, one would hope that an instrument would yield an accurate measurement every time. Unfortunately, this is not the case. Instruments used in the test and balance industry are calibrated under standard air conditions (70ºF, 29.92", 0.075 lbs/ft^3). If the airflow being measured, however, differs from standard air conditions, the instrument reading must be corrected prior to being documented.

Factors that change the condition of air being measured can range from whether one is in a coastal or mountain region, whether it's summer or winter, or whether the system is on full cooling or full heat. Therefore, referencing temperatures of the air being measured and having knowledge of the elevation of the city in which you are performing your test and balance work becomes important for the documented CFM values to be accurate.

Formulas:

Velocity at standard air

$$V_1 = \sqrt{VP_1 \text{ x } 4005} \qquad V1 = (VP_1)^{1/2} \text{ x } 4005$$

V_1 = Velocity (FPM) measured
VP_1 = Velocity pressure measured

Velocity at non-standard air

$$V1 = \sqrt{\frac{VP1 \text{ x } 1096.7}{\text{Density}}} \qquad V1 = (VP_1 \,/\, \text{density})^{1/2} \text{ x } 1096.7$$

V_1 = Velocity (FPM) measured
VP_1 = Velocity pressure measured
Density = Calculated density of the air being measured

Density of air

$$\textbf{Density} = 0.075 \text{ x } \frac{530}{460 + T} \text{ x } \frac{\text{BAR. PR.}}{29.2}$$

T = Actual temperature of air being measured °F
BAR. PR. = Actual atmosphere pressure in inches of mercury

Examples:

1. A pitot traverse reading was taken at standard air. The velocity pressure readings on Instrument = 0.2".

$V_1 = 0.2 \text{ x } 4005 \quad V1 = (0.2)^{1/2} \text{ x } 4005$
$V_1 = 1791$

2. A pitot traverse reading was taken at non-standard air conditions. The temperature of the air being measured = 155ºF. The elevation of the city which work is being done in = 5,000'. The standard atmosphere pressure at 5000' = 24.9"HG. The velocity pressure readings on the instrument = 0.2".

$$\textbf{Density} = 0.075 \text{ x } \frac{530}{460 + 155} \text{ x } \frac{24.9}{29.92}$$

Density = 0.054

$$V1 = \sqrt{\frac{0.2}{0.54}} \text{ x } 1096.7$$

$V1 = (0.2 \,/\, 0.054)^{1/2} \text{ x } 1096.7$
$V_1 = 2111$

Example 1 = 1,791 FPM
Example 2 = 2,111 FPM

This is a 17% difference. If the instrument reading in example 2 was not corrected, the documented value for tested FPM would be incorrect.
As you can see from this comparison, correcting your instrument readings for temperature and or elevation becomes an important step in producing an accurate and repeatable test and balance report.

The IAQ Plan: An Underrated Essential

Kit Brockles, TBE
Engineered Air Balance Co., Inc.

Maintaining proper indoor air quality (IAQ) during construction is becoming more important, particularly in healthcare and education facilities. The construction industry knows there can be an impact on a project schedule due to maintaining a higher level of air quality. What the industry still seems to be coming to terms with is that an IAQ Plan is nonetheless a critical component of the project.

Unfortunately, the trades most affected by the IAQ Plan are often excluded from the creation of a project schedule. It is hard to believe there is a way to install and program the direct digital control systems and sequences of operation before half of an air handling unit system is energized, let alone completely tested, adjusted and balanced, and commissioned.

Furthermore, owners and contractors can be so focused on the time required at the end of a project to get all the necessary tasks accomplished that properly maintaining good indoor air quality is essentially cast aside. On some sites, for example, return air systems are energized and started up the day before health department inspections or substantial completion. In other cases, efforts such as putting filters on return outlets, keeping the proper filtration at the air handing units, covering return air paths and keeping a job site clean are ignored when trades are on top of each other at the end of a project trying to complete their work.

But ignoring the IAQ Plan can have consequences that negatively impact the project as well. For example, painters working in the final stages during the return air balance compromise the cleanliness of the return system. Moreover, various trades will undoubtedly create construction dust to complete their scheduled tasks, which obviously affects air quality. It would be better to complete these activities before any of the return air balance work begins.

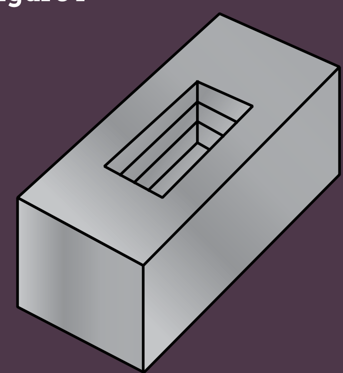
It is also not uncommon to see filter media covering the return air grilles in the construction area with the adhesive side of the media facing the grille. When the filters get dirty, new ones are installed in the same way. At the end of the project when the filters are finally removed, there is a layer of adhesive left on the grilles from multiple filters. Remaining particles in the air then get stuck to the adhesive on the grille's surface, which cause a buildup of dust and debris that affects IAQ.

