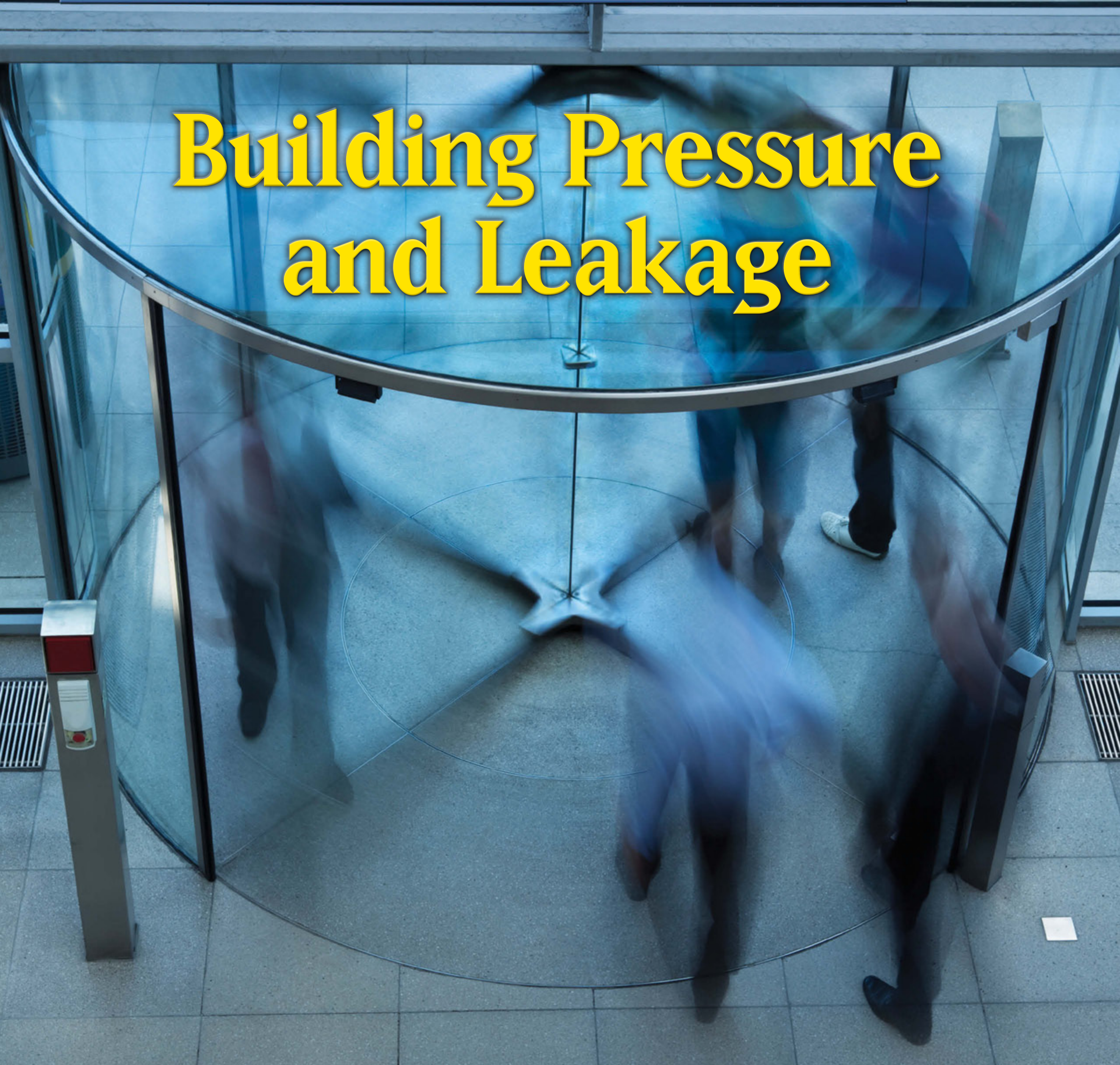
TAB Journal

THE MAGAZINE OF THE ASSOCIATED AIR BALANCE COUNCIL • SUMMER 2012



Building Pressure and Leakage

IN THIS ISSUE:

- Seasonal Stack Effect Variations
- Elevator Shaft Issues
- Room Leakage

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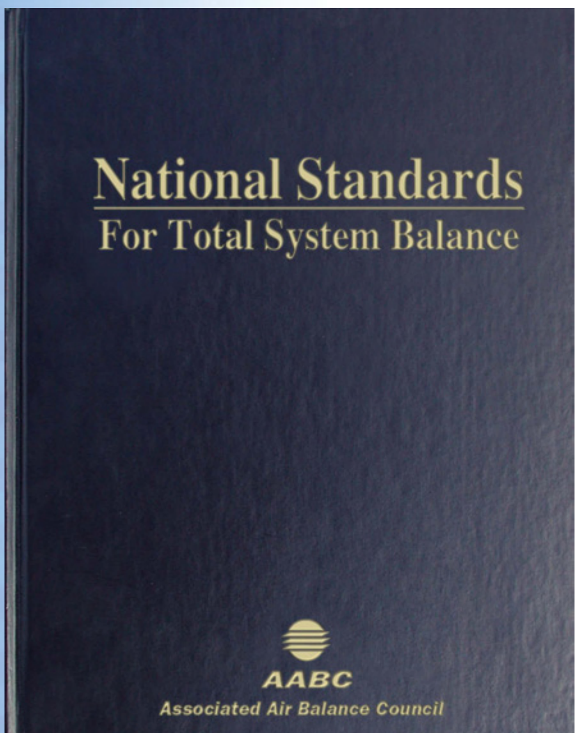

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From the Publisher

The summer 2012 issue of *TAB Journal* showcases the technical expertise of AABC test and balance engineers on topics related to building pressure and leakage, as well as others such as cleanrooms, hydronic circuit setters and automatic airflow regulators.

“Stack Effect and Building Pressurization,” by Terry Wright, TBE, of Engineered Air Balance Co., Inc., leads off the issue, discussing how cold snaps can throw a seemingly efficient building into a tailspin. Shawn Griffin, TBS, of Griffin Air Balance Ltd. discusses the importance of catching a mistake or oversight before it becomes a problem, such as air infiltration, in “Troubleshooting: Uncontrolled Air from Elevator Shaft.”

And Rudy Franz, TBE, of Senco Services, describes an unconventional cause of room leakage in specialty rooms in “Room Pressurization: You Never Know Where the Leak May Be.”

Other articles include “The Evolution of an Industry, and its Impact on Test and Balance.” By Jay Johnson, TBE, of Thermal Balance, who discusses how changes in the construction industry have affected TAB agencies. “Automatic Airflow Regulators in High-Rise Buildings” by Douglass Peterson, TBE, of RSAnalysis, Inc., provides insight on the efficiency and effectiveness of automatic balancing devices such as valves or dampers. Jonathan Young, TBE, of Southern Balance Company, explains the importance of choosing the correct circuit setter size in “Difficulties with Oversized Hydronic Circuit Setters.”

The research article “Airflow Validation in Cleanrooms” written by Antonio Estevez, Ph.D and Marco Fassina, discusses whether or not it is suitable to use propeller anemometers to measure airflow at the outlet of an absolute filter. And finally, this issue’s Tech Talk addresses a question regarding ANSI Balancing Quality Grades and which level will consume less energy.

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!

Stack Effect and Building Pressurization

Terry Wright, TBE
Engineered Air Balance Co., Inc.