Another potential issue is that outside air is commonly used throughout a project in an effort to create positive pressure on the building. Air handling units with economizer packages are placed in 100% outside air operation to accomplish this. While this practice continually flushes the building of contaminants created during construction and prevents dust and debris from entering the facility, without utilizing the return ductwork to relieve the inside air to keep it clean, the building's pressure becomes excessively positive. This can cause problems when buildings are turned over to the owner in phases.

Let's look at an example of this. Figures 1 and 2 show a three-story building with an atrium in the center. This atrium made all three floors common to one another. The owner planned to take over half of the first floor before the rest of the building was complete to use as an operating room suite. Operating suites are very pressure sensitive areas. The outside air serving the suite is controlled by tracking the return airflow at a specified amount below the supply airflow. The suite was set up as positive.

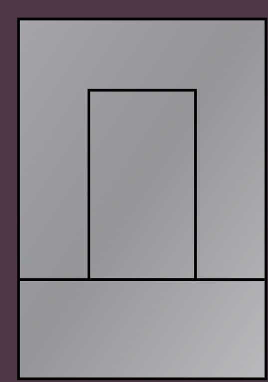
Unfortunately, the rest of the building was still being served by 100% outside air with no pressure relief. The remaining portion that was under construction was positive relative to the operating suite during normal operation. Temporary walls and a vestibule had to be erected to prevent cross contamination between the construction area and the operating suite.

Figure 1



Building without the roof

Figure 2



Floor Plan

To conclude, the important role IAQ plans play in the construction process should not be underestimated. Disregarding them can lead not only to compromised air quality but also to problematic scenarios for the building owner once he takes occupancy.

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Strategies for Maintaining Minimum Outside Airflow Rates

Thomas Bunner, Jr., TBE, *Palmetto Air & Water Balance, Inc.*

The overview to follow, examines common control strategies in variable airflow systems equipped with both supply and return fans, with the intent to maintain the minimum outside airflow rate. Three common control strategies that the testing, adjusting and balancing industry commonly faces when balancing variable volume systems equipped with return fans will be examined. For the sake of continuity, each strategy to be discussed will be based on the following air handling unit (AHU) example. AHU supply, return and minimum outside airflow rates are based on all terminal units being at maximum cooling set point.

AHU-1:

- Supply airflow rate = 10,000 CFM with supply fan speed operating at 1,800 RPM (60 Hz)
- Return airflow rate = 8,000 CFM with return fan speed operating at 1,200 RPM (60 Hz)
- Minimum outside airflow rate = 2,000 CFM
- Relief airflow rate = 0 CFM.
- Damper positions: Outside air = 25% open, Mixed air = 75% open, Relief damper = 0% open

(Note all damper positions are assumed constant for this paper)

- Supply duct static pressure set point = 1.5"
- Return duct static pressure = -0.20"
- Return fan discharge pressure = 0.01"
- Mixed air plenum = -0.10"
- Building pressure = 0.005"

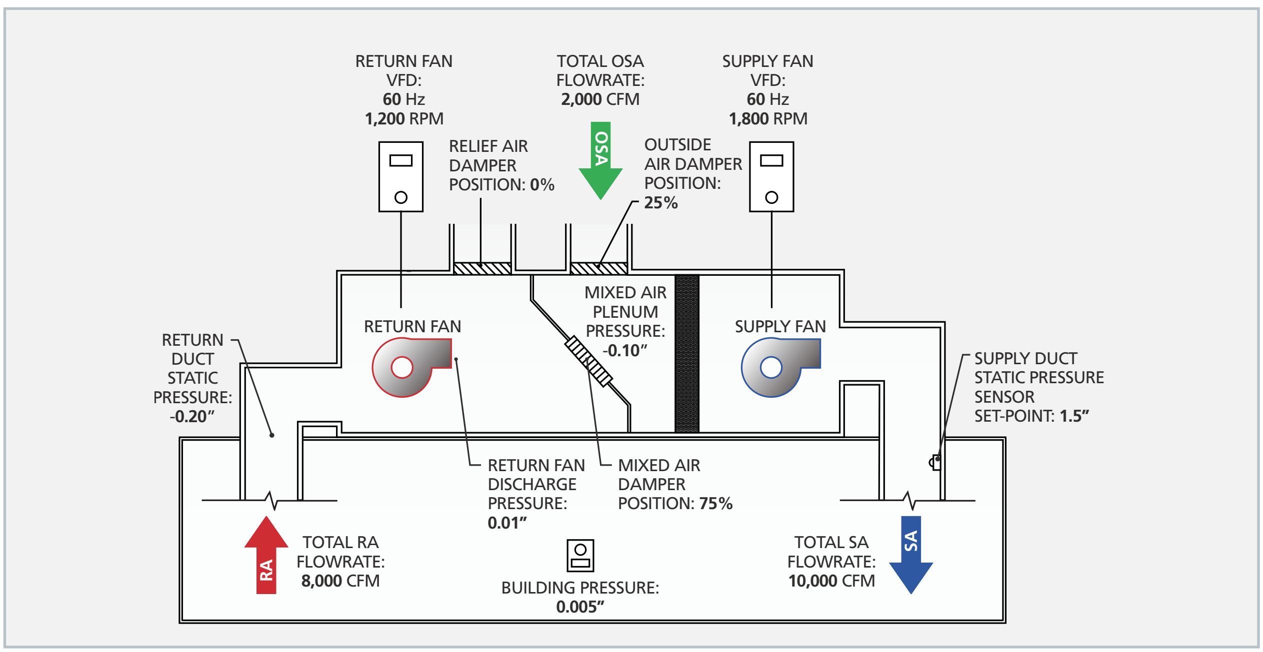

Figure 1: AHU-1 Baseline Readings

The first strategy to be examined is tracking the return fan to a set speed that references the supply fan speed. For example, if the supply fan variable frequency drive (VFD) output slows to 50 Hz, the return fan VFD output will likewise slow to 50 Hz. In this control strategy, assuming that both the supply fan and return fan are at design quantities at 60 Hz, the intent is to maintain the outside airflow minimum based on the fan laws. It is also common for the controls sequence to specify that the return fan VFD is to track a set percentage less than the supply fan VFD. The specifications usually allow for this point to be adjustable. It is, however, recommended that both the supply and return fans be set to design CFM per AABC standards at 60 Hz.

The problem with this strategy in a variable airflow system is that the supply fan does not follow the fan laws when the terminal units begin to modulate between there minimum and maximum airflow set points. The return fan however does have the capability of following the fan law as it is constant volume unit. This results in a lack of control of the minimum outside airflow rate as the fans' speeds modulate. Below is a scenario illustrating this strategy and its potential problem resulting from its use.

Scenario #1: Once testing is completed, the AHU is released to automatic control. The supply airflow rate and speed reduces to 7,500 CFM at 1,350 RPM with the VFD at 45 Hz. The constant volume return fan speed reduces to 900 RPM with the VFD at 45 Hz. The problem arises in that the constant volume return fan airflow rate using the fan laws calculates to 6,000 CFM. The net result is 1,500 CFM of outside air which is 75% of the outside minimum set point of 2,000 CFM. Therefore, Because this control strategy does not recognize the variable airflow rates of the supply side, it should only be used if systems are constant volume.

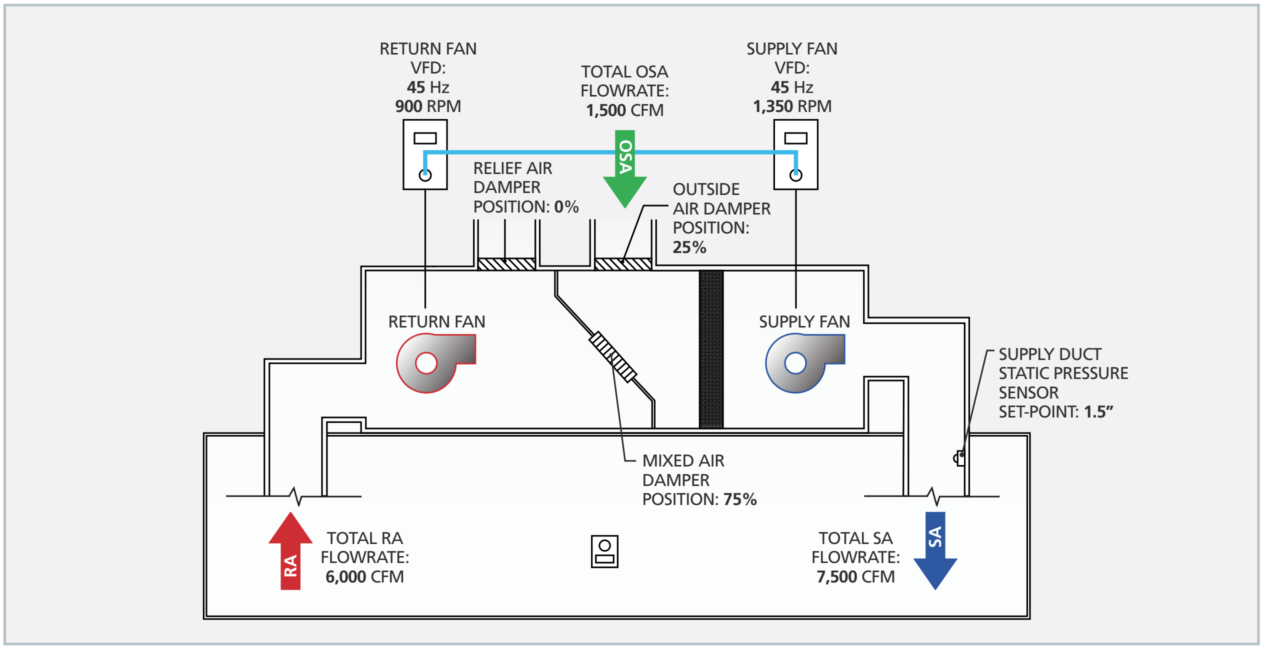

Figure 2: AHU-1 Scenario #1

Another possible strategy is one that controls the return fan speed independently of the supply fan. In this case, the return fan speed examines building pressure and the VFD will modulate to control it—regardless of where the supply fan is—with the intent of always maintaining a neutral or slightly positive building pressure. Several problems may arise with this strategy, as many factors affect building pressure. For example:

Building pressure is not only dependent upon building design, outside air and exhaust air quantities, but also air tightness of the building.