Have you ever had a project where the testing and balancing was progressing more or less as normal, and then a cold snap blows in and your building goes haywire? Elevator doors at times won't close, air noise is whistling around the elevator doors, ground-floor building entrances are very cold, and the pressures in the building are fluctuating.

To make the experience even more interesting, let's say this situation is occurring in two multiple-story buildings across the street from each other. Furthering the challenge, the buildings are connected to each other and to other buildings by an enclosed pedestrian walkway and a corridor.

"The building pressurization sensor displayed a positive value to the outside, but air was being sucked into the building quicker than you could zip up a jacket."

A TAB agency experienced this very situation. For a week, the weather was in the teens and 20s at night, and few days were above freezing. This is not typical in the area, and a week is a little too long to refrain from complaining about a 46 ºF lobby, or elevator doors that fail to close.

Building A consists of 12 floors and is tied to multiple buildings by way of a corridor. Building pressurization had been reviewed on a prior occasion due to odors migrating into the building. It was believed that the lobby was negative and the odors were infiltrating the building at the entrance doors. The lobby was tested early on a Sunday morning with a 70 ºF outdoor temperature; and the lobby was actually positive to the outdoor by 0.01" w.c.

Building A was isolated from the connecting buildings by shutting the corridor doors that separated them. It was found that the connected buildings were positive to Building A.

This was not the case when the temperature outside dropped into the 20s and 30s. The building pressurization sensor displayed a positive value to the outside, but air was being sucked into the building quicker than you could zip up a jacket.

Nearby Building B consists of 24 floors and is connected to several other buildings by multi-floor connectors. The cold front hit and they were having issues with the elevator doors not closing at all times. The TAB agency was given a video of a piece of paper being dropped at an open elevator door and being sucked up into the elevator shaft without ever coming close to the ground.

The elevator shafts in Building B have a relief duct at the top of the elevator shafts. When the barometric dampers were inspected on one shaft, they were found to be open 100%. There was a measurable amount of airflow exiting the building but unfortunately it would require a ride on the top of an elevator to measure.

The dampers were temporarily closed and airflow at the elevator doors became neutral. The stairwell, which had been positive to the floor, also decreased to neutral. It appeared the problem was solved, but that inspection doors would be required at the other elevator shafts to check those barometric dampers. When those inspection doors are installed, the agency will return to see if their suspicions were valid.

Concerning the 46 ºF lobby in Building A, the thought was that the lobby had to be pressurized. To pressurize the lobby of Building A, 30,000 CFM of outside air was forced into its basement and the first and second floors. The pressure sensor indicated the lobby was greater than 0.20” w.c. positive to the outside, but the TAB agency’s reading was bouncing from negative to slightly positive to the outside. The pressure sensor was found to be referencing Level 12 and not outdoor.

What the agency discovered was that, due to connecting corridors, they were attempting to pressurize the buildings connected to Building A. When the doors were shut to the connected buildings, the lobby was positive to the outside.

Based on the findings in Building B, the elevator shafts were checked for relief dampers, but none were found. However, the stairwells were found to have barometric relief dampers, and these were partially open. The relief dampers may be contributing to the problem.

Unfortunately, the weather did not cooperate and things were back to normal before suspicions could be validated that the relief dampers were the culprit in Building B and also possibly in Building A. It will take another cold snap to be able to come to a final conclusion as to whether the reliefs are causing the pressurization problem -- but the agency does know that cold weather is the main factor.

Stack effect refers to a condition caused when a substantial difference exists between the outdoor and indoor temperatures, and building pressure is affected as a result. Normal stack effect occurs when the outdoor temperature is much cooler than a building’s indoor air temperature, which was the condition in this case. Air in the building has a buoyant force because it is warmer and therefore less dense than outside air. The buoyant force causes air to rise within building shafts and reduces the pressure on the lower level, resulting in air infiltration. When the outdoor temperatures are much warmer than the indoor conditions, the opposite occurs, and there is a downward flow of air known as a reverse stack effect.

The pressure differential value between the shaft and the outdoors increases with the temperature differential, which is at its maximum during the winter conditions. The following is an equation for calculating the expected pressure difference between the shaft at the top floor and the outside, and it does not account for shaft friction losses which should be negligible – such as wind effects, the atmospheric pressure difference inside the shaft, and the outside. These shaft friction losses are considered negligible, and the neutral plane is estimated midway from the ground floor and top of the building.

Equation 1

where $$\Delta p_{so} = K_s\,(1/T_o - 1/T_s)H$$

Δp_{so} = pressure difference from shaft to outside, inches w.c.

T_o = absolute temperature of outside air, ˚R

T_s = absolute temperature of air inside shaft, ˚R

H = distance above neutral plane

K_s = 7.64

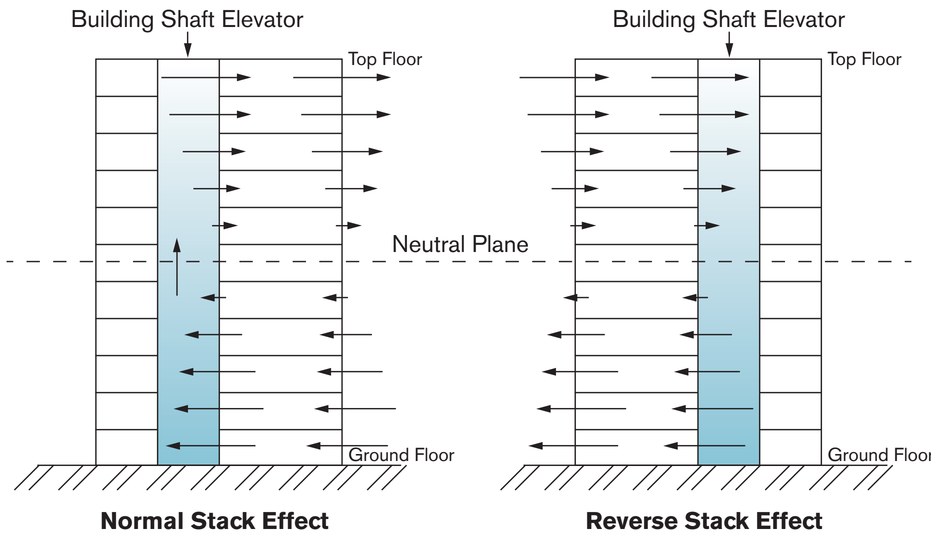


Figure 1- Air Movement due to normal and reverse stack effect.

Note: Arrows indicate direction of air movement

The neutral plane (Figure 1), the point where the inside pressure and the outside pressure are equal, is approximately midway between the ground floor and the top of the building when there are equal area openings on the top and bottom. Decreasing the opening on the ground level can shift the neutral plane towards the top of the building. This equation also assumes that the floor to floor leakage is uniform. A 100' building with uniform floor to floor leakage, with an inside temperature of 72 ºF and an outdoor temperature of 0 ºF, has a neutral plane 48.8' above the bottom of the building.

Building A is 180' in height while Building B is 360' in height. If the neutral plane on both buildings is midway of the building heights, the neutral plane is located on Building A at 90' and Building B at 180'. Using the equation on page 3 with an outdoor temperature of 30 ºF and an indoor temperature of 72 ºF, the pressure difference between the shaft and outside is 0.11" w.c. on Building A and 0.22" w.c. on Building B.