Depending on building and AHU size, the airflow rate changes in the AHU system may not be reflected in the building pressure.

The control sequence of mixed air, outside air and relief damper may affect the outside air quantities.

The building pressure may be more or less positive than desired with the outside air at minimum.

The building pressure may be such a low value, that it is not affected by system airflow changes. See the scenario below depicting how this may play out.

Scenario #2: Once again, when testing is completed, the AHU is released to automatic control. The supply airflow rate reduces to 7,500 CFM at 1,350 RPM with the VFD at 45 Hz. The potential problem arises is when the VAV boxes begin to close and the building pressure does not change. The return fan VFD will not necessarily change the return fan speed (due to the factors stated above regarding building pressure), which will reduce the minimum outside air to the building that is introduced through the AHU. So, this control strategy involves many factors that must all come together to effectively control as intended.

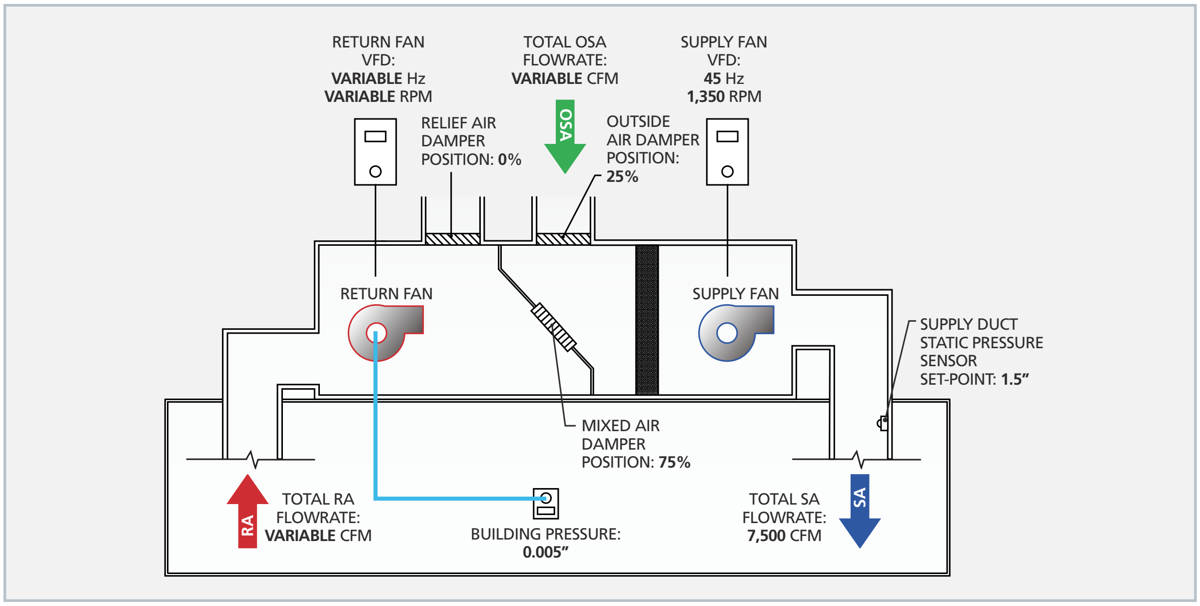

Figure 3: AHU-1 Scenario #2

The final strategy controls the supply fan, return fan and outside airflow rates by employing calibrated airflow stations. The advantage of this method is that the VFDs can be set up via the building automated system to track to the airflow stations to ensure they constantly maintain the minimum offset. Therefore, no matter what the supply airflow rate may be at any given time, the return fan tracking is set up to ensure minimum outside airflow rate is constant. Another advantage is that installing an additional airflow monitor in the outside air can serve as check even if it is not required. The outdoor airflow monitor should equal the supply airflow monitor minus the return airflow monitor quantity.

Unfortunately, there is no perfect strategy and this one has its flaws as well. The first disadvantage is that the locations for the air monitors may not be ideal, which will affect their ability to accurately track the airflow rates. Due to this fact, caution should be used to ensure that the airflow station is calibrated at different airflow rates and damper positions. Another disadvantage is the cost, which not only includes the device itself, but the constant need for recalibration of the airflow monitors to ensure proper operation. See the scenario below for an example:

Scenario #3: Testing completed, the AHU is released to automatic control. The supply airflow rate reduces to 7,500 CFM at 1,350 RPM with the VFD at 45 Hz. As the supply fan speed reduces, the return fan reduces to 5,500 CFM at 825 RPM. The offset is maintained and therefore the outside air minimum is maintained. This can be checked at the outside airflow monitor, if installed, which should read 2,000 CFM. Note only two of the three airflow monitors are absolutely necessary.

This method can be a very accurate as a control strategy if all the required equipment is installed per the manufacturer's requirements for proper operation. Failure to meet these requirements may adversely affect the performance of the airflow monitors, which occurs often because architects and design engineers do not always have the luxury of designing air and water systems where space is not a limitation.

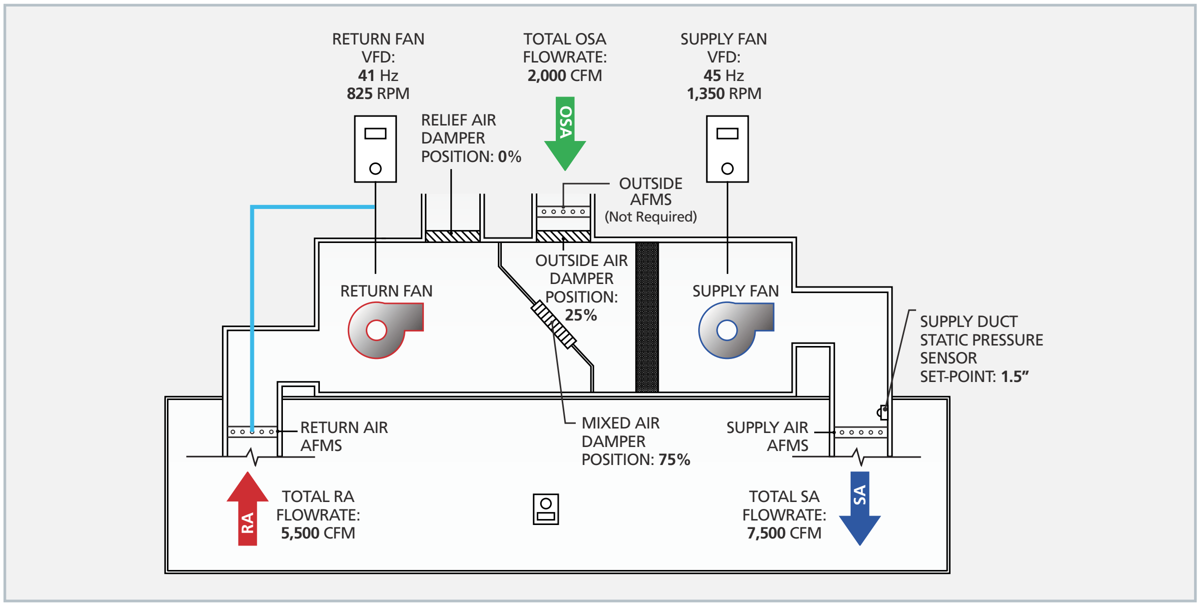

Figure 4: AHU-1 Scenario #3

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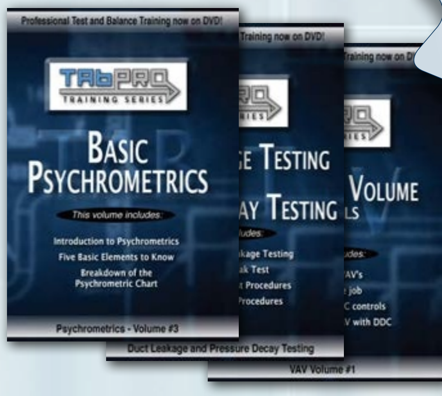

Basic Psychrometrics

DVD format
Run time: 19 minutes
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Member price: $90.00

This volume contains one lesson on basic psychrometrics. This provides the viewer with an introduction to psychrometric fundamentals and takes you through five of the basic elements found on the psychrometric chart. This lesson will break down these elements on the chart and provide fundamental concepts of chart usage.

Duct Leakage and Pressure Decay Testing

DVD format
Run time: 42 minutes
List price: $200.00
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This volume consists of two lessons covering standard duct leakage testing and pressure decay leakage testing. These lessons take the viewer through an introduction to leakage testing, essential job preparation, instrumentation used during testing, general procedures for leakage testing, multiple calculations used during testing and final reporting.

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DVD format
Run time: 45 minutes
List price: $200.00
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Design & Plan Review for TAB

Henry M. Long, TBE
Building Environmental Systems Testing, Inc. (BEST, Inc.)