There may be issues correcting the pressure problems if the source of the problem is the elevator shaft relief. As required by the 2009 International Building Code, the highest points of elevator shafts spanning three or more floors must include specific relief openings. There are exceptions to the requirements and further review of the documents may be helpful to resolve the issue if it indeed is not required that all buildings have specific relief openings on the top of the elevator shafts.

There were many assumptions taken in the calculation above, as calculating all of the variables would be very time-consuming and difficult, to say the least. The important point is that stack effect exists, and will be more apparent in the winter months than in the summer. Knowing the bases of the air movement within the building and the factors that control it will enable personnel to troubleshoot the source of building pressure issues.

Room Pressurization:
You Never Know Where the Leak May Be

Rudy Franz, TBE, Senco Services

Operating rooms, isolation rooms, cleanrooms, and bio-safety labs are a few examples of rooms that require an offset between supply and return/exhaust air to establish a room's pressure. This pressure is required to either prevent contaminants from entering a given space, or to exfiltrate any particles that have already settled.

The volume of air required to meet a specified pressure is dependent on the amount of leakage built into a room. A room with a lay-in ceiling, for example, would leak more than one with a drywall ceiling. Door seals and sweeps also minimize leakage.

On a recently balanced vivarium with over 300 rooms, an unusual room leakage situation was encountered. A series of rooms were losing pressure when adjoining room doors were opened.

The rooms were supplied and exhausted through Phoenix valves, so the first step was retesting the valves to ensure their stability with doors open and closed. There were no significant changes in airflow noted.

While shuffling through some data sheets, one fell to the floor and happened to land on top of a floor drain. While reaching to pick it up, it was observed floating above the drain. After checking that drain and the ones in the other problem rooms, it was determined that airflow was moving up from the drains—apparently the traps were not filled with water.

After contacting the plumbing contractor to have them fill the traps, it was confirmed that airflow was no longer moving through the drain system. The room pressures were now remaining constant when adjoining doors were opened.

You never know where room leakage will be, and "floor drain traps" is now another item to add to the list.

TROUBLESHOOTING:

Uncontrolled Air From Elevator Shaft

Shawn Griffin, TBS, Griffin Air Balance Ltd.

An engineering firm enlisted the help of a test and balance agency to solve a client's air problem. A federal government entity had recently taken over the top floor of an older six-story office tower and was experiencing excessive air entering the floor via the elevator and stairway doors.

The building had two roof-mounted variable air volume 24,000 CFM A/C units, and had undergone some renovations five years earlier. The post-renovation balancing report stated that there were no dampers on the floors for the return system.

Each floor had VAV boxes for supply. The ceiling was used as a return plenum, with ducts connecting back to the AHU shaft and supply air duct running down the center. The AHU shaft is drywall and is used as return air plenum back to the units. The supply ducts take up one-half of the shaft space. There were silencers between the fan and the ductwork on the sixth floor. The velocity in the silencers was between 2,500 and 3,500 ft/min.

Using velocity measurements in front of the elevator as an indicator, measurements and testing revealed the following information:

- With the air handlers off, there was no drafting or pressure difference to the floor.
- With one unit running and the fans on low speed, the face velocity was approximately 100 ft/min (barely noticeable).
- With both units running, the face velocity reached as high as 650 ft/min. During this test, all the VAV boxes were never commanded to the highest airflow and, as such, the AHUs were not supplying their maximum airflow.

A survey was conducted with all VAV boxes set to maximum flow. The A/C units supplied 22,000 CFM and returned 20,000 CFM, which was close to design. To each floor, the units supplied 4,000 CFM, which was also close to design.

However, the return from each floor was dramatically reduced the farther it was away from the unit, from 4,000 CFM on the sixth floor, to as low as 1,600 CFM on the second and third floors.

A return traverse was conducted on the sixth floor for floors two through five. When compared to the total measured at the return openings for each floor, it showed a leakage of 4,380 CFM and 5,893 CFM. This volume of air must have been leaking through unsealed openings in the drywall plenum.

> *"Given that approximately 4,000 CFM of return from the sixth floor was being eliminated, the only conclusion that could be drawn was that the return air had to have a sizable leak somewhere above the sixth floor ceiling."*

Next, the returns on the sixth floor were blocked off. It was expected that this would help positively pressurize the floor and force the return to come through the main duct. However, the velocity in front of the elevator was only reduced to 500 ft/min—still very noticeable.

Given that approximately 4,000 CFM of return from the sixth floor was being eliminated, the only conclusion that could be drawn was that the return air had to have a sizable leak somewhere above the sixth floor ceiling. Upon inspection of the return air plenum and the rooftop air handling units, it was discovered that some openings between the plenum and the building roof were not sealed properly.

Once these holes were sealed as best as possible given the existing conditions (namely a lack of access), the return system was re-measured. Dramatic improvements were evident in the return airflows on floors two through five, and there was a reduction in the air returning back through the elevator shaft.

However, given that unsealed holes remained in the plenum above the sixth floor, the lowest level of air infiltration through the elevator shaft was achieved by capping the returns on the sixth floor.

In conclusion, while the information in the post-renovation report was accurate, it was not complete. A more thorough investigation revealed an installation issue that had not been previously known, and was the source of the majority of the air infiltration.

The Evolution of an Industry, and

Jay Johnson, TBE, Thermal Balance, Inc.

Over the last several decades, expectations and procedures seem to have changed significantly in the construction industry, particularly with respect to the test and balance process.

In the 1970s, the engineering community provided a strong site presence. The site inspectors monitored installations, verified system start-ups, and basically tracked their project—in the office and in the field—from start to finish.

This site presence was also very beneficial for test and balance agencies because systems were verified as being correctly installed, started, and running as designed. Projects in these "days of old" were much more likely to be ready for balancing when the TAB agency was notified. Of course, TAB agencies still found issues, but in general it was not to the extent that is common today.

Over the last 20 years or so, the mechanical world experienced many changes that from all appearances, seemed to be ultimately driven by financial concerns. Some examples of these changes include the following:

- The full-time engineering site presence was eliminated.
- System start-ups by the manufacturer's representative were eliminated and performed in-house by the installing contractors.
- General contractors or construction managers were assumed to provide the assistance required to properly monitor the installation process.
- A new frame of mind set in, and it became expected that TAB agencies would/should uncover and help resolve issues found in the field.

Taken together, these changes above resulted in longer completion times and required many additional site visits by the balancing agency. These extra trips were also expected to be performed for free. The point is that the end result of these changes has frequently been to push problem identification and resolution increasingly onto the test and balance agency. The act of balancing now encompasses a great deal more.

TAB agencies are not alone in bearing the additional costs associated with poor installations; the design community and others also bore their share, in the form of additional site trips and other office time mandated by poor construction.

"The bottom line is that whether a building is commissioned or not, in order for owners to get the buildings

its Impact on Test & Balance

Since the mid-1990s, the commissioning process has evolved to help restore an important piece of the puzzle. Present-day commissioning agents do perform more services than the on-site engineers of old, but a certain portion of their scope duplicates the tasks that the on-site engineers once performed.

Being on the receiving side of quality commissioning services was a refreshing change. Minimized site visits, troubleshooting, and actually being able to begin TAB work when notified, were just a few of the benefits of commissioning. But, as with all good things, this comes at an additional expense to the owner—and inevitably there is pressure to reduce that cost.

Reducing costs in the commissioning process means eliminating various tasks. Most of the time, the tasks that are targeted deal with:

- Monitoring installation;
- Witnessing startups; and
- Verifying that all control points have been correctly entered and all systems are communicating.