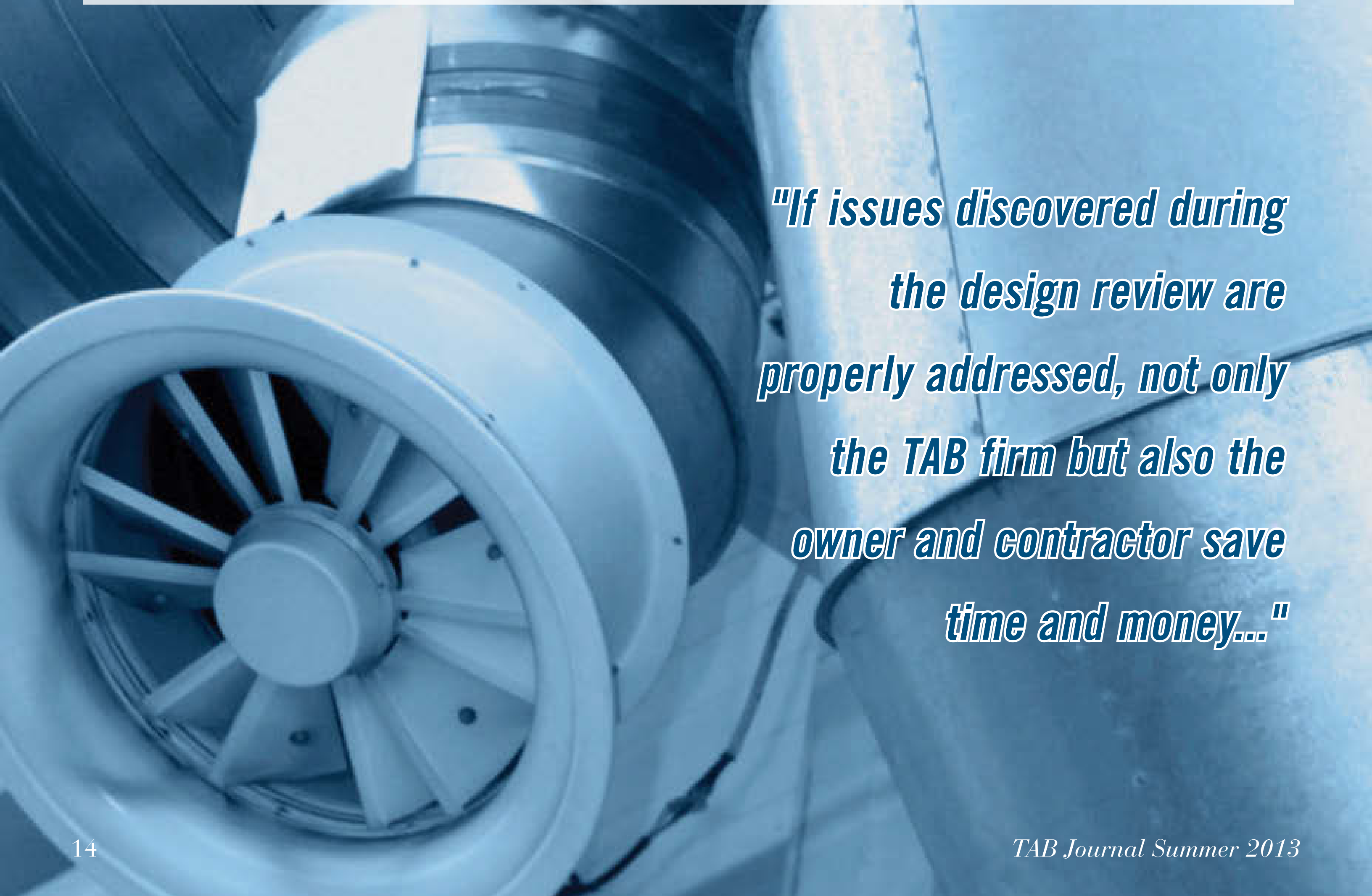
Frequently, specifications require TAB agents to provide a design review report for a project's HVAC systems. Specifications typically state "Submit typed report describing omissions and deficiencies in the HVAC system's design that would preclude the TAB team from accomplishing the TAB work requirement…"

This request often puts the TAB agent in an unfavorable position with the designer as the TBE has no prior knowledge of the owners requirements, basis of design, financial or physical limitations of the project. Additionally, design reviews are often required prior to approved mechanicals submittals further limiting the agent's ability to address the system design.

However, if design data or approved equipment submittals aren't available, one can submit a 'plan review' report. This type of design review would include a review of the *plans* of the air-side and temperature control systems, as well as the hydronic system, if present.

If issues discovered during the design review are properly addressed, they should save not only the TAB firm but the owner and contractor time and money because they won't need to deal with the problems at the end of the project. For this reason, a design review should become part of a TAB agent's pre-project planning and review whether required by specifications or not.

The checklists on the following page should aid the TAB agent in conducting a design review.



"If issues discovered during the design review are properly addressed, not only the TAB firm but also the owner and contractor save time and money..."

AIR-SIDE REVIEW

Ductwork & Air Distributions

- ❑ Are there adequate lengths of straight ductwork to provide traverses for supply/return/outside air?
- ❑ Are volume dampers shown/indicated at all take-offs to inlets/outlets?
- ❑ Do volume dampers have stand-off brackets & handle/operators with locking quadrants?
- ❑ Are volume dampers shown/indicated at all branch or splitter ducts?
- ❑ Are opposed blade dampers specified for ducts 16” or more in depth?
- ❑ Are turning vanes indicated for round and square elbows on supply and return ducts?
- ❑ Are duct runouts to air devices properly sized?
- ❑ Are there ducted outlets connected to a duct sock system?
- ❑ If dampers are shown above a hard ceiling (sheetrock), are access panels provided?

Constant Volume Units

- ❑ Does the sum of the supply outlets equal the total supply air at the unit?
- ❑ Does the sum of the return and outside air equal the supply?
- ❑ How is outside air introduced into the system? For plenum returns, are return dampers provided to balance the outside air?
- ❑ Are bypass ducts (from supply to return) shown on constant volume systems with VAV or other air flow limiting devices?
- ❑ If the unit has economizer operation, can airflow be accurately measured at minimum?
- ❑ Is an airflow monitoring station provided in the outside air duct? Can it be installed per manufacturer specification so that it can be calibrated and used.
- ❑ How does the total building exhaust compare to outside air quantity? Is a Building Air Balance Schedule provided?
- ❑ On multi-zone systems, are branch dampers provided on each zone?

Variable Volume Units

- ❑ Is the sum of the VAV boxes equal to or more than the unit supply CFM (diversity)?
- ❑ If a return fan is provided, how do the return inlets compare to the total return fan CFM?
- ❑ If the unit has a relief CFM, does the return minus the relief plus the outside air equal the supply CFM?
- ❑ How is minimum outside air controlled during changes in supply CFM?
- ❑ Is a sufficient length of straight duct provided, as recommended by the manufacturer, at the entrance to VAV boxes for uniform air flow measurement?
- ❑ With VAV boxes at minimum CFM, how does the total minimum supply CFM compare to the minimum outside air?
- ❑ On systems with series fan powered boxes, how does the fan CFM compare to the primary air?

HYDRONIC REVIEW *(if applicable)*

- ❑ Does the total coil GPM match pump capacity? Does the system have diversity?
- ❑ Are manual balance valves or autoflow valves installed?
- ❑ If manual valves are installed, is there sufficient length of straight pipe upstream of the valve to prevent turbulent flow or accurate measurement?
- ❑ If the distribution system has 2-way temperature control valves (TCC), how does the water get back to the pump if all valves close?
- ❑ If the distribution system has 3-way TCC and manual circuit setters, is a balancing device provided in the by pass leg?
- ❑ Are manual circuit setters sized based on pressure drop and GPM as recommended by manufacturer or are they pipe size?
- ❑ Is the supply and return piping connected together at the end of the loop, floor, or building? If so, is there a means of regulating flow, and is the additional GPM included in the total pump GPM?
- ❑ Are multiple pumps provided? Are pumps to operate in parallel or lead/lag operation?
- ❑ If a primary/secondary pumping system is provided, how does the primary GPM compare to the secondary GPM?
- ❑ Are taps, ports and stops provided on the discharge and suction flanges of the pump?
- ❑ Are pressure and temperature ports provided as close as possible to all coils and heat exchangers?
- ❑ Is a pressure gage installed downstream of the pressure reducing valve for make-up water?

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Tech tips

What's in a Noise?

Robert A. Severin, TBE, Kahoe Air Balance Company

Noise generated by an HVAC system can generally be heard throughout most buildings. From residences to commercial and industrial structures, the HVAC systems create their own particular music, as it were. The whistling of a diffuser, the hammering of a valve, the rumbling of a duct—these and other noises can all be signs of problem within the HVAC system. Knowledgeable TAB agents can utilize particular HVAC noises to assist them in analyzing how a system is performing.

Following is a small sampling and description of various system noises and their possible causes:

Hydronic Valve chatter or hammer: Causes could be excessive system pressure, or possibly a valve installed backwards.

Sound of rushing water: This is generally caused by a restriction of flow, usually at a balancing device that has been throttled too much.

Duct rumbling: Possible cause could be restricted flow by closed dampers, an age-weakened duct, broken duct joints, or excessive airflow through a duct.

Duct oil-canning: This is sometimes tied to system surge, which means the fan is surging across its performance curve due to an unstable operating point.

High-pitched whistle: In air systems, this can be the result of a closed damper, (especially when outlet dampers are used for a primary means of balancing), or excessive airflow at a diffuser relative to it's size. This noise can also be attributed to air leaks in a higher pressure duct system.

Tinny rattling: Usually caused by a loose balancing damper. Sometimes, when single blade dampers are installed in larger ducts they will actually move or vibrate when air is passing across them. This, in turn, may cause the damper handle to rattle. Single blade dampers, in and of themselves, are generally noisier that multi-blade dampers.

Erratic pressure sounds within a duct system: This is different from duct rumble or duct oil-canning. It sounds like the system is breathing. It is actually searching for it's operating pressure point. The cause may be an unstable fan, or a bad, or improperly, located pressure sensor in the control system.

Do you have a "Tech Tip" that you would like to share with our readers? If so, please contact AABC at:

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"One of the clearest descriptions of the commissioning process that we've seen." —Environmental Building News