Those dollars saved in the commissioning process, however, can result in additional responsibilities, time, and expenses—but not necessarily fees—for the TAB agency. The bottom line is that whether a building is commissioned or not, in order for owners to get the buildings that they paid for as they were designed, all parties need to be held accountable during the installation phase.

In conclusion, the TAB industry cannot continue to bear the additional work that always seems to fall in its lap. It seems as though the construction industry has forgotten that TAB agencies only bid to balance the systems. You could say that reputable TAB agencies have brought this on themselves to some degree, because many of them know if they don't resolve certain issues, regardless of whether they are in the TAB scope, they will never get done.

A question many TAB agencies have probably asked themselves more than once is: "What if this time we make only one visit to the site when told the systems are ready for TAB, then we balance each system the best we can, and turn in the report—warts and all?" The engineering community (not to mention the contractors) might not like what they see, but it might help them to understand the difficulties from the perspective of the TAB agency, and help the industry evolve again to a better state that is beneficial for everyone.

that they paid for as they were designed, all parties need to be held accountable during the installation phase."

Automatic Airflow Regulators in High-Rise Buildings

Douglass Peterson, TBE, RSAnalysis, Inc.

As technology advances in the HVAC industry, there seems to be an increase in the number of "automatic balancing devices," meaning a balancing device (whether a valve or a damper) that is factory set to deliver a predetermined quantity. In theory, this predetermined quantity only needs to be verified in the field and is generally not adjusted by the TAB agency.

In the 1960s, Griswold released a hydronic autoflow balancing valve that was the first of its kind. Now it seems that just about every balancing valve manufacturer makes an autoflow balancing valve of some type, and they are increasingly becoming the norm. As the industry has progressed, so has the need for some type of constant airflow regulator.

There are a few different types available. The ones we see most often, and mainly in laboratories, are Phoenix valves, which are not necessarily factory pre-set, but do operate along the same principles. Also fairly common are constant flow dampers installed in larger kitchen hood systems, and the smaller low-flow toilet exhaust systems in high-rise buildings.

There is one common factor between autoflow balancing valves and self-regulating dampers: they need to operate within a specific differential pressure range across the regulating device to satisfy the design criteria. As most in the industry are more familiar with the hydronic autoflow valve, this article will discuss the positive and negative aspects of the constant airflow regulator damper that you would find in a typical hotel, apartment, or condominium.

Case Study

As recently as 2009, a large hotel/condo project in the Las Vegas area had a constant exhaust register system installed in the toilet inlets in each room. They used a constant airflow regulator (CAR) damper mounted in an exhaust grille.

Each room also had a supply grille for outside air ventilation that also served as part of the hotel/tower fire life safety system. The supply grilles did not have any type of constant air device; they were connected to a constant volume air handler.

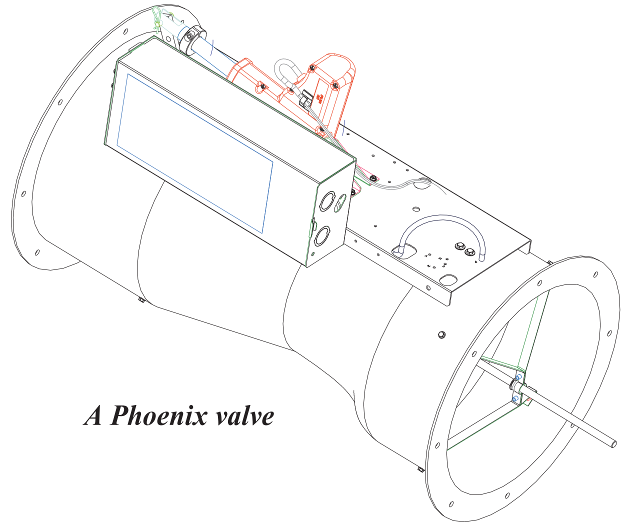
At the time of the balance, these dampers were relatively new, so available background research was almost nonexistent with the exception of what was posted on the manufacturer's website. The manufacturer showed buildings up to 12 stories in height, and airflow readings before and after the CAR dampers were installed in the exhaust systems. The manufacturer also showed the system performing substantially better with the CAR II dampers installed as opposed to no damper at all. Finally, the website said that there was no risk of dust deposits or obstructions because the CAR-II damper had no airways subject to clogging. The CAR dampers installed on this job were required to operate within 0.20" to 0.80" static pressure behind the damper in the duct to deliver design CFM.

"In taller buildings the pressures at the top and the bottom of the shaft are probably outside of the operating range of the CAR damper."

Negative aspects of the damper: It was determined that at the lower end of the static range of 0.20" to 0.25", the measured CFM varied from design to -15%. On the top end of the range, which was 0.75" to 0.80", the measured CFM varied from design to +20%.

When the TAB agency arrived at the jobsite, the contractor was finishing the installation of the dampers, and it was observed that the dampers installed on the already completed floors were relatively clean. However, the dampers installed on floors where construction was taking place were very dirty. After measuring the dampers, it was clear that they needed to be cleaned by the contractor prior to test and balance being performed. After being cleaned, the dampers worked much better.

The manufacturer's website showed the dampers in a 12-story building; this may be because in taller buildings the pressures at the top and the bottom of the shaft are probably outside of the operating range of the damper. In the case of this building, a 50-story hotel/condo tower, at the top of the shafts there was



A Phoenix valve

over 1.0" of static pressure closer to the fan and at the bottom of the shafts, while at the farthest point from the fan there was less than the required 0.20".

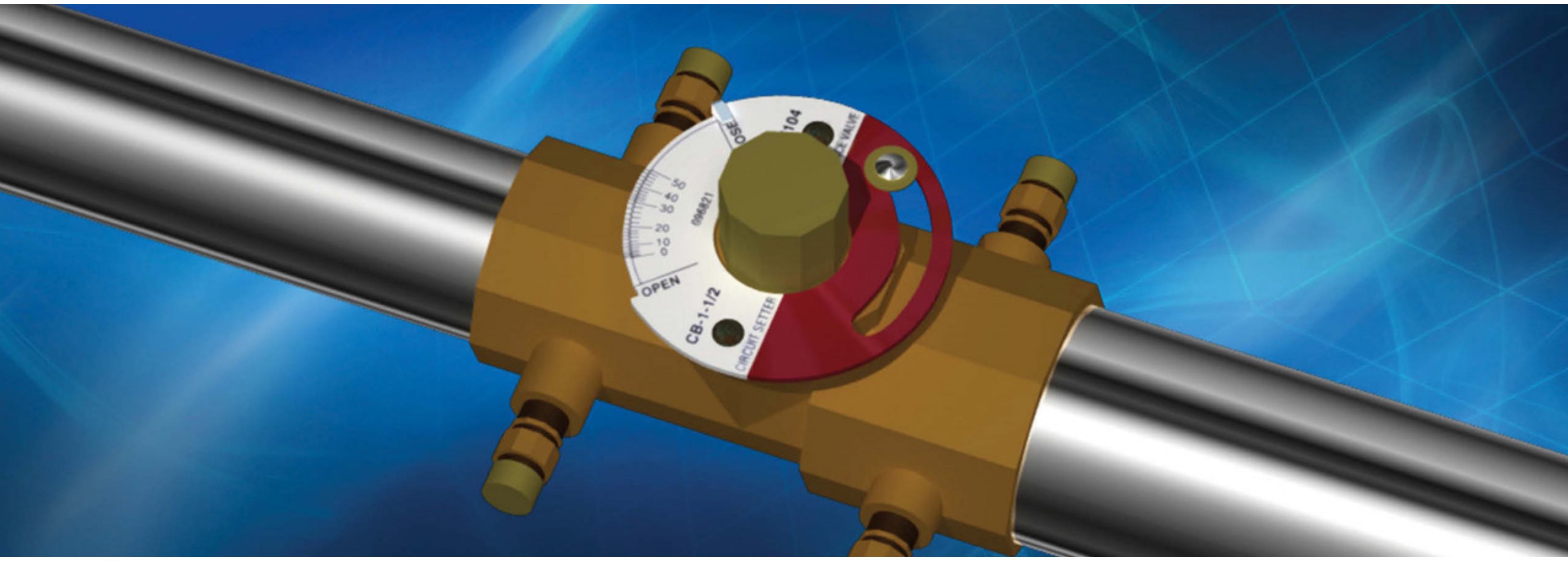
To compensate for the low static pressure at the bottom of the shafts and the excessive static pressure at the top of the shafts, the mechanical contractor had to remove the bottom 5 to 10 floors of dampers and install manual dampers in their place. Manual dampers and blank-offs were also added behind the CAR dampers on the top 7 to 10 floors so they could be throttled to attain the proper static pressure upstream of the CAR damper.