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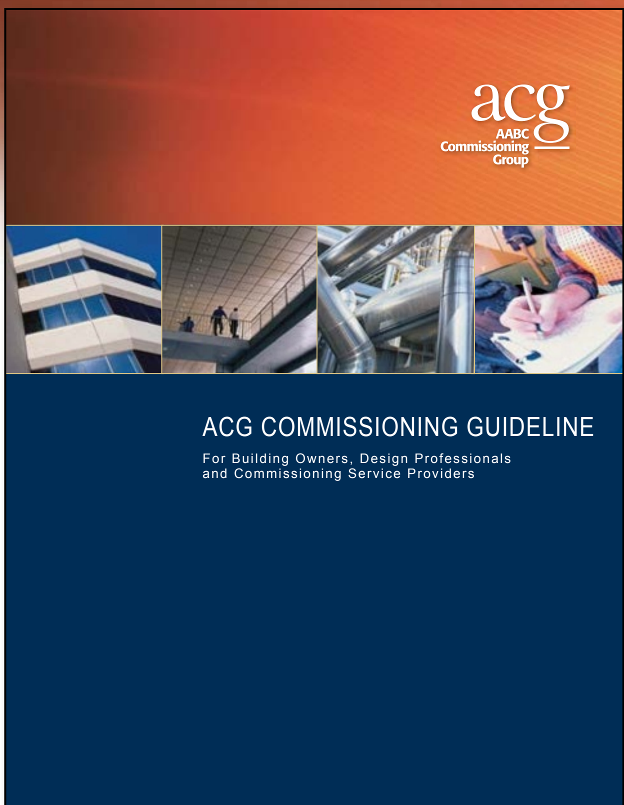

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Tech Talk

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).**

QUESTION: ***According to many of the new ventilation standards and code requirements for multi-family dwellings, a minimum airflow for continuous exhaust is required from each bathroom and from the kitchen. This minimum airflow is as low as 20 CFM per space. Is it practical to ask a balancing contractor to balance an inlet grille or register to such a low air quantity? With such a low air quantity would it be best to use a grille and a damper in the connecting branch duct or a register with an opposed blade damper?***

AABC: It is imperative that the system is balanced to assure the required flow is being exhausted from each dwelling. While I would prefer to see a higher design CFM for a more accurate measurement, 20 CFM is measureable. My preference for the ducting arrangement would be a grille in the space with a duct volume damper at the branch connection.

Registers with opposed blade dampers can be opened by residents which would negate all of the balancing done.

It is likewise imperative that the exhaust system be a sheet metal ducted system from the fan to the inlet terminals. I have seen masonry shafts and drywall shafts employed as ducting systems for risers from the fans with sheet metal run outs on the floors. That type of system is set up for failure due to leakage in the shafts.

Even with the sheet metal ducted system, duct pressure testing should be specified.

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

AABC: I concur. A 4" x 4" grille balanced with a rotating vane would yield accurate and repeatable results at 20 CFM.

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

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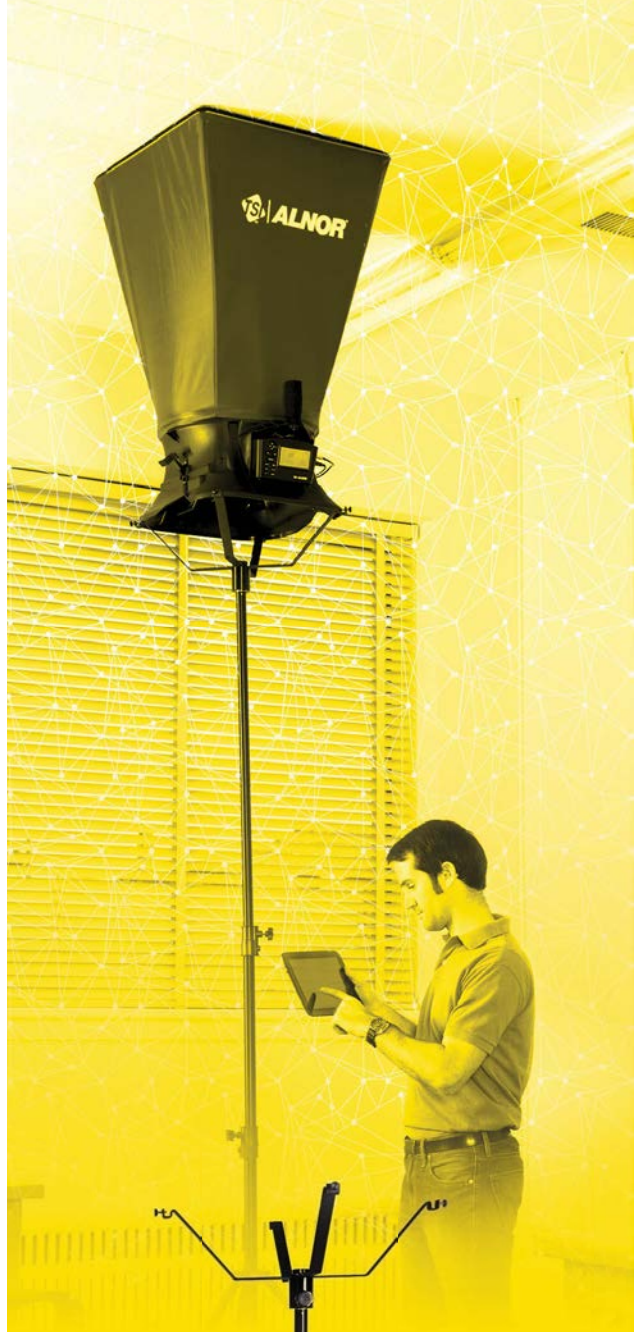

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