Another issue in this building was the way the shaft itself was designed. The shaft was tapered, which the agency believes added to the static pressure loss down the shaft because on another hotel tower of the same height, the shafts were constant size throughout and the static pressure loss throughout the shafts was minimal. Further compounding the shaft loss was that there were exhaust boots on every exhaust grille that stuck into the exhaust shaft, reducing the shaft size.

Positive aspects of the damper: The biggest positive to this damper is its ability to self-regulate against effects such as wind or stack effect. In Las Vegas, the winter stack effect can really take its toll on high-rise towers. The summer reverse stack effect is very noticeable as well. These self-regulating dampers can have a positive impact, as they will compensate on the ever changing pressures in the building.

Another positive is that the dampers appear to be very reliable. There were very few dampers that needed to be replaced because they did not operate per their design intent. When the static pressure in the shaft was near the middle of the operating range, the airflow was almost always within 5% of design. The TAB agency made cardboard cones that were used in the field with rotating vane anemometers, as opposed to a flow hood, due to the small size and low air volume of the grilles. Finally, the CAR dampers are relatively quiet. When the static pressure was near 1.0" as mentioned above, there was noise, but not as loud as one would expect.

Difficulties with Oversized Hydronic



In today's design-build industry, it has become very common to find circuit setters sized to match the pipe sizes to every hydronic coil. This built-in balancing problem then requires using a gauge or differential pressure meter accurate enough to measure extremely low pressure drops. Oversized valves may also prohibit balancing to AABC standards, which require leaving at least one valve fully open.

For example, a recently completed project had thirty-six, ¾" circuit setters requiring 0.9 to 2.8 GPM. This system also had two thermostatically controlled 3-way valves to maintain space temperatures, and a VFD-controlled hot-water pump.

The coils requiring 0.9 GPM have a required circuit setter pressure drop of only 0.52" w.c., so accurate determination of water flow in the wide-open position is virtually impossible. In fact, the initial pressure drop readings were less than the lowest graduation of the first test gauge, suggesting that there was low flow; yet the pump curve indicated substantially higher than design total flow.

A different digital manometer capable of reading very low pressures was then brought in, however even with this meter's stated accuracy of the largest of +/-1% of the reading or 0.036 psi, measurement errors can be +/-70% of actual flow. In this case, there was no way to accurately record initial water flows, and to set the VFD to operate at the lowest possible system head.

As is more common than it should be in the TAB industry, this project was awarded after the systems were already installed. The only method to measure flows was by pre-setting every circuit setter closed at an arbitrary point, at which the required pressure drop would be increased enough to obtain higher differential pressure measurements, and more accurately, determine the individual coil flows relative to design. The resulting settings on all valves were between 50% and 75% closed.

To obtain a pressure differential at approximately 2.0' w.c., many of the valves had to be cut down further to approximately 12% open. If the situation had been correctly assessed initially, the labor could have been cut in half by pre-setting every circuit setter.

This problem could have been eliminated at the design stage by providing accurate feedback to the mechanical engineers. In this circumstance, the customer agreed with the findings and was satisfied with the throttled circuit setter conditions. Designing circuit setter sizes consists of selecting a valve size that requires a pressure drop of approximately 2.0' w.c. with the valve wide open, as per some device manufacturers.

In this project, all of the valves could have been selected for ½" sizes to avoid errors in the flow determinations, with a potential cost savings on the device purchases, as well. The problem could have been compounded on larger projects requiring commissioning oversight and balance verifications. It may also reflect poorly on the engineer and the balancing contractor when questions arise at the very end of the project about potential "inaccuracies" of the test and balance measurements.

Circuit Setters

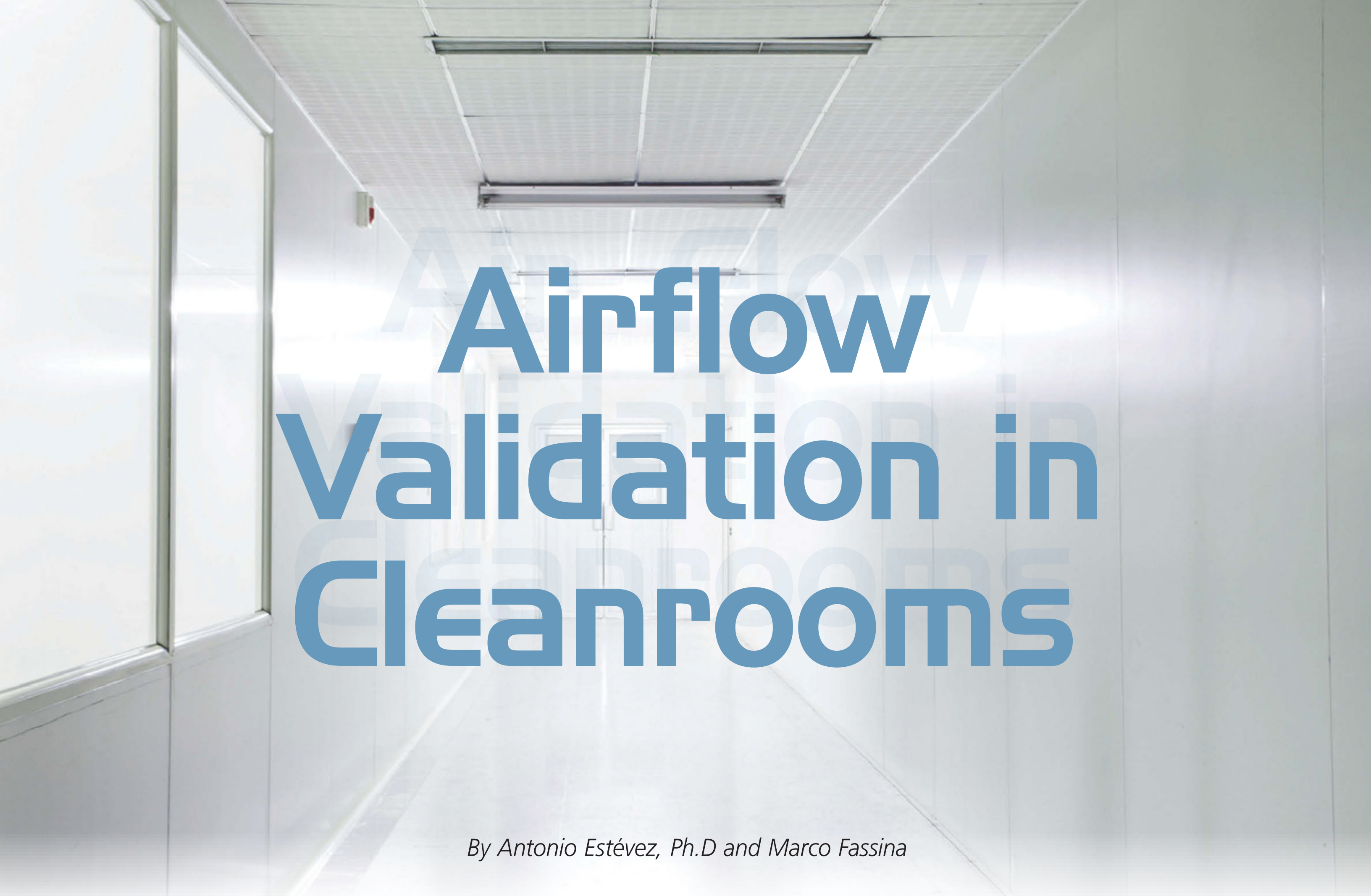
Jonathan Young, TBE
Southern Balance Company

Oversized circuit setters that end up being shut-down 75% or more during the balancing process can also become a very effective strainer, catching every piece of floating debris. If they stop up completely, maintenance technicians will normally open up the valves, flush out the debris, and destroy the proportional system balance. Proper resetting of many valves could be inadequate without the use of a good differential pressure meter, due to high pressure drop fluctuations with even minute movement of the valves.

Later, facility maintenance technicians may also erroneously assume that the systems were not balanced properly, as they see most of the circuit setters significantly throttled, even with properly marked valves and setting of memory stops. This can lead to some excess costs involved with rehashing and/or demonstrating proper balancing efforts.

In conclusion, efficient project hydronic balancing should contain at least the following points:

- During pre-engineering/design-review process, take a look and research the expected pressure drops of the circuit setters based on the valve submittal. Address any potential flow-determination accuracy issues up front to avoid complications and discussions at the end of the project.
- Develop a pre-setting agenda to establish an accurately measurable differential pressure when faced with over-sized valves.
- Encourage engineers to follow the manufacturer's recommended guidelines for circuit setter selection. One circuit setter manufacturer even documents on their balancing wheels that the highest accuracies are obtained when the valves are fully opened and accuracies diminish as the valve is closed off.



Airflow Validation in Cleanrooms

By Antonio Estévez, Ph.D and Marco Fassina

This article discusses whether or not it is suitable to use propeller anemometers to measure airflow at the outlet of an absolute filter; and in particular, whether they are appropriate for use as measuring instruments in the validation of cleanrooms in air conditioning systems.

Airflow rates found by means of an anemometer are less accurate than rates found through the use of a differential pressure device embedded in the duct. Due to this variation in rates from the two devices, the authors designed an experiment to estimate the error involved when the anemometer procedure is used in a simulation for normal data-gathering conditions in cleanrooms.

Normal Procedure for System Validation

With regard to the normal procedure for system validation, the service provider responsible for starting up and validating the air conditioning system in a cleanroom follows an established procedure settled with the client in advance. This procedure is well established in UNE-EN 12599 and UNE-EN ISO 14644-3. The validation service provider and the client must agree on and clearly establish the measuring instruments that will be used, in addition to the procedures that will be used to measure the airflow at the absolute filter's outlet and in the supply ducts.

The duct's airflow is regularly measured with a Pitot tube, while the filter outlet flow is normally measured with a propeller anemometer. This article focuses on the latter situation.

A fundamental process exists for measuring flow at the filter output. Though there is a wide variety of absolute filter models and sizes on the market, the sizes most regularly used are the 600 x 300-mm, 600 x 600-mm, and 1200 x 600-mm versions.

The airflow issuing from the filter is found the same way in all three cases, and includes the following steps: the filter's area is divided into cells, and the air speed in each cell is measured utilizing a propeller anemometer, which is usually set at a distance of roughly 20 centimeters from the filter; next, the average speed is calculated, and airflow is determined by multiplying the average speed by the area of the filter. The number of cells into which the filter must be divided depends on several parameters, including the filter's size, the number of measurement repetitions, and other details of the measuring process.

Once the flow issuing from each of the filters has been measured, the sum of each filter's individual flows is assigned as the total flow of air delivered into the room.

The flow blown into the supply ducts by the machine, on the other hand, is measured by a Pitot tube. The Pitot tube is inserted into the main duct if the main duct does not branch out before reaching the first filter. If the main duct splits into secondary branches, the customary practice is to measure airflow with the Pitot tube in each secondary branch, a procedure that is particularly useful for checking and ensuring that the system is balanced. In this case, the sum of the flows for the different secondary branches is assigned as the flow of air blown out by the machine.

Theoretically, the airflow delivered into the room and the airflow blown out into the main duct by the machine ought to be equal.

However, when a system is first started up or when it undergoes regular maintenance, discrepancies are often found between the figures for the two flows, which ought to match within the applicable margins of error. Often the sum of the flow found by the anemometer at the filter output is up to 20% to 30% greater than the sum of the flow measured by the Pitot tube in the ducts.

The professionals who start up and validate HVAC systems commonly agree that the Pitot-tube-based measuring procedure is more accurate than the procedure based on the propeller anemometer. In fact, when the discrepancy between the airflow rates calculated by the two procedures is large, the flow rate found by the Pitot tube is usually assigned to the system.

However, in many cases when a system is started up, validated, or simply undergoing maintenance, the flow blown into the cleanroom can only be found by measuring the filter output flow rate using a propeller anemometer. In these cases, the measured airflow rate is probably subject to a high margin of error, and in all probability it is greater than the real airflow.

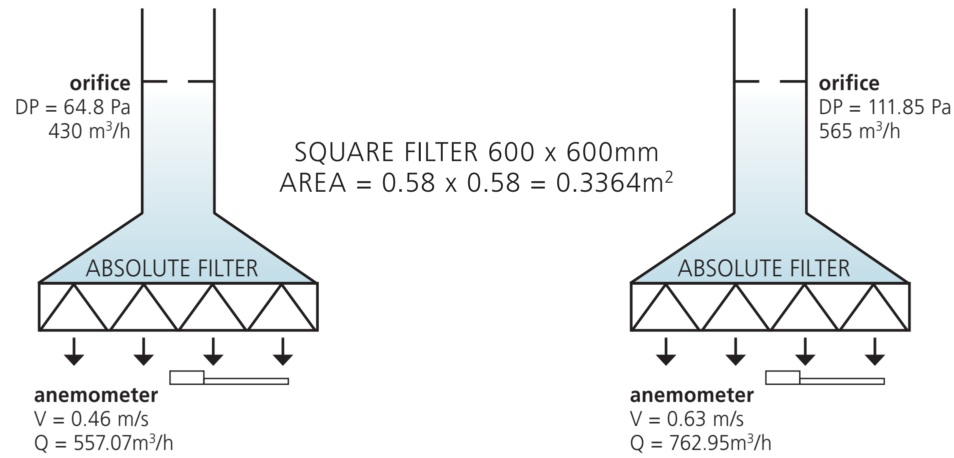


Figure 1. Comparison of measurements between an anemometer and a differential pressure device.

This fact has one obvious, important implication: the rate of airflow blown into the cleanroom is falsely recorded as greater than it really is. If this rate is accepted without applying a correction factor of any sort, the anticipated rate of air turnover per hour cannot be guaranteed, and the air quality inside the cleanroom may be affected as a result *(see Figure 1).*

Motivated by the situation described above, and with the objective of quantifying the error that exists when a propeller anemometer is used to find airflow in cleanrooms, the authors designed a laboratory experiment that attempts to simulate real conditions in a system. The details of this experiment are presented in the next section.

Experimental Set-Up

The authors designed a simple laboratory experiment to ascertain the precise flow of air delivered to an absolute filter, and compare it to the determined flow rate using a propeller anemometer to measure the airflow at the filter output. The authors strove to recreate the conditions that usually accompany the data-gathering process in a cleanroom. The experimental set-up was assembled in the physics laboratory at the Madrid Polytechnic University School of Aeronautics, and is described below.

1. A fan powered by an adjustable voltage source provided an airflow at an adjustable flow rate. This airflow was channeled toward the frame of an absolute filter through a straight, circular metal duct having a diameter of d = 250-mm and a length of l = 6 m *(see Figure 2).*
2. To determine the airflow rate delivered to the filter accurately, an orifice flowmeter was designed, built, and installed inside the duct. This flowmeter was designed pursuant to UNE-EN ISO 5167 (parts 1 and 2)[3,4]. It basically consisted of a flow straightener installed between the start of the duct (the connection to the fan) and the orifice plate; the orifice plate itself, inserted into the conduit at a distance of d = 4.5 m from the start of the duct, with an orifice whose diameter is d = 150-mm; and two pressure ports situated asymmetrically before and after the plate, with probes connected to a certified, calibrated differential pressure gauge. The flow rates measured with this flowmeter were subject to a relative error of less than e = 0.5% (margin of error calculated pursuant to UNE-EN ISO 5167-2[4]).
3. The frame containing the absolute (HEPA H14) filter was connected by a flexible pipe to the terminal end of the conduit. Work was done with two filter sizes, 600 x 300-mm and 600 x 600-mm.

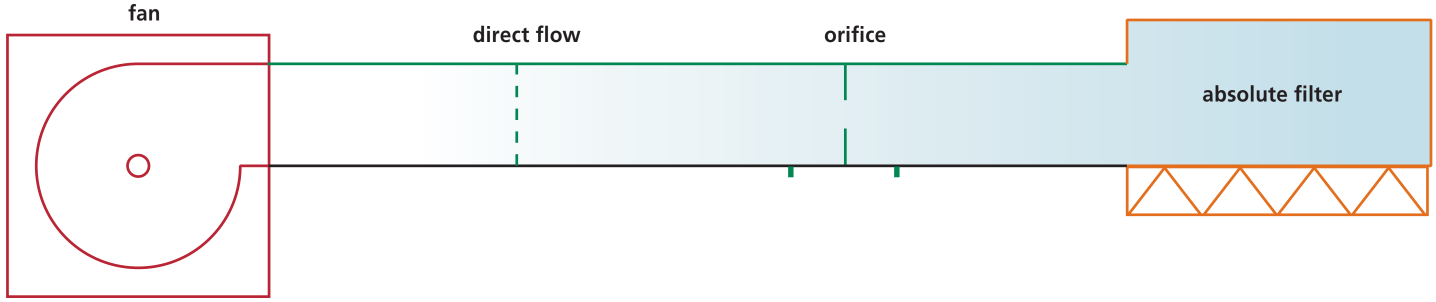


Figure 2. Diagram of a device for measuring airflow

4. The propeller anemometer rested on a metal structure situated beneath the filter. The anemometer was set up horizontally, with the windmill set parallel to the filter's surface, at a distance of d = 20 cm. The anemometer could shift freely in the horizontal plane by means of a device with a pair of guides, which was anchored to the structure. When gathering data, the anemometer was situated just beneath each of the cells into which the filter was divided and was then used to find the air speed in that cell. The anemometer used was certified and calibrated.
5. Lastly, a temperature probe was installed in the duct, downstream of the orifice plate. The temperature reading taken inside the duct enabled the air density to be calculated.

As indicated before, the data-gathering process included recording measurements for two absolute (HEPA H14) filters, a rectangular 600 x 300-mm filter and a square 600 x 600-mm filter. In both cases, measurements were taken for a set of airflow rates or, in equivalent terms, for a set of air speeds at the filter outlet. In both cases these speeds fell within the margin of $v \approx 0.3\text{-}0.9$ m/s.

Each of the filters was imaginarily divided into cells; n = 10 cells in the case of the rectangular filter and n = 25 cells in the case of the square filter. The air speed was measured just beneath each of the cells, with the anemometer situated at a distance of d = 20 cm. These readings were used to find the airflow rate according to the following expression:

$$Q_{anemometer} = \frac{\Sigma v}{n}\ 3600s$$

Where

Q = volumetric flowrate in m³/h

v = air speed in m/s

n = number of cells (10 for the rectangular filter, 25 for the square filter)

s = net area of a rectangular filter in m^2 (0.1595)

s = net area of a square filter in m^2 (0.3364)

While the airflow was measured using the propeller anemometer, another measurement was taken simultaneously using the flowmeter. In this case, the differential pressure of the air on passing through the orifice plate (Δp) was measured. In terms of this magnitude, the airflow was calculated according to the following expression:

$$Q_{flowmeter} = \frac{c}{\sqrt{1-\beta^4}}\ \varepsilon\ \frac{\pi}{4}\ d^2\sqrt{2\,\Delta p\rho}$$

Where

Q = airflow rate (m^3/h)

c = discharge coefficient of flow rate (0.6)

ε = air expandability factor (ε=1)

d = orifice diameter (d=150mm)

D = duct diameter (D=250mm)

β = diameter ratio (β=d/D)

ρ = density of air

Δp = differential pressure across the orifice plate

Findings

The two tables on page 17 show the airflows measured for each of the two HEPA H14 filters. The flows measured by both procedures, via anemometer and via orifice flowmeter, are shown for different air speeds at the filter outlet *(see Figures 3a and b).*

From these two tables, the conclusion may be drawn that, in the given speed interval, the flow reading taken with the propeller anemometer is always greater than the reading taken with the orifice flowmeter. The difference between the two flow readings increases with speed *(see Figure 4).*

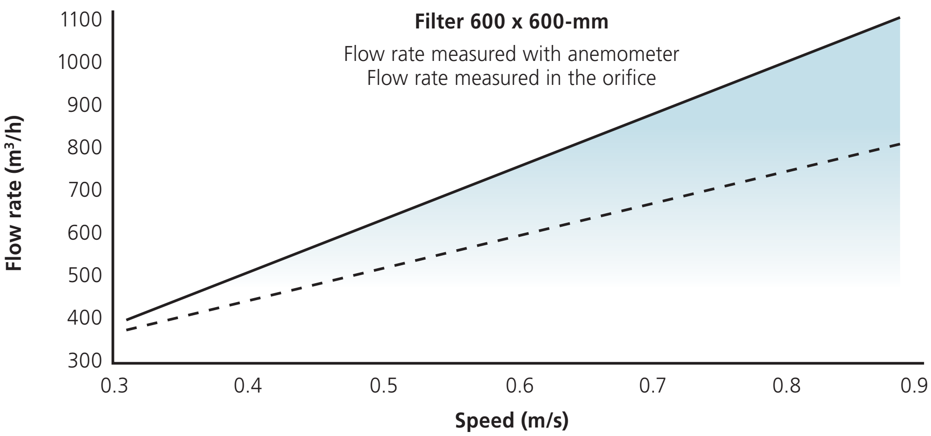
This table shows the correlation between the two flows ($Q_{flowmeter}/Q_{anemometer}$) depending on the speed of the airflow at the filter outlet.

From an analysis of this table, an estimate of the order of magnitude of the difference in readings between the two flows can be reached. In the target area in terms of speed, the difference between the two flows lies within the 15% to 20% range for the square (580 x 580-mm) filter, while it lies within the 20% to 25% range for the rectangular (580 x 275-mm) filter.

Conclusion

The results discussed in this article confirm that the propeller anemometer is not the most suitable instrument for measuring airflow at the outlet of an absolute filter.

a.



b.

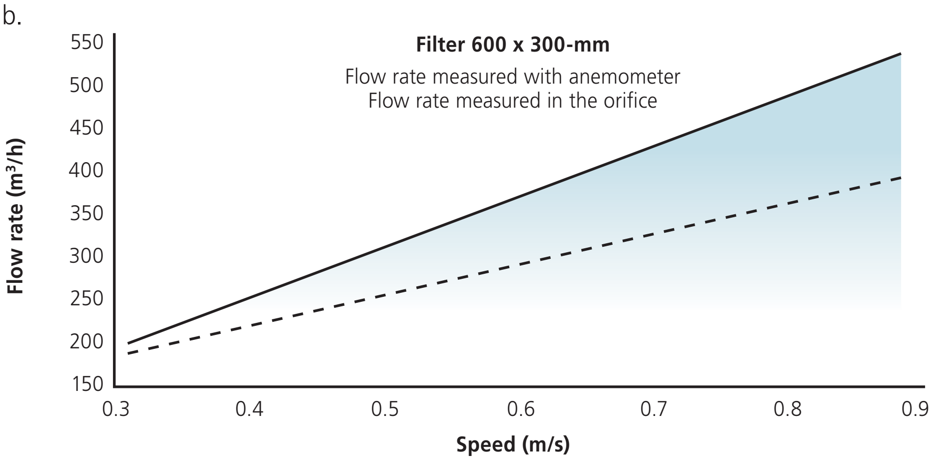


Figure 3. a and b flow readings for two HEPA H14 filters.

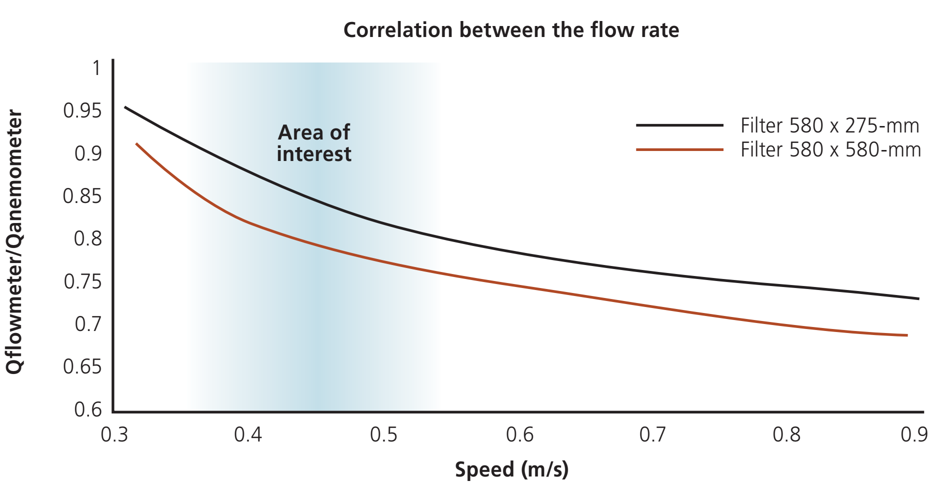


Figure 4. Relationship between airflow and air speed.

Measuring the airflow issuing from an absolute filter by means of a propeller anemometer leads to an overestimate that, in round terms, we estimate at between 15% and 25%. This figure is too high to allow the readings found in the measuring process to be regarded as valid.

It may be concluded that propeller anemometers should no longer be used as measuring instruments but should instead be replaced by balometers. For situations when there is good reason for using a propeller anemometer to find the airflow rate, its use should be supplemented by simultaneous flow measurements in the duct using a Pitot tube, which enables a correction factor to be applied. In our opinion, the propeller anemometer should be used only to measure airflow speed, not to find the airflow rate (unless a correction factor is applied).

Acknowledgements

We would like to thank General Filter Italia for having provided the two HEPA H14 filters.

References

[1] EN 12599 2000: Test procedures and measuring methods for handing over installed ventilation and air conditioning systems

[2] UNE ISO 14644/3: Cleanrooms and associated controlled environments Part 3: Test methods

[3]UNE ISO 5167/1: Measurement of fluid flow by means of pressure differential devices inserted In circular cross-section conduits running full. Part 1: General principles and requirements.

[4] UNE ISO 5167/2: Measurement of fluid flow by means of pressure differential devices inserted in circular cross-section conduits running full. Part 2: Orifice plates.

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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.

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

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Vibration Testing & Balance Quality Grade

Q

QUESTION: ***Our engineering firm was recently asked by a client whether they had requested that all equipment be dynamically balanced to ANSI S2.19 Balance Quality Grade 2.5, or if the specification only covered ANSI S2.19 Balance Quality Grade 6.3. They indicated that a grade 2.5 balance is a higher quality level which will increase bearing life and consume less energy. Can you shed any light on this?***

A

AABC: The specification is based on the Standards and both only speak of vibration testing. Neither makes any reference to ANSI in the vibration testing.

The gentleman is speaking of Dynamic Balancing, and that is the responsibility of the manufacturer. AMCA and fan manufacturers reference ANSI 2.19 for limits on residual unbalance. The balance quality grade for fans depends on the fan application category, ranging from G16 to G1.0 for fan application categories BV-1 to BV-5. This should all be part of the equipment section of the specs and not the balancing section.

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

A

AABC: To be clear, Balance Quality Grade does not refer to the quality of the test and balance work, but how well-balanced system components are, which affects vibration. ANSI S2.19-75, "Balance Quality Requirements of Rigid Bodies" adopted the ISO 1940/1 and it addresses the standard of Balance Quality Grade (G) for rigid motors. Balance Quality Grade for rigid rotors is the product of specific unbalance (e) and rotor maximum service angular velocity (ω). Equation: $G = e \times \omega$

Most HVAC applications are acceptable with a G 6.3 grade unless it is a medium or large size armature with special requirements. i.e. an installation with low structure-borne noise limits. Efforts to provide equipment with a G 2.5 grade may not be cost effective for smaller applications.

The balance grade standard gives everyone a reference for balance quality expectations, but it is up to designers to interpret the need for their application. For recommended Balance Quality Grades for specific equipment, per ISO 1940/1, see: http://tinyurl.com/BalanceQualityGrades.

— *Brian LaFleur, Vibration Analyst Category II, Engineered Air Balance Co., Inc.*

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